

Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
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2nd Edition, 2008

Chapter 1
INTRODUCTION AND OVERVIEW

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Thermodynamics, Heat Transfer, and Fluid Mechanics

1-1C Classical thermodynamics is based on experimental observations whereas statistical thermodynamics is based on the average behavior of large groups of particles.

1-2C On a downhill road the potential energy of the bicyclist is being converted to kinetic energy, and thus the bicyclist picks up speed. There is no creation of energy, and thus no violation of the conservation of energy principle.

1-3C There is no truth to his claim. It violates the second law of thermodynamics.

1-4C A car going uphill without the engine running would increase the energy of the car, and thus it would be a violation of the first law of thermodynamics. Therefore, this cannot happen. Using a level meter (a device with an air bubble between two marks of a horizontal water tube) it can be shown that the road that looks uphill to the eye is actually downhill.

1-5C Thermodynamics deals with the amount of heat transfer as a system undergoes a process from one equilibrium state to another. Heat transfer, on the other hand, deals with the rate of heat transfer as well as the temperature distribution within the system at a specified time.

1-6C (a) The driving force for heat transfer is the temperature difference. (b) The driving force for electric current flow is the electric potential difference (voltage). (c) The driving force for fluid flow is the pressure difference.

1-7C Heat transfer is a non-equilibrium phenomena since in a system that is in equilibrium there can be no temperature differences and thus no heat flow.

1-8C No, there cannot be any heat transfer between two bodies that are at the same temperature (regardless of pressure) since the driving force for heat transfer is temperature difference.

1-9C The absolute minimum energy needed to move this car horizontally is zero since the acceleration is zero. Note that $\text{Force} = \text{mass} \times \text{acceleration}$ and $\text{Work} = \text{Force} \times \text{distance}$.

Mass, Force, and Units

1-10C Pound-mass lbm is the mass unit in English system whereas pound-force lbf is the force unit. One pound-force is the force required to accelerate a mass of 32.174 lbm by 1 ft/s^2 . In other words, the weight of a 1-lbm mass at sea level is 1 lbf.

1-11C In this unit, the word *light* refers to the speed of light. The light-year unit is then the product of a velocity and time. Hence, this product forms a distance dimension and unit.

1-12C There is no acceleration, thus the net force is zero in both cases.

1-13E The weight of a man on earth is given. His weight on the moon is to be determined.

Analysis Applying Newton's second law to the weight force gives

$$W = mg \longrightarrow m = \frac{W}{g} = \frac{180 \text{ lbf}}{32.10 \text{ ft/s}^2} \left(\frac{32.174 \text{ lbm} \cdot \text{ft/s}^2}{1 \text{ lbf}} \right) = 180.4 \text{ lbm}$$

Mass is invariant and the man will have the same mass on the moon. Then, his weight on the moon will be

$$W = mg = (180.4 \text{ lbm})(5.47 \text{ ft/s}^2) \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm} \cdot \text{ft/s}^2} \right) = \mathbf{30.7 \text{ lbf}}$$

1-14 The interior dimensions of a room are given. The mass and weight of the air in the room are to be determined.

Assumptions The density of air is constant throughout the room.

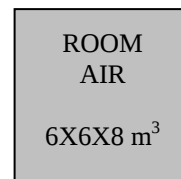
Properties The density of air is given to be $\rho = 1.16 \text{ kg/m}^3$.

Analysis The mass of the air in the room is

$$m = \rho V = (1.16 \text{ kg/m}^3)(6 \times 6 \times 8 \text{ m}^3) = \mathbf{334.1 \text{ kg}}$$

Thus,

$$W = mg = (334.1 \text{ kg})(9.81 \text{ m/s}^2) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) = \mathbf{3277 \text{ N}}$$



1-15 The variation of gravitational acceleration above the sea level is given as a function of altitude. The height at which the weight of a body will decrease by 1% is to be determined.

Analysis The weight of a body at the elevation z can be expressed as

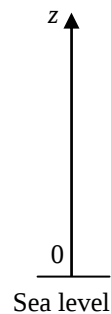
$$W = mg = m(9.807 - 3.32 \times 10^{-6} z)$$

In our case,

$$W = 0.99W_s = 0.99mg_s = 0.99(m)(9.807)$$

Substituting,

$$0.99(9.81) = (9.81 - 3.32 \times 10^{-6} z) \longrightarrow z = \mathbf{29,539 \text{ m}}$$



1-16E The mass of an object is given. Its weight is to be determined.

Analysis Applying Newton's second law, the weight is determined to be

$$W = mg = (10 \text{ lbm})(32.0 \text{ ft/s}^2) \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm} \cdot \text{ft/s}^2} \right) = \mathbf{9.95 \text{ lbf}}$$

1-17 The acceleration of an aircraft is given in g 's. The net upward force acting on a man in the aircraft is to be determined.

Analysis From the Newton's second law, the force applied is

$$F = ma = m(6g) = (90 \text{ kg})(6 \times 9.81 \text{ m/s}^2) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) = \mathbf{5297 \text{ N}}$$

1-18 CDEES A rock is thrown upward with a specified force. The acceleration of the rock is to be determined.

Analysis The weight of the rock is

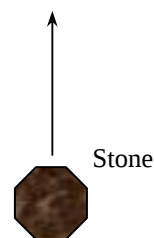
$$W = mg = (5 \text{ kg})(9.79 \text{ m/s}^2) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) = 48.95 \text{ N}$$

Then the net force that acts on the rock is

$$F_{\text{net}} = F_{\text{up}} - F_{\text{down}} = 150 - 48.95 = 101.05 \text{ N}$$

From the Newton's second law, the acceleration of the rock becomes

$$a = \frac{F}{m} = \frac{101.05 \text{ N}}{5 \text{ kg}} \left(\frac{1 \text{ kg} \cdot \text{m/s}^2}{1 \text{ N}} \right) = \mathbf{20.2 \text{ m/s}^2}$$



1-19 EES Problem 1-18 is reconsidered. The entire EES solution is to be printed out, including the numerical results with proper units.

Analysis The problem is solved using EES, and the solution is given below.

```
W=m*g "[N]"
m=5 [kg]
g=9.79 [m/s^2]
```

"The force balance on the rock yields the net force acting on the rock as"

```
F_net = F_up - F_down "[N]"
F_up=150 [N]
F_down=W "[N]"
```

"The acceleration of the rock is determined from Newton's second law."

```
F_net=a*m
```

"To Run the program, press F2 or click on the calculator icon from the Calculate menu"

```
SOLUTION
a=20.21 [m/s^2]
F_down=48.95 [N]
F_net=101.1 [N]
F_up=150 [N]
g=9.79 [m/s^2]
m=5 [kg]
W=48.95 [N]
```

1-20 Gravitational acceleration g and thus the weight of bodies decreases with increasing elevation. The percent reduction in the weight of an airplane cruising at 13,000 m is to be determined.

Properties The gravitational acceleration g is given to be 9.807 m/s^2 at sea level and 9.767 m/s^2 at an altitude of 13,000 m.

Analysis Weight is proportional to the gravitational acceleration g , and thus the percent reduction in weight is equivalent to the percent reduction in the gravitational acceleration, which is determined from

$$\% \text{Reduction in weight} = \% \text{Reduction in } g = \frac{\Delta g}{g} \times 100 = \frac{9.807 - 9.767}{9.807} \times 100 = \mathbf{0.41\%}$$

Therefore, the airplane and the people in it will weight 0.41% less at 13,000 m altitude.

Discussion Note that the weight loss at cruising altitudes is negligible.



Modeling and Solving Engineering problems

1-21C The *rating* problems deal with the determination of the *heat transfer rate* for an existing system at a specified temperature difference. The *sizing* problems deal with the determination of the *size* of a system in order to transfer heat at a *specified rate* for a *specified temperature difference*.

1-22C The experimental approach (testing and taking measurements) has the advantage of dealing with the actual physical system, and getting a physical value within the limits of experimental error. However, this approach is expensive, time consuming, and often impractical. The analytical approach (analysis or calculations) has the advantage that it is fast and inexpensive, but the results obtained are subject to the accuracy of the assumptions and idealizations made in the analysis.

1-23C Modeling makes it possible to predict the course of an event before it actually occurs, or to study various aspects of an event mathematically without actually running expensive and time-consuming experiments. When preparing a mathematical model, all the variables that affect the phenomena are identified, reasonable assumptions and approximations are made, and the interdependence of these variables are studied. The relevant physical laws and principles are invoked, and the problem is formulated mathematically. Finally, the problem is solved using an appropriate approach, and the results are interpreted.

1-24C The right choice between a crude and complex model is usually the *simplest* model which yields *adequate* results. Preparing very accurate but complex models is not necessarily a better choice since such models are not much use to an analyst if they are very difficult and time consuming to solve. At the minimum, the model should reflect the essential features of the physical problem it represents.

Solving Engineering Problems and EES

1-25C Despite the convenience and capability the engineering software packages offer, they are still just tools, and they will not replace the traditional engineering courses. They will simply cause a shift in emphasis in the course material from mathematics to physics. They are of great value in engineering practice, however, as engineers today rely on software packages for solving large and complex problems in a short time, and perform optimization studies efficiently.

1-26 EES Determine a positive real root of the following equation using EES:

$$2x^3 - 10x^{0.5} - 3x = -3$$

Solution by EES Software (Copy the following line and paste on a blank EES screen to verify solution):

$$2*x^3-10*x^{0.5}-3*x = -3$$

Answer: $x = 2.063$ (using an initial guess of $x=2$)

1-27 EES Solve the following system of 2 equations with 2 unknowns using EES:

$$x^3 - y^2 = 7.75$$

$$3xy + y = 3.5$$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

$$x^3 - y^2 = 7.75$$

$$3*x*y + y = 3.5$$

Answer $x=2$ $y=0.5$

1-28 EES Solve the following system of 3 equations with 3 unknowns using EES:

$$2x - y + z = 5$$

$$3x^2 + 2y = z + 2$$

$$xy + 2z = 8$$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

$$2*x - y + z = 5$$

$$3*x^2 + 2*y = z + 2$$

$$x*y + 2*z = 8$$

Answer $x=1.141$, $y=0.8159$, $z=3.535$

1-29 EES Solve the following system of 3 equations with 3 unknowns using EES:

$$x^2y - z = 1$$

$$x - 3y^{0.5} + xz = -2$$

$$x + y - z = 2$$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

$$x^2*y - z = 1$$

$$x - 3*y^{0.5} + x*z = -2$$

$$x + y - z = 2$$

Answer $x=1$, $y=1$, $z=0$

1-30E EES Specific heat of water is to be expressed at various units using unit conversion capability of EES.

Analysis The problem is solved using EES, and the solution is given below.

EQUATION WINDOW

"GIVEN"

$$C_p = 4.18 \text{ [kJ/kg-C]}$$

"ANALYSIS"

$$C_{p,1} = C_p \cdot \text{Convert}(\text{kJ/kg-C}, \text{kJ/kg-K})$$

$$C_{p,2} = C_p \cdot \text{Convert}(\text{kJ/kg-C}, \text{Btu/lbm-F})$$

$$C_{p,3} = C_p \cdot \text{Convert}(\text{kJ/kg-C}, \text{Btu/lbm-R})$$

$$C_{p,4} = C_p \cdot \text{Convert}(\text{kJ/kg-C}, \text{kCal/kg-C})$$

FORMATTED EQUATIONS WINDOW

GIVEN

$$C_p = 4.18 \text{ [kJ/kg-C]}$$

ANALYSIS

$$C_{p,1} = C_p \cdot \left| 1 \cdot \frac{\text{kJ/kg-K}}{\text{kJ/kg-C}} \right|$$

$$C_{p,2} = C_p \cdot \left| 0.238846 \cdot \frac{\text{Btu/lbm-F}}{\text{kJ/kg-C}} \right|$$

$$C_{p,3} = C_p \cdot \left| 0.238846 \cdot \frac{\text{Btu/lbm-R}}{\text{kJ/kg-C}} \right|$$

$$C_{p,4} = C_p \cdot \left| 0.238846 \cdot \frac{\text{kCal/kg-C}}{\text{kJ/kg-C}} \right|$$

SOLUTION WINDOW

$$C_p = 4.18 \text{ [kJ/kg-C]}$$

$$C_{p,1} = 4.18 \text{ [kJ/kg-K]}$$

$$C_{p,2} = 0.9984 \text{ [Btu/lbm-F]}$$

$$C_{p,3} = 0.9984 \text{ [Btu/lbm-R]}$$

$$C_{p,4} = 0.9984 \text{ [kCal/kg-C]}$$

Review Problems

1-31 The weight of a lunar exploration module on the moon is to be determined.

Analysis Applying Newton's second law, the weight of the module on the moon can be determined from

$$W_{\text{moon}} = mg_{\text{moon}} = \frac{W_{\text{earth}}}{g_{\text{earth}}} g_{\text{moon}} = \frac{4000 \text{ N}}{9.8 \text{ m/s}^2} (1.64 \text{ m/s}^2) = \mathbf{669 \text{ N}}$$

1-32 The gravitational acceleration changes with altitude. Accounting for this variation, the weights of a body at different locations are to be determined.

Analysis The weight of an 80-kg man at various locations is obtained by substituting the altitude z (values in m) into the relation

$$W = mg = (80 \text{ kg})(9.807 - 3.32 \times 10^{-6} z \text{ m/s}^2) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right)$$

$$\text{Sea level:} \quad (z = 0 \text{ m}): W = 80 \times (9.807 - 3.32 \times 10^{-6} \times 0) = 80 \times 9.807 = \mathbf{784.6 \text{ N}}$$

$$\text{Denver:} \quad (z = 1610 \text{ m}): W = 80 \times (9.807 - 3.32 \times 10^{-6} \times 1610) = 80 \times 9.802 = \mathbf{784.2 \text{ N}}$$

$$\text{Mt. Ev.:} \quad (z = 8848 \text{ m}): W = 80 \times (9.807 - 3.32 \times 10^{-6} \times 8848) = 80 \times 9.778 = \mathbf{782.2 \text{ N}}$$

1-33E A man is considering buying a 12-oz steak for \$3.15, or a 320-g steak for \$2.80. The steak that is a better buy is to be determined.

Assumptions The steaks are of identical quality.

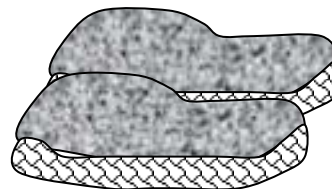
Analysis To make a comparison possible, we need to express the cost of each steak on a common basis. Let us choose 1 kg as the basis for comparison. Using proper conversion factors, the unit cost of each steak is determined to be

12 ounce steak:

$$\text{Unit Cost} = \left(\frac{\$3.15}{12 \text{ oz}} \right) \left(\frac{16 \text{ oz}}{1 \text{ lbm}} \right) \left(\frac{1 \text{ lbm}}{0.45359 \text{ kg}} \right) = \mathbf{\$9.26/\text{kg}}$$

320 gram steak:

$$\text{Unit Cost} = \left(\frac{\$2.80}{320 \text{ g}} \right) \left(\frac{1000 \text{ g}}{1 \text{ kg}} \right) = \mathbf{\$8.75/\text{kg}}$$



Therefore, the steak at the international market is a better buy.

1-34E The thrust developed by the jet engine of a Boeing 777 is given to be 85,000 pounds. This thrust is to be expressed in N and kgf.

Analysis Noting that 1 lbf = 4.448 N and 1 kgf = 9.81 N, the thrust developed can be expressed in two other units as

$$\text{Thrust in N:} \quad \text{Thrust} = (85,000 \text{ lbf}) \left(\frac{4.448 \text{ N}}{1 \text{ lbf}} \right) = \mathbf{3.78 \times 10^5 \text{ N}}$$

$$\text{Thrust in kgf:} \quad \text{Thrust} = (37.8 \times 10^5 \text{ N}) \left(\frac{1 \text{ kgf}}{9.81 \text{ N}} \right) = \mathbf{3.85 \times 10^4 \text{ kgf}}$$



1-35 Design and Essay Problems



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Chapter 2
INTRODUCTION AND BASIC
CONCEPTS

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Systems, Properties, State, and Processes

2-1C This system is a region of space or open system in that mass such as air and food can cross its control boundary. The system can also interact with the surroundings by exchanging heat and work across its control boundary. By tracking these interactions, we can determine the energy conversion characteristics of this system.

2-2C The system is taken as the air contained in the piston-cylinder device. This system is a closed or fixed mass system since no mass enters or leaves it.

2-3C Carbon dioxide is generated by the combustion of fuel in the engine. Any system selected for this analysis must include the fuel and air while it is undergoing combustion. The volume that contains this air-fuel mixture within piston-cylinder device can be used for this purpose. One can also place the entire engine in a control boundary and trace the system-surroundings interactions to determine the rate at which the engine generates carbon dioxide.

2-4C When analyzing the control volume selected, we must account for all forms of water entering and leaving the control volume. This includes all streams entering or leaving the lake, any rain falling on the lake, any water evaporated to the air above the lake, any seepage to the underground earth, and any springs that may be feeding water to the lake.

2-5C Intensive properties do not depend on the size (extent) of the system but extensive properties do.

2-6C The original specific weight is

$$\gamma_1 = \frac{W}{V}$$

If we were to divide the system into two halves, each half weighs $W/2$ and occupies a volume of $V/2$. The specific weight of one of these halves is

$$\gamma = \frac{W/2}{V/2} = \gamma_1$$

which is the same as the original specific weight. Hence, specific weight is an *intensive property*.

2-7C The number of moles of a substance in a system is directly proportional to the number of atomic particles contained in the system. If we divide a system into smaller portions, each portion will contain fewer atomic particles than the original system. The number of moles is therefore an *extensive property*.

2-8C For a system to be in thermodynamic equilibrium, the temperature has to be the same throughout but the pressure does not. However, there should be no unbalanced pressure forces present. The increasing pressure with depth in a fluid, for example, should be balanced by increasing weight.

2-9C A process during which a system remains almost in equilibrium at all times is called a quasi-equilibrium process. Many engineering processes can be approximated as being quasi-equilibrium. The work output of a device is maximum and the work input to a device is minimum when quasi-equilibrium processes are used instead of nonquasi-equilibrium processes.

2-10C A process during which the temperature remains constant is called isothermal; a process during which the pressure remains constant is called isobaric; and a process during which the volume remains constant is called isochoric.

2-11C The state of a simple compressible system is completely specified by two independent, intensive properties.

2-12C In order to describe the state of the air, we need to know the value of all its properties. Pressure, temperature, and water content (i.e., relative humidity or dew point temperature) are commonly cited by weather forecasters. But, other properties like wind speed and chemical composition (i.e., pollen count and smog index, for example) are also important under certain circumstances.

Assuming that the air composition and velocity do not change and that no pressure front motion occurs during the day, the warming process is one of constant pressure (i.e., isobaric).

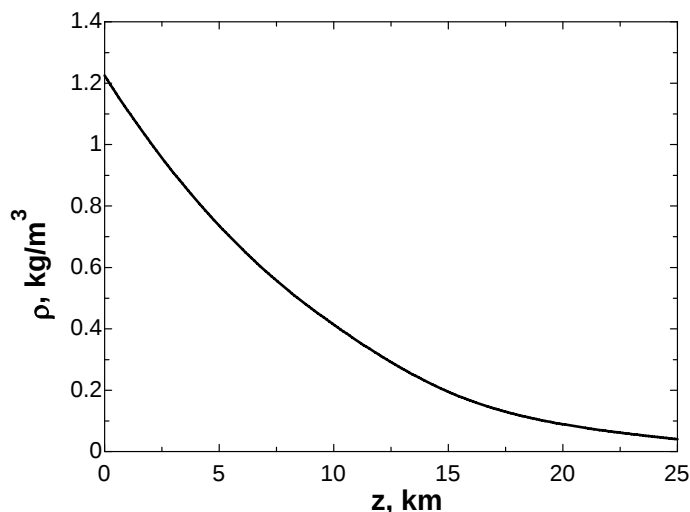
2-13C A process is said to be steady-flow if it involves no changes with time anywhere within the system or at the system boundaries.

2-14 EES The variation of density of atmospheric air with elevation is given in tabular form. A relation for the variation of density with elevation is to be obtained, the density at 7 km elevation is to be calculated, and the mass of the atmosphere using the correlation is to be estimated.

Assumptions 1 Atmospheric air behaves as an ideal gas. 2 The earth is perfectly sphere with a radius of 6377 km, and the thickness of the atmosphere is 25 km.

Properties The density data are given in tabular form as

r , km	z , km	ρ , kg/m ³
6377	0	1.225
6378	1	1.112
6379	2	1.007
6380	3	0.9093
6381	4	0.8194
6382	5	0.7364
6383	6	0.6601
6385	8	0.5258
6387	10	0.4135
6392	15	0.1948
6397	20	0.08891
6402	25	0.04008



Analysis Using EES, (1) Define a trivial function $\rho = a + bz + cz^2$ in equation window, (2) select new parametric table from Tables, and type the data in a two-column table, (3) select Plot and plot the data, and (4) select plot and click on “curve fit” to get curve fit window. Then specify 2nd order polynomial and enter/edit equation. The results are:

$$\rho(z) = a + bz + cz^2 = 1.20252 - 0.101674z + 0.0022375z^2 \quad \text{for the unit of kg/m}^3,$$

$$(\text{or, } \rho(z) = (1.20252 - 0.101674z + 0.0022375z^2) \times 10^9 \quad \text{for the unit of kg/km}^3)$$

where z is the vertical distance from the earth surface at sea level. At $z = 7$ km, the equation would give $\rho = 0.60 \text{ kg/m}^3$.

(b) The mass of atmosphere can be evaluated by integration to be

$$m = \int_V \rho dV = \int_{z=0}^h (a + bz + cz^2) 4\pi(r_0 + z)^2 dz = 4\pi \int_{z=0}^h (a + bz + cz^2)(r_0^2 + 2r_0z + z^2) dz$$

$$= 4\pi \left[ar_0^2 h + r_0(2a + br_0)h^2 / 2 + (a + 2br_0 + cr_0^2)h^3 / 3 + (b + 2cr_0)h^4 / 4 + ch^5 / 5 \right]$$

where $r_0 = 6377$ km is the radius of the earth, $h = 25$ km is the thickness of the atmosphere, and $a = 1.20252$, $b = -0.101674$, and $c = 0.0022375$ are the constants in the density function. Substituting and multiplying by the factor 10^9 for the density unit kg/km^3 , the mass of the atmosphere is determined to be

$$m = 5.092 \times 10^{18} \text{ kg}$$

Discussion Performing the analysis with excel would yield exactly the same results.

EES Solution for final result:

```

a=1.2025166;          b=-0.10167
c=0.0022375;          r=6377;      h=25
m=4*pi*(a*r^2*h+r*(2*a+b*r)*h^2/2+(a+2*b*r+c*r^2)*h^3/3+(b+2*c*r)*h^4/4+c*h^5/5)*1E+9

```

Temperature

2-15C The zeroth law of thermodynamics states that two bodies are in thermal equilibrium if both have the same temperature reading, even if they are not in contact.

2-16C They are Celsius ($^{\circ}\text{C}$) and kelvin (K) in the SI, and fahrenheit ($^{\circ}\text{F}$) and rankine (R) in the English system.

2-17C Probably, but not necessarily. The operation of these two thermometers is based on the thermal expansion of a fluid. If the thermal expansion coefficients of both fluids vary linearly with temperature, then both fluids will expand at the same rate with temperature, and both thermometers will always give identical readings. Otherwise, the two readings may deviate.

2-18 A temperature is given in $^{\circ}\text{C}$. It is to be expressed in K.

Analysis The Kelvin scale is related to Celsius scale by

$$T(\text{K}) = T(^{\circ}\text{C}) + 273$$

Thus, $T(\text{K}) = 37^{\circ}\text{C} + 273 = \mathbf{310\text{ K}}$

2-19E A temperature is given in $^{\circ}\text{C}$. It is to be expressed in $^{\circ}\text{F}$, K, and R.

Analysis Using the conversion relations between the various temperature scales,

$$T(\text{K}) = T(^{\circ}\text{C}) + 273 = 18^{\circ}\text{C} + 273 = \mathbf{291\text{ K}}$$

$$T(^{\circ}\text{F}) = 1.8T(^{\circ}\text{C}) + 32 = (1.8)(18) + 32 = \mathbf{64.4^{\circ}\text{F}}$$

$$T(\text{R}) = T(^{\circ}\text{F}) + 460 = 64.4 + 460 = \mathbf{524.4\text{ R}}$$

2-20 A temperature change is given in $^{\circ}\text{C}$. It is to be expressed in K.

Analysis This problem deals with temperature changes, which are identical in Kelvin and Celsius scales.

Thus, $\Delta T(\text{K}) = \Delta T(^{\circ}\text{C}) = \mathbf{15\text{ K}}$

2-21E The temperature of steam given in K unit is to be converted to °F unit.

Analysis Using the conversion relations between the various temperature scales,

$$T(^{\circ}\text{C}) = T(\text{K}) - 273 = 300 - 273 = 27^{\circ}\text{C}$$

$$T(^{\circ}\text{F}) = 1.8T(^{\circ}\text{C}) + 32 = (1.8)(27) + 32 = \mathbf{80.6^{\circ}\text{F}}$$

2-22E The temperature of oil given in °F unit is to be converted to °C unit.

Analysis Using the conversion relation between the temperature scales,

$$T(^{\circ}\text{C}) = \frac{T(^{\circ}\text{F}) - 32}{1.8} = \frac{150 - 32}{1.8} = \mathbf{65.6^{\circ}\text{C}}$$

2-23E The temperature of air given in °C unit is to be converted to °F unit.

Analysis Using the conversion relation between the temperature scales,

$$T(^{\circ}\text{F}) = 1.8T(^{\circ}\text{C}) + 32 = (1.8)(150) + 32 = \mathbf{302^{\circ}\text{F}}$$

2-24E A temperature range given in °F unit is to be converted to °C unit and the temperature difference in °F is to be expressed in K, °C, and R.

Analysis The lower and upper limits of comfort range in °C are

$$T(^{\circ}\text{C}) = \frac{T(^{\circ}\text{F}) - 32}{1.8} = \frac{65 - 32}{1.8} = \mathbf{18.3^{\circ}\text{C}}$$

$$T(^{\circ}\text{C}) = \frac{T(^{\circ}\text{F}) - 32}{1.8} = \frac{75 - 32}{1.8} = \mathbf{23.9^{\circ}\text{C}}$$

A temperature change of 10°F in various units are

$$\Delta T(\text{R}) = \Delta T(^{\circ}\text{F}) = \mathbf{10 \text{ R}}$$

$$\Delta T(^{\circ}\text{C}) = \frac{\Delta T(^{\circ}\text{F})}{1.8} = \frac{10}{1.8} = \mathbf{5.6^{\circ}\text{C}}$$

$$\Delta T(\text{K}) = \Delta T(^{\circ}\text{C}) = \mathbf{5.6 \text{ K}}$$

Pressure, Manometer, and Barometer

2-25C The pressure relative to the atmospheric pressure is called the *gage pressure*, and the pressure relative to an absolute vacuum is called *absolute pressure*.

2-26C The blood vessels are more restricted when the arm is parallel to the body than when the arm is perpendicular to the body. For a constant volume of blood to be discharged by the heart, the blood pressure must increase to overcome the increased resistance to flow.

2-27C No, the absolute pressure in a liquid of constant density does not double when the depth is doubled. It is the *gage pressure* that doubles when the depth is doubled.

2-28C If the lengths of the sides of the tiny cube suspended in water by a string are very small, the magnitudes of the pressures on all sides of the cube will be the same.

2-29C *Pascal's principle* states that *the pressure applied to a confined fluid increases the pressure throughout by the same amount*. This is a consequence of the pressure in a fluid remaining constant in the horizontal direction. An example of Pascal's principle is the operation of the hydraulic car jack.

2-30E The maximum pressure of a tire is given in English units. It is to be converted to SI units.

Assumptions The listed pressure is gage pressure.

Analysis Noting that $1 \text{ atm} = 101.3 \text{ kPa} = 14.7 \text{ psi}$, the listed maximum pressure can be expressed in SI units as

$$P_{\max} = 35 \text{ psi} = (35 \text{ psi}) \left(\frac{101.3 \text{ kPa}}{14.7 \text{ psi}} \right) = \mathbf{241 \text{ kPa}}$$

Discussion We could also solve this problem by using the conversion factor $1 \text{ psi} = 6.895 \text{ kPa}$.

2-31 The pressure in a tank is given. The tank's pressure in various units are to be determined.

Analysis Using appropriate conversion factors, we obtain

$$(a) \quad P = (1500 \text{ kPa}) \left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}} \right) = \mathbf{1500 \text{ kN/m}^2}$$

$$(b) \quad P = (1500 \text{ kPa}) \left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}} \right) \left(\frac{1000 \text{ kg} \cdot \text{m/s}^2}{1 \text{ kN}} \right) = \mathbf{1,500,000 \text{ kg/m} \cdot \text{s}^2}$$

$$(c) \quad P = (1500 \text{ kPa}) \left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}} \right) \left(\frac{1000 \text{ kg} \cdot \text{m/s}^2}{1 \text{ kN}} \right) \left(\frac{1000 \text{ m}}{1 \text{ km}} \right) = \mathbf{1,500,000,000 \text{ kg/km} \cdot \text{s}^2}$$

2-32E The pressure given in kPa unit is to be converted to psia.

Analysis Using the kPa to psia units conversion factor,

$$P = (200 \text{ kPa}) \left(\frac{1 \text{ psia}}{6.895 \text{ kPa}} \right) = \mathbf{29.0 \text{ psia}}$$

2-33E A manometer measures a pressure difference as inches of water. This is to be expressed in psia unit.

Properties The density of water is taken to be 62.4 lbm/ft^3 (Table A-3E).

Analysis Applying the hydrostatic equation,

$$\begin{aligned} \Delta P &= \rho g h \\ &= (62.4 \text{ lbm/ft}^3)(32.174 \text{ ft/s}^2)(40/12 \text{ ft}) \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm} \cdot \text{ft/s}^2} \right) \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2} \right) \\ &= 1.44 \text{ lbf/in}^2 = \mathbf{1.44 \text{ psia}} \end{aligned}$$

2-34 The pressure given in mm Hg unit is to be converted to kPa.

Analysis Using the mm Hg to kPa units conversion factor,

$$P = (1000 \text{ mm Hg}) \left(\frac{0.1333 \text{ kPa}}{1 \text{ mm Hg}} \right) = \mathbf{133.3 \text{ kPa}}$$

2-35 The pressure in a pressurized water tank is measured by a multi-fluid manometer. The gage pressure of air in the tank is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus we can determine the pressure at the air-water interface.

Properties The densities of mercury, water, and oil are given to be 13,600, 1000, and 850 kg/m³, respectively.

Analysis Starting with the pressure at point 1 at the air-water interface, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach point 2, and setting the result equal to P_{atm} since the tube is open to the atmosphere gives

$$P_1 + \rho_{\text{water}}gh_1 + \rho_{\text{oil}}gh_2 - \rho_{\text{mercury}}gh_3 = P_{\text{atm}}$$

Solving for P_1 ,

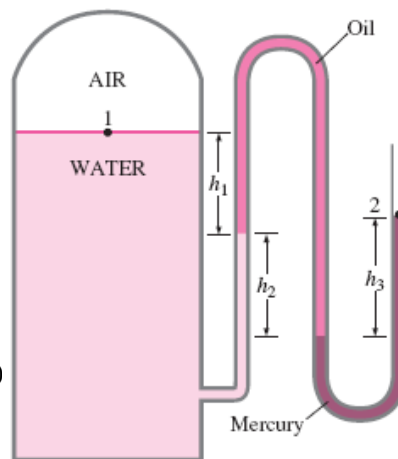
$$P_1 = P_{\text{atm}} - \rho_{\text{water}}gh_1 - \rho_{\text{oil}}gh_2 + \rho_{\text{mercury}}gh_3$$

or,

$$P_1 - P_{\text{atm}} = g(\rho_{\text{mercury}}h_3 - \rho_{\text{water}}h_1 - \rho_{\text{oil}}h_2)$$

Noting that $P_{1,\text{gage}} = P_1 - P_{\text{atm}}$ and substituting,

$$\begin{aligned} P_{1,\text{gage}} &= (9.81 \text{ m/s}^2)[(13,600 \text{ kg/m}^3)(0.46 \text{ m}) - (1000 \text{ kg/m}^3)(0.2 \text{ m}) \\ &\quad - (850 \text{ kg/m}^3)(0.3 \text{ m})] \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{56.9 \text{ kPa}} \end{aligned}$$



Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

2-36 The barometric reading at a location is given in height of mercury column. The atmospheric pressure is to be determined.

Properties The density of mercury is given to be 13,600 kg/m³.

Analysis The atmospheric pressure is determined directly from

$$\begin{aligned} P_{\text{atm}} &= \rho gh \\ &= (13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.750 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{100.1 \text{ kPa}} \end{aligned}$$

2-37 The gage pressure in a liquid at a certain depth is given. The gage pressure in the same liquid at a different depth is to be determined.

Assumptions The variation of the density of the liquid with depth is negligible.

Analysis The gage pressure at two different depths of a liquid can be expressed as

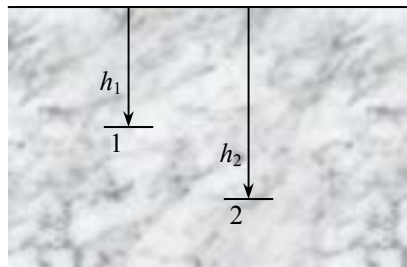
$$P_1 = \rho g h_1 \quad \text{and} \quad P_2 = \rho g h_2$$

Taking their ratio,

$$\frac{P_2}{P_1} = \frac{\rho g h_2}{\rho g h_1} = \frac{h_2}{h_1}$$

Solving for P_2 and substituting gives

$$P_2 = \frac{h_2}{h_1} P_1 = \frac{9 \text{ m}}{3 \text{ m}} (28 \text{ kPa}) = \mathbf{84 \text{ kPa}}$$



Discussion Note that the gage pressure in a given fluid is proportional to depth.

2-38 The absolute pressure in water at a specified depth is given. The local atmospheric pressure and the absolute pressure at the same depth in a different liquid are to be determined.

Assumptions The liquid and water are incompressible.

Properties The specific gravity of the fluid is given to be $SG = 0.85$. We take the density of water to be 1000 kg/m^3 . Then density of the liquid is obtained by multiplying its specific gravity by the density of water,

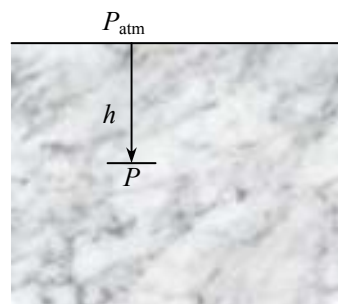
$$\rho = SG \times \rho_{H_2O} = (0.85)(1000 \text{ kg/m}^3) = 850 \text{ kg/m}^3$$

Analysis (a) Knowing the absolute pressure, the atmospheric pressure can be determined from

$$\begin{aligned} P_{\text{atm}} &= P - \rho g h \\ &= (145 \text{ kPa}) - (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(5 \text{ m}) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{96.0 \text{ kPa}} \end{aligned}$$

(b) The absolute pressure at a depth of 5 m in the other liquid is

$$\begin{aligned} P &= P_{\text{atm}} + \rho g h \\ &= (96.0 \text{ kPa}) + (850 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(5 \text{ m}) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{137.7 \text{ kPa}} \end{aligned}$$



Discussion Note that at a given depth, the pressure in the lighter fluid is lower, as expected.

2-39E It is to be shown that $1 \text{ kgf/cm}^2 = 14.223 \text{ psi}$.

Analysis Noting that $1 \text{ kgf} = 9.80665 \text{ N}$, $1 \text{ N} = 0.22481 \text{ lbf}$, and $1 \text{ in} = 2.54 \text{ cm}$, we have

$$1 \text{ kgf} = 9.80665 \text{ N} = (9.80665 \text{ N}) \left(\frac{0.22481 \text{ lbf}}{1 \text{ N}} \right) = 2.20463 \text{ lbf}$$

and

$$1 \text{ kgf/cm}^2 = 2.20463 \text{ lbf/cm}^2 = (2.20463 \text{ lbf/cm}^2) \left(\frac{2.54 \text{ cm}}{1 \text{ in}} \right)^2 = 14.223 \text{ lbf/in}^2 = \mathbf{14.223 \text{ psi}}$$

2-40E The pressure in chamber 3 of the two-piston cylinder shown in the figure is to be determined.

Analysis The area upon which pressure 1 acts is

$$A_1 = \pi \frac{D_1^2}{4} = \pi \frac{(3 \text{ in})^2}{4} = 7.069 \text{ in}^2$$

and the area upon which pressure 2 acts is

$$A_2 = \pi \frac{D_2^2}{4} = \pi \frac{(2 \text{ in})^2}{4} = 3.142 \text{ in}^2$$

The area upon which pressure 3 acts is given by

$$A_3 = A_1 - A_2 = 7.069 - 3.142 = 3.927 \text{ in}^2$$

The force produced by pressure 1 on the piston is then

$$F_1 = P_1 A_1 = (150 \text{ psia}) \left(\frac{1 \text{ lbf/in}^2}{1 \text{ psia}} \right) (7.069 \text{ in}^2) = 1060 \text{ lbf}$$

while that produced by pressure 2 is

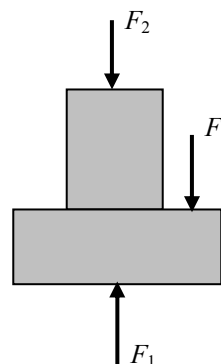
$$F_2 = P_2 A_2 = (200 \text{ psia}) (3.142 \text{ in}^2) = 628 \text{ lbf}$$

According to the vertical force balance on the piston free body diagram

$$F_3 = F_1 - F_2 = 1060 - 628 = 432 \text{ lbf}$$

Pressure 3 is then

$$P_3 = \frac{F_3}{A_3} = \frac{432 \text{ lbf}}{3.927 \text{ in}^2} = \mathbf{110 \text{ psia}}$$



2-41 The pressure in chamber 2 of the two-piston cylinder shown in the figure is to be determined.

Analysis Summing the forces acting on the piston in the vertical direction gives

$$F_2 + F_3 = F_1$$

$$P_2 A_2 + P_3 (A_1 - A_2) = P_1 A_1$$

which when solved for P_2 gives

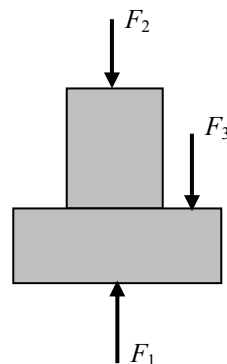
$$P_2 = P_1 \frac{A_1}{A_2} - P_3 \left(\frac{A_1}{A_2} - 1 \right)$$

since the areas of the piston faces are given by $A = \pi D^2 / 4$ the above equation becomes

$$P_2 = P_1 \left(\frac{D_1}{D_2} \right)^2 - P_3 \left[\left(\frac{D_1}{D_2} \right)^2 - 1 \right]$$

$$= (1000 \text{ kPa}) \left(\frac{10}{4} \right)^2 - (500 \text{ kPa}) \left[\left(\frac{10}{4} \right)^2 - 1 \right]$$

$$= \mathbf{3625 \text{ kPa}}$$



2-42 The pressure in chamber 1 of the two-piston cylinder shown in the figure is to be determined.

Analysis Summing the forces acting on the piston in the vertical direction gives

$$F_2 + F_3 = F_1$$

$$P_2 A_2 + P_3 (A_1 - A_2) = P_1 A_1$$

which when solved for P_1 gives

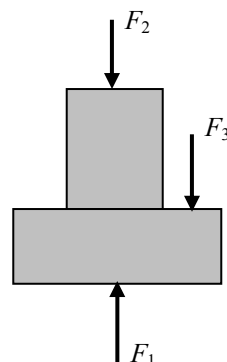
$$P_1 = P_2 \frac{A_2}{A_1} + P_3 \left(1 - \frac{A_2}{A_1} \right)$$

since the areas of the piston faces are given by $A = \pi D^2 / 4$ the above equation becomes

$$P_1 = P_2 \left(\frac{D_2}{D_1} \right)^2 + P_3 \left[1 - \left(\frac{D_2}{D_1} \right)^2 \right]$$

$$= (2000 \text{ kPa}) \left(\frac{4}{10} \right)^2 + (700 \text{ kPa}) \left[1 - \left(\frac{4}{10} \right)^2 \right]$$

$$= \mathbf{908 \text{ kPa}}$$



2-43 The mass of a woman is given. The minimum imprint area per shoe needed to enable her to walk on the snow without sinking is to be determined.

Assumptions 1 The weight of the person is distributed uniformly on the imprint area of the shoes. 2 One foot carries the entire weight of a person during walking, and the shoe is sized for walking conditions (rather than standing). 3 The weight of the shoes is negligible.

Analysis The mass of the woman is given to be 70 kg. For a pressure of 0.5 kPa on the snow, the imprint area of one shoe must be

$$A = \frac{W}{P} = \frac{mg}{P}$$

$$= \frac{(70 \text{ kg})(9.81 \text{ m/s}^2)}{0.5 \text{ kPa}} \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) = \mathbf{1.37 \text{ m}^2}$$



Discussion This is a very large area for a shoe, and such shoes would be impractical to use. Therefore, some sinking of the snow should be allowed to have shoes of reasonable size.

2-44 The vacuum pressure reading of a tank is given. The absolute pressure in the tank is to be determined.

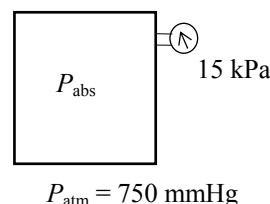
Properties The density of mercury is given to be $\rho = 13,590 \text{ kg/m}^3$.

Analysis The atmospheric (or barometric) pressure can be expressed as

$$P_{\text{atm}} = \rho gh$$

$$= (13,590 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.750 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right)$$

$$= 100.0 \text{ kPa}$$



Then the absolute pressure in the tank becomes

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} = 100.0 - 15 = \mathbf{85.0 \text{ kPa}}$$

2-45E A pressure gage connected to a tank reads 50 psi. The absolute pressure in the tank is to be determined.

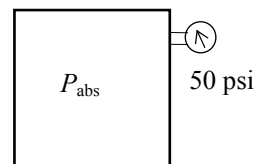
Properties The density of mercury is given to be $\rho = 848.4 \text{ lbf/ft}^3$.

Analysis The atmospheric (or barometric) pressure can be expressed as

$$P_{\text{atm}} = \rho gh$$

$$= (848.4 \text{ lbf/ft}^3)(32.2 \text{ ft/s}^2)(29.1/12 \text{ ft}) \left(\frac{1 \text{ lbf}}{32.2 \text{ lbf} \cdot \text{ft/s}^2} \right) \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2} \right)$$

$$= 14.29 \text{ psia}$$



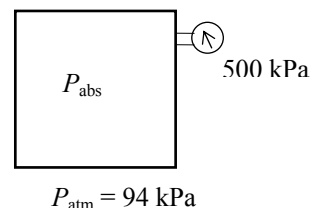
Then the absolute pressure in the tank is

$$P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} = 50 + 14.29 = \mathbf{64.3 \text{ psia}}$$

2-46 A pressure gage connected to a tank reads 500 kPa. The absolute pressure in the tank is to be determined.

Analysis The absolute pressure in the tank is determined from

$$P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} = 500 + 94 = \mathbf{594 \text{ kPa}}$$

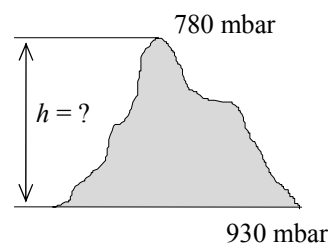


2-47 A mountain hiker records the barometric reading before and after a hiking trip. The vertical distance climbed is to be determined.

Assumptions The variation of air density and the gravitational acceleration with altitude is negligible.

Properties The density of air is given to be $\rho = 1.20 \text{ kg/m}^3$.

Analysis Taking an air column between the top and the bottom of the mountain and writing a force balance per unit base area, we obtain



$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}}$$

$$(\rho gh)_{\text{air}} = P_{\text{bottom}} - P_{\text{top}}$$

$$(1.20 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(h) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ bar}}{100,000 \text{ N/m}^2} \right) = (0.930 - 0.780) \text{ bar}$$

It yields

$$h = \mathbf{1274 \text{ m}}$$

which is also the distance climbed.

2-48 A barometer is used to measure the height of a building by recording reading at the bottom and at the top of the building. The height of the building is to be determined.

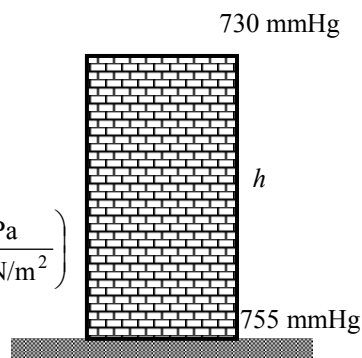
Assumptions The variation of air density with altitude is negligible.

Properties The density of air is given to be $\rho = 1.18 \text{ kg/m}^3$. The density of mercury is $13,600 \text{ kg/m}^3$.

Analysis Atmospheric pressures at the top and at the bottom of the building are

$$\begin{aligned} P_{\text{top}} &= (\rho g h)_{\text{top}} \\ &= (13,600 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.730 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= 97.36 \text{ kPa} \end{aligned}$$

$$\begin{aligned} P_{\text{bottom}} &= (\rho g h)_{\text{bottom}} \\ &= (13,600 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.755 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= 100.70 \text{ kPa} \end{aligned}$$



Taking an air column between the top and the bottom of the building and writing a force balance per unit base area, we obtain

$$\begin{aligned} W_{\text{air}} / A &= P_{\text{bottom}} - P_{\text{top}} \\ (\rho g h)_{\text{air}} &= P_{\text{bottom}} - P_{\text{top}} \\ (1.18 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(h) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) &= (100.70 - 97.36) \text{ kPa} \end{aligned}$$

It yields $h = \mathbf{288.6 \text{ m}}$

which is also the height of the building.

2-49 EES Problem 2-48 is reconsidered. The entire EES solution is to be printed out, including the numerical results with proper units.

Analysis The problem is solved using EES, and the solution is given below.

```
P_bottom=755 [mmHg]
P_top=730 [mmHg]
g=9.807 [m/s^2]    "local acceleration of gravity at sea level"
rho=1.18 [kg/m^3]
DELTAP_abs=(P_bottom-P_top)*CONVERT('mmHg','kPa')"[kPa]"    "Delta P reading from
the barometers, converted from mmHg to kPa."
DELTAP_h=rho*g*h/1000 "[kPa]"    "Delta P due to the air fluid column height, h,
between the top and bottom of the building."
"Instead of dividing by 1000 Pa/kPa we could have multiplied rho*g*h by the EES function,
CONVERT('Pa','kPa')"
```

```
DELTAP_abs=DELTAP_h
```

SOLUTION

Variables in Main

```
DELTAP_abs=3.333 [kPa]
```

```
DELTAP_h=3.333 [kPa]
```

```
g=9.807 [m/s^2]
```

```
h=288 [m]
```

```
P_bottom=755 [mmHg]
```

```
P_top=730 [mmHg]
```

```
rho=1.18 [kg/m^3]
```

2-50 A diver is moving at a specified depth from the water surface. The pressure exerted on the surface of the diver by water is to be determined.

Assumptions The variation of the density of water with depth is negligible.

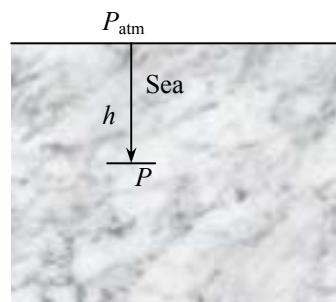
Properties The specific gravity of seawater is given to be $SG = 1.03$. We take the density of water to be 1000 kg/m^3 .

Analysis The density of the seawater is obtained by multiplying its specific gravity by the density of water which is taken to be 1000 kg/m^3 :

$$\rho = SG \times \rho_{H_2O} = (1.03)(1000 \text{ kg/m}^3) = 1030 \text{ kg/m}^3$$

The pressure exerted on a diver at 30 m below the free surface of the sea is the absolute pressure at that location:

$$\begin{aligned} P &= P_{\text{atm}} + \rho gh \\ &= (101 \text{ kPa}) + (1030 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(30 \text{ m}) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{404.0 \text{ kPa}} \end{aligned}$$



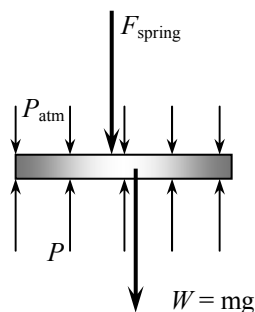
2-51 A gas contained in a vertical piston-cylinder device is pressurized by a spring and by the weight of the piston. The pressure of the gas is to be determined.

Analysis Drawing the free body diagram of the piston and balancing the vertical forces yield

$$PA = P_{\text{atm}}A + W + F_{\text{spring}}$$

Thus,

$$\begin{aligned} P &= P_{\text{atm}} + \frac{mg + F_{\text{spring}}}{A} \\ &= (95 \text{ kPa}) + \frac{(4 \text{ kg})(9.81 \text{ m/s}^2) + 60 \text{ N}}{35 \times 10^{-4} \text{ m}^2} \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{123.4 \text{ kPa}} \end{aligned}$$

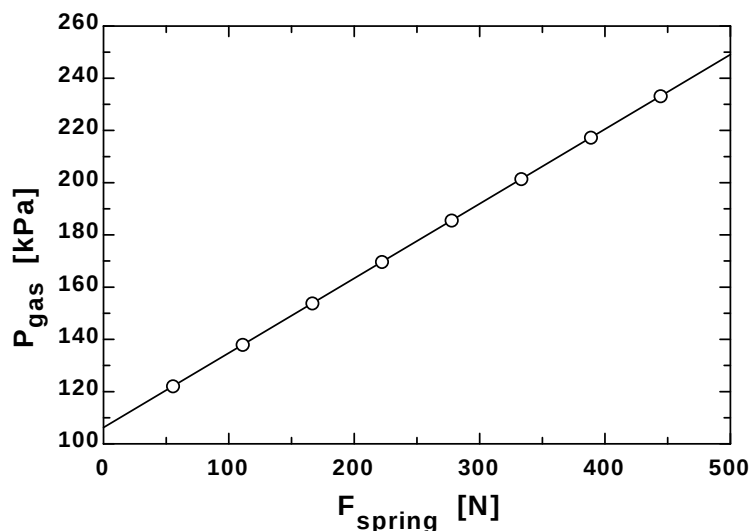


2-52 EES Problem 2-51 is reconsidered. The effect of the spring force in the range of 0 to 500 N on the pressure inside the cylinder is to be investigated. The pressure against the spring force is to be plotted, and results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
g=9.807 [m/s^2]
P_atm= 95 [kPa]
m_piston=4 [kg]
{F_spring=60 [N]}
A=35*CONVERT('cm^2','m^2')"[m^2]"
W_piston=m_piston*g"[N]"
F_atm=P_atm*A*CONVERT('kPa','N/m^2')"[N]"
"From the free body diagram of the piston, the balancing vertical forces yield:"
F_gas= F_atm+F_spring+W_piston"[N]"
P_gas=F_gas/A*CONVERT('N/m^2','kPa')"[kPa]"
```

F _{spring} [N]	P _{gas} [kPa]
0	106.2
55.56	122.1
111.1	138
166.7	153.8
222.2	169.7
277.8	185.6
333.3	201.4
388.9	217.3
444.4	233.2
500	249.1

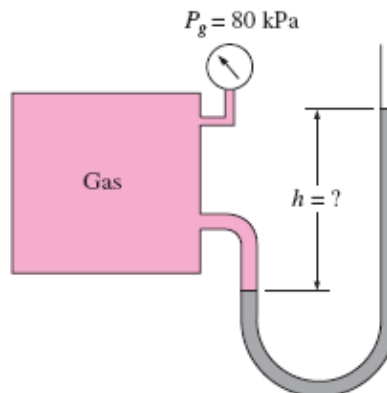


2-53 CD EES Both a gage and a manometer are attached to a gas to measure its pressure. For a specified reading of gage pressure, the difference between the fluid levels of the two arms of the manometer is to be determined for mercury and water.

Properties The densities of water and mercury are given to be $\rho_{\text{water}} = 1000 \text{ kg/m}^3$ and be $\rho_{\text{Hg}} = 13,600 \text{ kg/m}^3$.

Analysis The gage pressure is related to the vertical distance h between the two fluid levels by

$$P_{\text{gage}} = \rho g h \longrightarrow h = \frac{P_{\text{gage}}}{\rho g}$$



(a) For mercury,

$$\begin{aligned} h &= \frac{P_{\text{gage}}}{\rho_{\text{Hg}} g} \\ &= \frac{80 \text{ kPa}}{(13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}} \right) \left(\frac{1000 \text{ kg/m} \cdot \text{s}^2}{1 \text{ kN}} \right) = \mathbf{0.60 \text{ m}} \end{aligned}$$

(b) For water,

$$h = \frac{P_{\text{gage}}}{\rho_{\text{H}_2\text{O}} g} = \frac{80 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}} \right) \left(\frac{1000 \text{ kg/m} \cdot \text{s}^2}{1 \text{ kN}} \right) = \mathbf{8.16 \text{ m}}$$

2-54 EES Problem 2-53 is reconsidered. The effect of the manometer fluid density in the range of 800 to 13,000 kg/m³ on the differential fluid height of the manometer is to be investigated. Differential fluid height against the density is to be plotted, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

Function fluid_density(Fluid\$)

 If fluid\$='Mercury' then fluid_density=13600 else fluid_density=1000
end

{Input from the diagram window. If the diagram window is hidden, then all of the input must come from the equations window. Also note that brackets can also denote comments - but these comments do not appear in the formatted equations window.}

{Fluid\$='Mercury'

P_atm = 101.325 "kpa"

DELTAP=80 "kPa Note how DELTAP is displayed on the Formatted Equations Window."}

g=9.807

"m/s2, local acceleration of gravity at sea level"

rho=Fluid_density(Fluid\$) "Get the fluid density, either Hg or H2O, from the function"

"To plot fluid height against density place {} around the above equation. Then set up the parametric table and solve."

DELTAP = RHO*g*h/1000

"Instead of dividing by 1000 Pa/kPa we could have multiplied by the EES function,

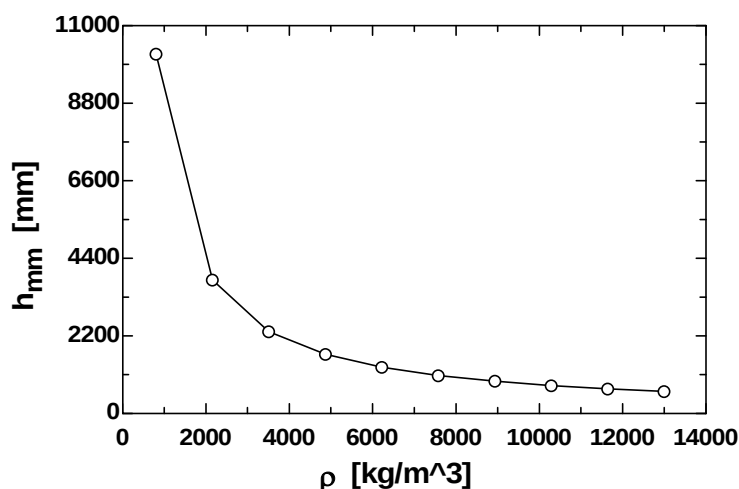
CONVERT('Pa','kPa')"

h_mm=h*convert('m','mm') "The fluid height in mm is found using the built-in CONVERT function."

P_abs= P_atm + DELTAP

h_{mm} [mm]	ρ [kg/m ³]
10197	800
3784	2156
2323	3511
1676	4867
1311	6222
1076	7578
913.1	8933
792.8	10289
700.5	11644
627.5	13000

Manometer Fluid Height vs Manometer Fluid Density

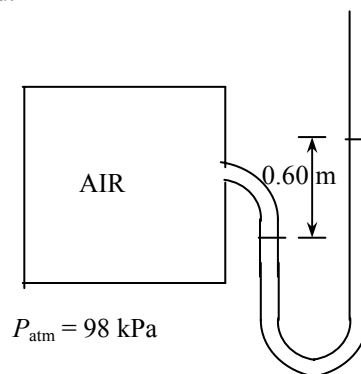


2-55 The air pressure in a tank is measured by an oil manometer. For a given oil-level difference between the two columns, the absolute pressure in the tank is to be determined.

Properties The density of oil is given to be $\rho = 850 \text{ kg/m}^3$.

Analysis The absolute pressure in the tank is determined from

$$\begin{aligned} P &= P_{\text{atm}} + \rho gh \\ &= (98 \text{ kPa}) + (850 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.60 \text{ m}) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{103 \text{ kPa}} \end{aligned}$$



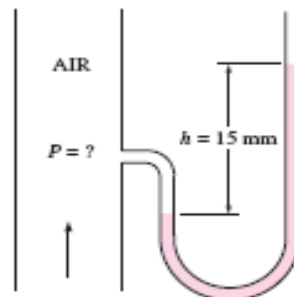
2-56 The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

Properties The density of mercury is given to be $\rho = 13,600 \text{ kg/m}^3$.

Analysis (a) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(b) The absolute pressure in the duct is determined from

$$\begin{aligned} P &= P_{\text{atm}} + \rho gh \\ &= (100 \text{ kPa}) + (13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.015 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{102 \text{ kPa}} \end{aligned}$$



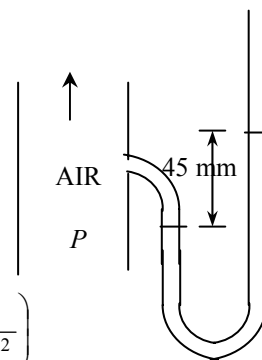
2-57 The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

Properties The density of mercury is given to be $\rho = 13,600 \text{ kg/m}^3$.

Analysis (a) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(b) The absolute pressure in the duct is determined from

$$\begin{aligned} P &= P_{\text{atm}} + \rho gh \\ &= (100 \text{ kPa}) + (13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.045 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{106 \text{ kPa}} \end{aligned}$$



2-58E The systolic and diastolic pressures of a healthy person are given in mmHg. These pressures are to be expressed in kPa, psi, and meter water column.

Assumptions Both mercury and water are incompressible substances.

Properties We take the densities of water and mercury to be 1000 kg/m^3 and $13,600 \text{ kg/m}^3$, respectively.

Analysis Using the relation $P = \rho gh$ for gage pressure, the high and low pressures are expressed as

$$P_{\text{high}} = \rho gh_{\text{high}} = (13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.12 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) = \mathbf{16.0 \text{ kPa}}$$

$$P_{\text{low}} = \rho gh_{\text{low}} = (13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.08 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) = \mathbf{10.7 \text{ kPa}}$$

Noting that $1 \text{ psi} = 6.895 \text{ kPa}$,

$$P_{\text{high}} = (16.0 \text{ Pa}) \left(\frac{1 \text{ psi}}{6.895 \text{ kPa}} \right) = \mathbf{2.32 \text{ psi}} \quad \text{and} \quad P_{\text{low}} = (10.7 \text{ Pa}) \left(\frac{1 \text{ psi}}{6.895 \text{ kPa}} \right) = \mathbf{1.55 \text{ psi}}$$

For a given pressure, the relation $P = \rho gh$ can be expressed for mercury and water as $P = \rho_{\text{water}} gh_{\text{water}}$ and $P = \rho_{\text{mercury}} gh_{\text{mercury}}$.

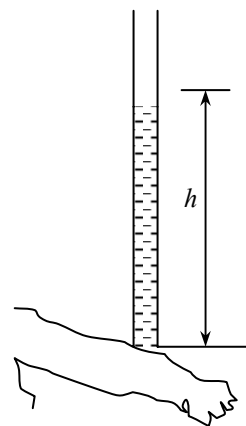
Setting these two relations equal to each other and solving for water height gives

$$P = \rho_{\text{water}} gh_{\text{water}} = \rho_{\text{mercury}} gh_{\text{mercury}} \rightarrow h_{\text{water}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury}}$$

Therefore,

$$h_{\text{water, high}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, high}} = \frac{13,600 \text{ kg/m}^3}{1000 \text{ kg/m}^3} (0.12 \text{ m}) = \mathbf{1.63 \text{ m}}$$

$$h_{\text{water, low}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, low}} = \frac{13,600 \text{ kg/m}^3}{1000 \text{ kg/m}^3} (0.08 \text{ m}) = \mathbf{1.09 \text{ m}}$$



Discussion Note that measuring blood pressure with a “water” monometer would involve differential fluid heights higher than the person, and thus it is impractical. This problem shows why mercury is a suitable fluid for blood pressure measurement devices.

2-59 A vertical tube open to the atmosphere is connected to the vein in the arm of a person. The height that the blood will rise in the tube is to be determined.

Assumptions 1 The density of blood is constant. 2 The gage pressure of blood is 120 mmHg.

Properties The density of blood is given to be $\rho = 1050 \text{ kg/m}^3$.

Analysis For a given gage pressure, the relation $P = \rho gh$ can be expressed for mercury and blood as $P = \rho_{\text{blood}} gh_{\text{blood}}$ and $P = \rho_{\text{mercury}} gh_{\text{mercury}}$.

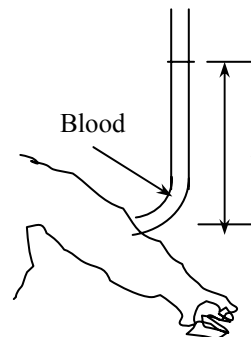
Setting these two relations equal to each other we get

$$P = \rho_{\text{blood}} gh_{\text{blood}} = \rho_{\text{mercury}} gh_{\text{mercury}}$$

Solving for blood height and substituting gives

$$h_{\text{blood}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{blood}}} h_{\text{mercury}} = \frac{13,600 \text{ kg/m}^3}{1050 \text{ kg/m}^3} (0.12 \text{ m}) = \mathbf{1.55 \text{ m}}$$

Discussion Note that the blood can rise about one and a half meters in a tube connected to the vein. This explains why IV tubes must be placed high to force a fluid into the vein of a patient.



2-60 A man is standing in water vertically while being completely submerged. The difference between the pressures acting on the head and on the toes is to be determined.

Assumptions Water is an incompressible substance, and thus the density does not change with depth.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis The pressures at the head and toes of the person can be expressed as

$$P_{\text{head}} = P_{\text{atm}} + \rho gh_{\text{head}} \quad \text{and} \quad P_{\text{toe}} = P_{\text{atm}} + \rho gh_{\text{toe}}$$

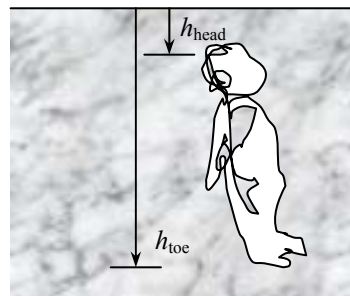
where h is the vertical distance of the location in water from the free surface. The pressure difference between the toes and the head is determined by subtracting the first relation above from the second,

$$P_{\text{toe}} - P_{\text{head}} = \rho gh_{\text{toe}} - \rho gh_{\text{head}} = \rho g(h_{\text{toe}} - h_{\text{head}})$$

Substituting,

$$P_{\text{toe}} - P_{\text{head}} = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(1.80 \text{ m} - 0) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) = \mathbf{17.7 \text{ kPa}}$$

Discussion This problem can also be solved by noting that the atmospheric pressure ($1 \text{ atm} = 101.325 \text{ kPa}$) is equivalent to 10.3-m of water height, and finding the pressure that corresponds to a water height of 1.8 m.

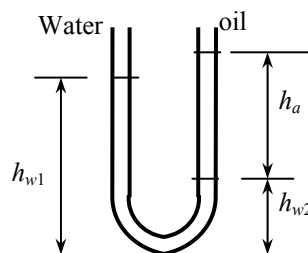


2-61 Water is poured into the U-tube from one arm and oil from the other arm. The water column height in one arm and the ratio of the heights of the two fluids in the other arm are given. The height of each fluid in that arm is to be determined.

Assumptions Both water and oil are incompressible substances.

Properties The density of oil is given to be $\rho = 790 \text{ kg/m}^3$. We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis The height of water column in the left arm of the manometer is given to be $h_{w1} = 0.70 \text{ m}$. We let the height of water and oil in the right arm to be h_{w2} and h_a , respectively. Then, $h_a = 4h_{w2}$. Noting that both arms are open to the atmosphere, the pressure at the bottom of the U-tube can be expressed as



$$P_{\text{bottom}} = P_{\text{atm}} + \rho_w g h_{w1} \quad \text{and} \quad P_{\text{bottom}} = P_{\text{atm}} + \rho_w g h_{w2} + \rho_a g h_a$$

Setting them equal to each other and simplifying,

$$\rho_w g h_{w1} = \rho_w g h_{w2} + \rho_a g h_a \quad \rightarrow \quad \rho_w h_{w1} = \rho_w h_{w2} + \rho_a h_a \quad \rightarrow \quad h_{w1} = h_{w2} + (\rho_a / \rho_w) h_a$$

Noting that $h_a = 4h_{w2}$, the water and oil column heights in the second arm are determined to be

$$0.7 \text{ m} = h_{w2} + (790/1000) 4h_{w2} \quad \rightarrow \quad h_{w2} = \mathbf{0.168 \text{ m}}$$

$$0.7 \text{ m} = 0.168 \text{ m} + (790/1000) h_a \quad \rightarrow \quad h_a = \mathbf{0.673 \text{ m}}$$

Discussion Note that the fluid height in the arm that contains oil is higher. This is expected since oil is lighter than water.

2-62 Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

Assumptions 1 All the liquids are incompressible. **2**

The effect of air column on pressure is negligible.

Properties The densities of seawater and mercury are given to be $\rho_{\text{sea}} = 1035 \text{ kg/m}^3$ and $\rho_{\text{Hg}} = 13,600 \text{ kg/m}^3$. We take the density of water to be $\rho_w = 1000 \text{ kg/m}^3$.

Analysis Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the sea water pipe (point 2), and setting the result equal to P_2 gives

$$P_1 + \rho_w gh_w - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{air}} gh_{\text{air}} + \rho_{\text{sea}} gh_{\text{sea}} = P_2$$

Rearranging and neglecting the effect of air column on pressure,

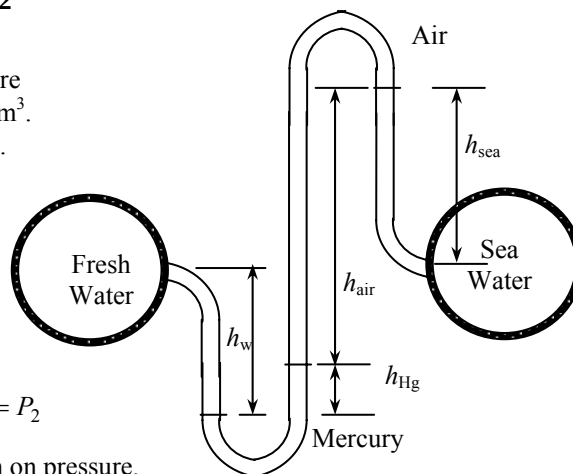
$$P_1 - P_2 = -\rho_w gh_w + \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{sea}} gh_{\text{sea}} = g(\rho_{\text{Hg}} h_{\text{Hg}} - \rho_w h_w - \rho_{\text{sea}} h_{\text{sea}})$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81 \text{ m/s}^2)[(13600 \text{ kg/m}^3)(0.1 \text{ m}) \\ &\quad - (1000 \text{ kg/m}^3)(0.6 \text{ m}) - (1035 \text{ kg/m}^3)(0.4 \text{ m})] \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \\ &= 3.39 \text{ kN/m}^2 = \mathbf{3.39 \text{ kPa}} \end{aligned}$$

Therefore, the pressure in the fresh water pipe is 3.39 kPa higher than the pressure in the sea water pipe.

Discussion A 0.70-m high air column with a density of 1.2 kg/m^3 corresponds to a pressure difference of 0.008 kPa. Therefore, its effect on the pressure difference between the two pipes is negligible.



2-63 Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

Assumptions All the liquids are incompressible.

Properties The densities of seawater and mercury are given to be $\rho_{\text{sea}} = 1035 \text{ kg/m}^3$ and $\rho_{\text{Hg}} = 13,600 \text{ kg/m}^3$. We take the density of water to be $\rho_w = 1000 \text{ kg/m}^3$. The specific gravity of oil is given to be 0.72, and thus its density is 720 kg/m^3 .

Analysis Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the sea water pipe (point 2), and setting the result equal to P_2 gives

$$P_1 + \rho_w gh_w - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{oil}} gh_{\text{oil}} + \rho_{\text{sea}} gh_{\text{sea}} = P_2$$

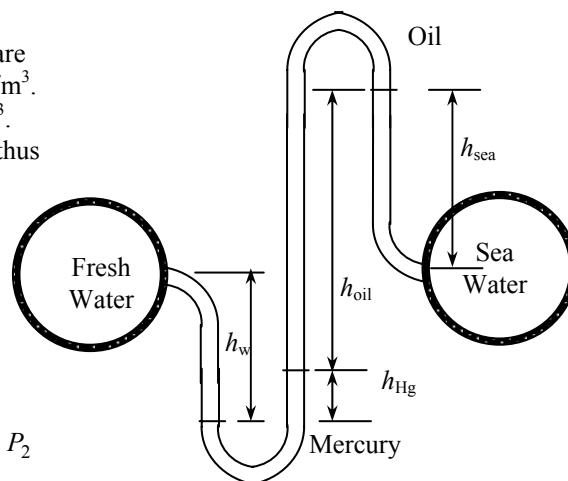
Rearranging,

$$\begin{aligned} P_1 - P_2 &= -\rho_w gh_w + \rho_{\text{Hg}} gh_{\text{Hg}} + \rho_{\text{oil}} gh_{\text{oil}} - \rho_{\text{sea}} gh_{\text{sea}} \\ &= g(\rho_{\text{Hg}} h_{\text{Hg}} + \rho_{\text{oil}} h_{\text{oil}} - \rho_w h_w - \rho_{\text{sea}} h_{\text{sea}}) \end{aligned}$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81 \text{ m/s}^2)[(13600 \text{ kg/m}^3)(0.1 \text{ m}) + (720 \text{ kg/m}^3)(0.7 \text{ m}) - (1000 \text{ kg/m}^3)(0.6 \text{ m}) \\ &\quad - (1035 \text{ kg/m}^3)(0.4 \text{ m})] \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \\ &= 8.34 \text{ kN/m}^2 = \mathbf{8.34 \text{ kPa}} \end{aligned}$$

Therefore, the pressure in the fresh water pipe is 8.34 kPa higher than the pressure in the sea water pipe.



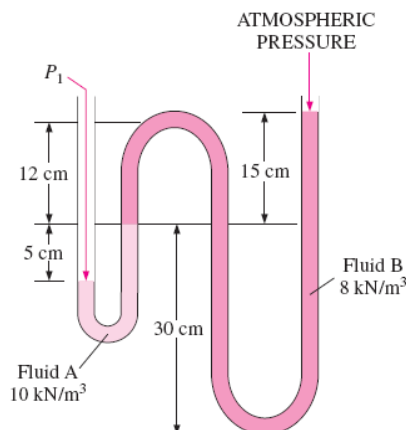
2-64 The pressure indicated by a manometer is to be determined.

Properties The specific weights of fluid A and fluid B are given to be 10 kN/m^3 and 8 kN/m^3 , respectively.

Analysis The absolute pressure P_1 is determined from

$$\begin{aligned} P_1 &= P_{\text{atm}} + (\rho gh)_A + (\rho gh)_B \\ &= P_{\text{atm}} + \gamma_A h_A + \gamma_B h_B \\ &= (758 \text{ mm Hg}) \left(\frac{0.1333 \text{ kPa}}{1 \text{ mm Hg}} \right) \\ &\quad + (10 \text{ kN/m}^3)(0.05 \text{ m}) + (8 \text{ kN/m}^3)(0.15 \text{ m}) \\ &= \mathbf{102.7 \text{ kPa}} \end{aligned}$$

Note that $1 \text{ kPa} = 1 \text{ kN/m}^2$.



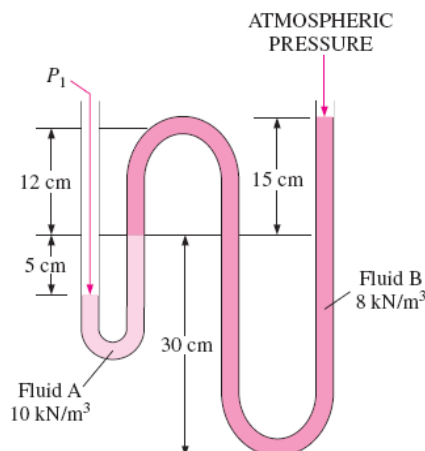
2-65 The pressure indicated by a manometer is to be determined.

Properties The specific weights of fluid A and fluid B are given to be 100 kN/m^3 and 8 kN/m^3 , respectively.

Analysis The absolute pressure P_1 is determined from

$$\begin{aligned} P_1 &= P_{\text{atm}} + (\rho g h)_A + (\rho g h)_B \\ &= P_{\text{atm}} + \gamma_A h_A + \gamma_B h_B \\ &= 90 \text{ kPa} + (100 \text{ kN/m}^3)(0.05 \text{ m}) + (8 \text{ kN/m}^3)(0.15 \text{ m}) \\ &= \mathbf{96.2 \text{ kPa}} \end{aligned}$$

Note that $1 \text{ kPa} = 1 \text{ kN/m}^2$.



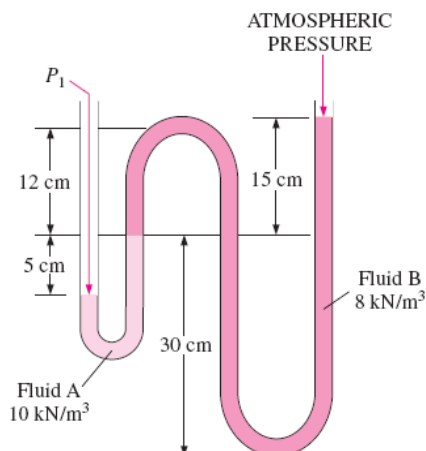
2-66 The pressure indicated by a manometer is to be determined.

Properties The specific weights of fluid A and fluid B are given to be 10 kN/m^3 and 20 kN/m^3 , respectively.

Analysis The absolute pressure P_1 is determined from

$$\begin{aligned} P_1 &= P_{\text{atm}} + (\rho g h)_A + (\rho g h)_B \\ &= P_{\text{atm}} + \gamma_A h_A + \gamma_B h_B \\ &= (745 \text{ mm Hg}) \left(\frac{0.1333 \text{ kPa}}{1 \text{ mm Hg}} \right) \\ &\quad + (10 \text{ kN/m}^3)(0.05 \text{ m}) + (20 \text{ kN/m}^3)(0.15 \text{ m}) \\ &= \mathbf{102.8 \text{ kPa}} \end{aligned}$$

Note that $1 \text{ kPa} = 1 \text{ kN/m}^2$.



2-67 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height h of the mercury column is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

Properties We take the density of water to be $\rho_w = 1000 \text{ kg/m}^3$. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

Analysis Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

$$P_1 + \rho_w gh_w - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{oil}} gh_{\text{oil}} = P_{\text{atm}}$$

Rearranging,

$$P_1 - P_{\text{atm}} = \rho_{\text{oil}} gh_{\text{oil}} + \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_w gh_w$$

or,

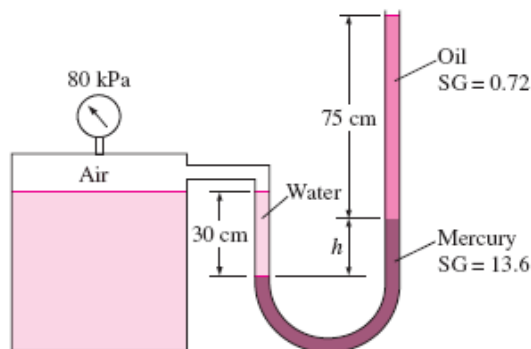
$$\frac{P_{1,\text{gage}}}{\rho_w g} = \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} - h_w$$

Substituting,

$$\left(\frac{80 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \right) \left(\frac{1000 \text{ kg} \cdot \text{m/s}^2}{1 \text{ kPa} \cdot \text{m}^2} \right) = 0.72 \times (0.75 \text{ m}) + 13.6 \times h_{\text{Hg}} - 0.3 \text{ m}$$

Solving for h_{Hg} gives $h_{\text{Hg}} = \mathbf{0.582 \text{ m}}$. Therefore, the differential height of the mercury column must be 58.2 cm.

Discussion Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.



2-68 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height h of the mercury column is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

Properties We take the density of water to be $\rho_w = 1000 \text{ kg/m}^3$. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

Analysis Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

$$P_1 + \rho_w gh_w - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{oil}} gh_{\text{oil}} = P_{\text{atm}}$$

Rearranging,

$$P_1 - P_{\text{atm}} = \rho_{\text{oil}} gh_{\text{oil}} + \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_w gh_w$$

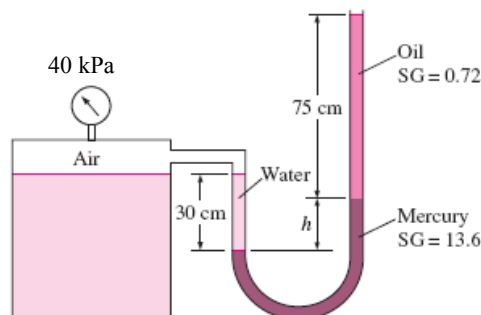
or,
$$\frac{P_{1,\text{gage}}}{\rho_w g} = \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} - h_w$$

Substituting,

$$\left[\frac{40 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \right] \left(\frac{1000 \text{ kg} \cdot \text{m/s}^2}{1 \text{ kPa} \cdot \text{m}^2} \right) = 0.72 \times (0.75 \text{ m}) + 13.6 \times h_{\text{Hg}} - 0.3 \text{ m}$$

Solving for h_{Hg} gives $h_{\text{Hg}} = \mathbf{0.282 \text{ m}}$. Therefore, the differential height of the mercury column must be 28.2 cm.

Discussion Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.



2-69 The top part of a water tank is divided into two compartments, and a fluid with an unknown density is poured into one side. The levels of the water and the liquid are measured. The density of the fluid is to be determined.

Assumptions 1 Both water and the added liquid are incompressible substances. **2** The added liquid does not mix with water.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

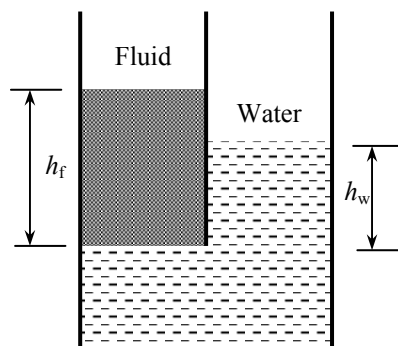
Analysis Both fluids are open to the atmosphere. Noting that the pressure of both water and the added fluid is the same at the contact surface, the pressure at this surface can be expressed as

$$P_{\text{contact}} = P_{\text{atm}} + \rho_f gh_f = P_{\text{atm}} + \rho_w gh_w$$

Simplifying and solving for ρ_f gives

$$\rho_f gh_f = \rho_w gh_w \rightarrow \rho_f = \frac{h_w}{h_f} \rho_w = \frac{45 \text{ cm}}{80 \text{ cm}} (1000 \text{ kg/m}^3) = \mathbf{562.5 \text{ kg/m}^3}$$

Discussion Note that the added fluid is lighter than water as expected (a heavier fluid would sink in water).

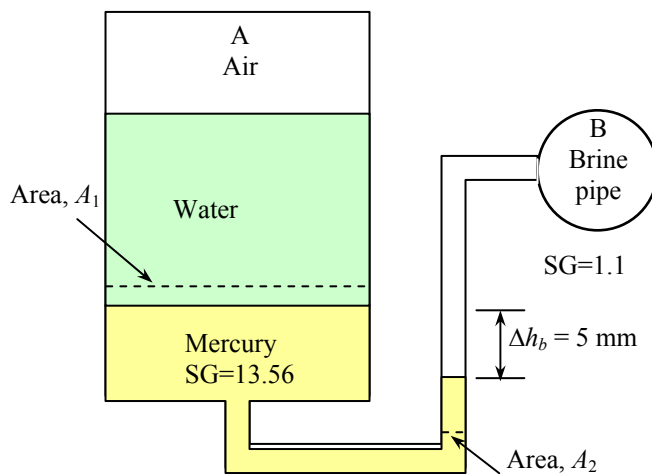


2-70 The fluid levels in a multi-fluid U-tube manometer change as a result of a pressure drop in the trapped air space. For a given pressure drop and brine level change, the area ratio is to be determined.

Assumptions 1 All the liquids are incompressible. **2** Pressure in the brine pipe remains constant. **3** The variation of pressure in the trapped air space is negligible.

Properties The specific gravities are given to be 13.56 for mercury and 1.1 for brine. We take the standard density of water to be $\rho_w = 1000 \text{ kg/m}^3$.

Analysis It is clear from the problem statement and the figure that the brine pressure is much higher than the air pressure, and when the air pressure drops by 0.7 kPa, the pressure difference between the brine and the air space increases also by the same amount.



Starting with the air pressure (point A) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the brine pipe (point B), and setting the result equal to P_B before and after the pressure change of air give

$$\text{Before: } P_{A1} + \rho_w gh_w + \rho_{\text{Hg}} gh_{\text{Hg},1} - \rho_{\text{br}} gh_{\text{br},1} = P_B$$

$$\text{After: } P_{A2} + \rho_w gh_w + \rho_{\text{Hg}} gh_{\text{Hg},2} - \rho_{\text{br}} gh_{\text{br},2} = P_B$$

Subtracting,

$$P_{A2} - P_{A1} + \rho_{\text{Hg}} g \Delta h_{\text{Hg}} - \rho_{\text{br}} g \Delta h_{\text{br}} = 0 \rightarrow \frac{P_{A1} - P_{A2}}{\rho_w g} = SG_{\text{Hg}} \Delta h_{\text{Hg}} - SG_{\text{br}} \Delta h_{\text{br}} = 0 \quad (1)$$

where Δh_{Hg} and Δh_{br} are the changes in the differential mercury and brine column heights, respectively, due to the drop in air pressure. Both of these are positive quantities since as the mercury-brine interface drops, the differential fluid heights for both mercury and brine increase. Noting also that the volume of mercury is constant, we have $A_1 \Delta h_{\text{Hg, left}} = A_2 \Delta h_{\text{Hg, right}}$ and

$$P_{A2} - P_{A1} = -0.7 \text{ kPa} = -700 \text{ N/m}^2 = -700 \text{ kg/m} \cdot \text{s}^2$$

$$\Delta h_{\text{br}} = 0.005 \text{ m}$$

$$\Delta h_{\text{Hg}} = \Delta h_{\text{Hg, right}} + \Delta h_{\text{Hg, left}} = \Delta h_{\text{br}} + \Delta h_{\text{br}} A_2 / A_1 = \Delta h_{\text{br}} (1 + A_2 / A_1)$$

Substituting,

$$\frac{700 \text{ kg/m} \cdot \text{s}^2}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} = [13.56 \times 0.005(1 + A_2 / A_1) - (1.1 \times 0.005)] \text{ m}$$

It gives

$$A_2 / A_1 = \mathbf{0.134}$$

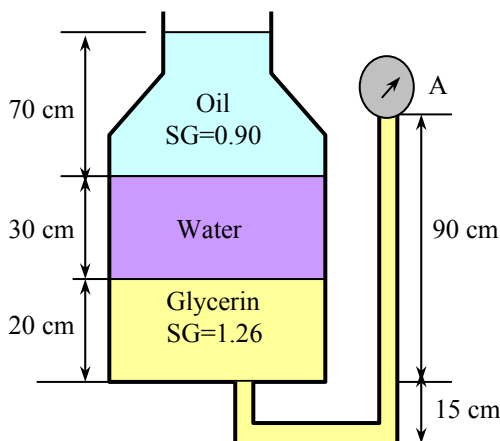
2-71 A multi-fluid container is connected to a U-tube. For the given specific gravities and fluid column heights, the gage pressure at A and the height of a mercury column that would create the same pressure at A are to be determined.

Assumptions 1 All the liquids are incompressible.

2 The multi-fluid container is open to the atmosphere.

Properties The specific gravities are given to be 1.26 for glycerin and 0.90 for oil. We take the standard density of water to be $\rho_w = 1000 \text{ kg/m}^3$, and the specific gravity of mercury to be 13.6.

Analysis Starting with the atmospheric pressure on the top surface of the container and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach point A, and setting the result equal to P_A give



$$P_{\text{atm}} + \rho_{\text{oil}} g h_{\text{oil}} + \rho_w g h_w - \rho_{\text{gly}} g h_{\text{gly}} = P_A$$

Rearranging and using the definition of specific gravity,

$$P_A - P_{\text{atm}} = SG_{\text{oil}} \rho_w g h_{\text{oil}} + SG_w \rho_w g h_w - SG_{\text{gly}} \rho_w g h_{\text{gly}}$$

or

$$P_{A,\text{gage}} = g \rho_w (SG_{\text{oil}} h_{\text{oil}} + SG_w h_w - SG_{\text{gly}} h_{\text{gly}})$$

Substituting,

$$\begin{aligned} P_{A,\text{gage}} &= (9.81 \text{ m/s}^2)(1000 \text{ kg/m}^3)[0.90(0.70 \text{ m}) + 1(0.3 \text{ m}) - 1.26(0.70 \text{ m})] \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \\ &= 0.471 \text{ kN/m}^2 = \mathbf{0.471 \text{ kPa}} \end{aligned}$$

The equivalent mercury column height is

$$h_{\text{Hg}} = \frac{P_{A,\text{gage}}}{\rho_{\text{Hg}} g} = \frac{0.471 \text{ kN/m}^2}{(13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \left(\frac{1000 \text{ kg} \cdot \text{m/s}^2}{1 \text{ kN}} \right) = 0.00353 \text{ m} = \mathbf{0.353 \text{ cm}}$$

Discussion Note that the high density of mercury makes it a very suitable fluid for measuring high pressures in manometers.

Review Problems

2-72 The deflection of the spring of the two-piston cylinder with a spring shown in the figure is to be determined.

Analysis Summing the forces acting on the piston in the vertical direction gives

$$F_s + F_2 + F_3 = F_1$$

$$kx + P_2 A_2 + P_3 (A_1 - A_2) = P_1 A_1$$

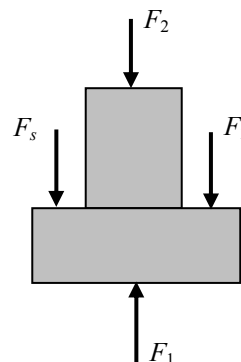
which when solved for the deflection of the spring and substituting $A = \pi D^2 / 4$ gives

$$x = \frac{\pi}{4k} [P_1 D_1^2 - P_2 D_2^2 - P_3 (D_1^2 - D_2^2)]$$

$$= \frac{\pi}{4 \times 800} [5000 \times 0.08^2 - 10,000 \times 0.03^2 - 1000(0.08^2 - 0.03^2)]$$

$$= 0.0172 \text{ m}$$

$$= \mathbf{1.72 \text{ cm}}$$



We expressed the spring constant k in kN/m, the pressures in kPa (i.e., kN/m²) and the diameters in m units.

2-73 The pressure in chamber 1 of the two-piston cylinder with a spring shown in the figure is to be determined.

Analysis Summing the forces acting on the piston in the vertical direction gives

$$F_s + F_1 = F_2 + F_3$$

$$kx + P_1 A_1 = P_2 A_2 + P_3 (A_1 - A_2)$$

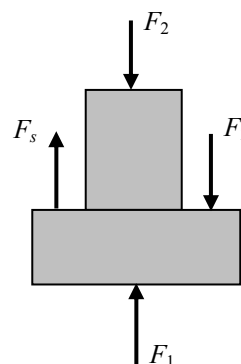
which when solved for the P_3 and substituting $A = \pi D^2 / 4$ gives

$$P_1 = P_2 \frac{A_2}{A_1} + P_3 \left(1 - \frac{A_2}{A_1} \right) - \frac{kx}{A_1}$$

$$= P_2 \left(\frac{D_2}{D_1} \right)^2 + P_3 \left[1 - \left(\frac{D_2}{D_1} \right)^2 \right] - \frac{4kx}{\pi D_1^2}$$

$$= (8000 \text{ kPa}) \left(\frac{3}{7} \right)^2 + (300 \text{ kPa}) \left[1 - \left(\frac{3}{7} \right)^2 \right] - \frac{4(1200 \text{ kN/m})(0.05 \text{ m})}{\pi (0.07 \text{ m})^2}$$

$$= 13,880 \text{ kPa} = \mathbf{13.9 \text{ MPa}}$$



2-74E The pressure in chamber 2 of the two-piston cylinder with a spring shown in the figure is to be determined.

Analysis The areas upon which pressures act are

$$A_1 = \pi \frac{D_1^2}{4} = \pi \frac{(5 \text{ in})^2}{4} = 19.63 \text{ in}^2$$

$$A_2 = \pi \frac{D_2^2}{4} = \pi \frac{(2 \text{ in})^2}{4} = 3.142 \text{ in}^2$$

$$A_3 = A_1 - A_2 = 19.63 - 3.142 = 16.49 \text{ in}^2$$

The forces generated by pressure 1 and 3 are

$$F_1 = P_1 A_1 = (100 \text{ psia}) \left(\frac{1 \text{ lbf/in}^2}{1 \text{ psia}} \right) (19.63 \text{ in}^2) = 1963 \text{ lbf}$$

$$F_3 = P_2 A_2 = (20 \text{ psia}) (16.49 \text{ in}^2) = 330 \text{ lbf}$$

The force exerted by the spring is

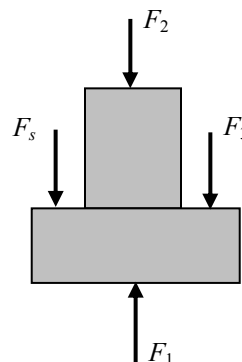
$$F_s = kx = (200 \text{ lbf/in})(2 \text{ in}) = 400 \text{ lbf}$$

Summing the vertical forces acting on the piston gives

$$F_2 = F_1 - F_3 - F_s = 1963 - 330 - 400 = 1233 \text{ lbf}$$

The pressure at 2 is then

$$P_2 = \frac{F_2}{A_2} = \frac{1233 \text{ lbf}}{3.142 \text{ in}^2} = \mathbf{392 \text{ psia}}$$



2-75 An airplane is flying over a city. The local atmospheric pressure in that city is to be determined.

Assumptions The gravitational acceleration does not change with altitude.

Properties The densities of air and mercury are given to be 1.15 kg/m^3 and $13,600 \text{ kg/m}^3$.

Analysis The local atmospheric pressure is determined from

$$\begin{aligned} P_{\text{atm}} &= P_{\text{plane}} + \rho g h \\ &= 58 \text{ kPa} + (1.15 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(3000 \text{ m}) \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) = 91.84 \text{ kN/m}^2 = \mathbf{91.8 \text{ kPa}} \end{aligned}$$

The atmospheric pressure may be expressed in mmHg as

$$h_{\text{Hg}} = \frac{P_{\text{atm}}}{\rho g} = \frac{91.8 \text{ kPa}}{(13,600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \left(\frac{1000 \text{ Pa}}{1 \text{ kPa}} \right) \left(\frac{1000 \text{ mm}}{1 \text{ m}} \right) = \mathbf{688 \text{ mmHg}}$$

2-76E The efficiency of a refrigerator increases by 3% per °C rise in the minimum temperature. This increase is to be expressed per °F, K, and R rise in the minimum temperature.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the increase in efficiency is

- (a) **3%** for each K rise in temperature, and
- (b), (c) $3/1.8 = \mathbf{1.67\%}$ for each R or °F rise in temperature.

2-77E The boiling temperature of water decreases by 3°C for each 1000 m rise in altitude. This decrease in temperature is to be expressed in °F, K, and R.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the decrease in the boiling temperature is

- (a) **3 K** for each 1000 m rise in altitude, and
- (b), (c) $3 \times 1.8 = \mathbf{5.4^\circ F} = \mathbf{5.4 R}$ for each 1000 m rise in altitude.

2-78E Hyperthermia of 5°C is considered fatal. This fatal level temperature change of body temperature is to be expressed in °F, K, and R.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the fatal level of hypothermia is

- (a) **5 K**
- (b) $5 \times 1.8 = \mathbf{9^\circ F}$
- (c) $5 \times 1.8 = \mathbf{9 R}$

2-79E A house is losing heat at a rate of 4500 kJ/h per °C temperature difference between the indoor and the outdoor temperatures. The rate of heat loss is to be expressed per °F, K, and R of temperature difference between the indoor and the outdoor temperatures.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the rate of heat loss from the house is

- (a) **4500 kJ/h** per K difference in temperature, and
- (b), (c) $4500/1.8 = \mathbf{2500 kJ/h}$ per R or °F rise in temperature.

2-80 The average temperature of the atmosphere is expressed as $T_{\text{atm}} = 288.15 - 6.5z$ where z is altitude in km. The temperature outside an airplane cruising at 12,000 m is to be determined.

Analysis Using the relation given, the average temperature of the atmosphere at an altitude of 12,000 m is determined to be

$$\begin{aligned} T_{\text{atm}} &= 288.15 - 6.5z \\ &= 288.15 - 6.5 \times 12 \\ &= \mathbf{210.15 \text{ K} = -63^\circ\text{C}} \end{aligned}$$

Discussion This is the “average” temperature. The actual temperature at different times can be different.

2-81 A new “Smith” absolute temperature scale is proposed, and a value of 1000 S is assigned to the boiling point of water. The ice point on this scale, and its relation to the Kelvin scale are to be determined.

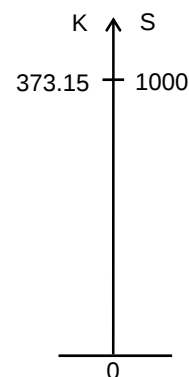
Analysis All linear absolute temperature scales read zero at absolute zero pressure, and are constant multiples of each other. For example, $T(\text{R}) = 1.8 T(\text{K})$. That is, multiplying a temperature value in K by 1.8 will give the same temperature in R.

The proposed temperature scale is an acceptable absolute temperature scale since it differs from the other absolute temperature scales by a constant only. The boiling temperature of water in the Kelvin and the Smith scales are 373.15 K and 1000 K, respectively. Therefore, these two temperature scales are related to each other by

$$T(\text{S}) = \frac{1000}{373.15} T(\text{K}) = \mathbf{2.6799 \text{ T(K)}}$$

The ice point of water on the Smith scale is

$$T(\text{S})_{\text{ice}} = 2.6799 T(\text{K})_{\text{ice}} = 2.6799 \times 273.15 = \mathbf{732.0 \text{ S}}$$



2-82E An expression for the equivalent wind chill temperature is given in English units. It is to be converted to SI units.

Analysis The required conversion relations are $1 \text{ mph} = 1.609 \text{ km/h}$ and $T(^{\circ}\text{F}) = 1.8T(^{\circ}\text{C}) + 32$. The first thought that comes to mind is to replace $T(^{\circ}\text{F})$ in the equation by its equivalent $1.8T(^{\circ}\text{C}) + 32$, and V in mph by 1.609 km/h , which is the “regular” way of converting units. However, the equation we have is not a regular dimensionally homogeneous equation, and thus the regular rules do not apply. The V in the equation is a constant whose value is equal to the numerical value of the velocity in mph. Therefore, if V is given in km/h, we should divide it by 1.609 to convert it to the desired unit of mph. That is,

$$T_{\text{equiv}}(^{\circ}\text{F}) = 91.4 - [91.4 - T_{\text{ambient}}(^{\circ}\text{F})][0.475 - 0.0203(V / 1.609) + 0.304\sqrt{V / 1.609}]$$

or

$$T_{\text{equiv}}(^{\circ}\text{F}) = 91.4 - [91.4 - T_{\text{ambient}}(^{\circ}\text{F})][0.475 - 0.0126V + 0.240\sqrt{V}]$$

where V is in km/h. Now the problem reduces to converting a temperature in $^{\circ}\text{F}$ to a temperature in $^{\circ}\text{C}$, using the proper convection relation:

$$1.8T_{\text{equiv}}(^{\circ}\text{C}) + 32 = 91.4 - [91.4 - (1.8T_{\text{ambient}}(^{\circ}\text{C}) + 32)][0.475 - 0.0126V + 0.240\sqrt{V}]$$

which simplifies to

$$T_{\text{equiv}}(^{\circ}\text{C}) = 33.0 - (33.0 - T_{\text{ambient}})(0.475 - 0.0126V + 0.240\sqrt{V})$$

where the ambient air temperature is in $^{\circ}\text{C}$.

2-83E EES Problem 2-82E is reconsidered. The equivalent wind-chill temperatures in °F as a function of wind velocity in the range of 4 mph to 100 mph for the ambient temperatures of 20, 40, and 60°F are to be plotted, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

"Obtain V and T_ambient from the Diagram Window"

{T_ambient=10

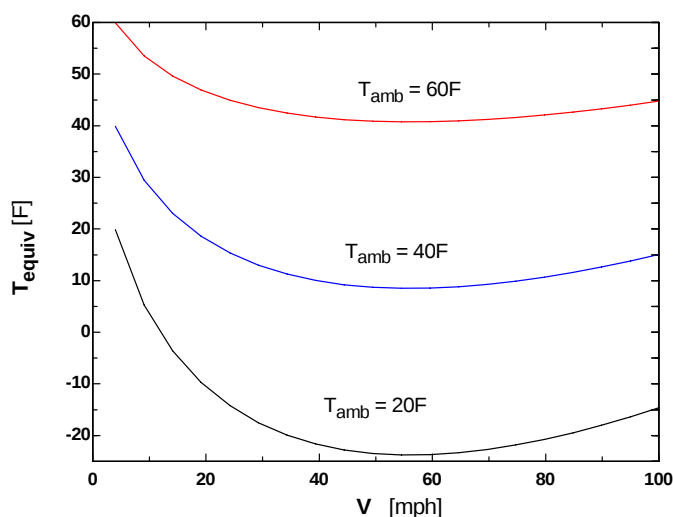
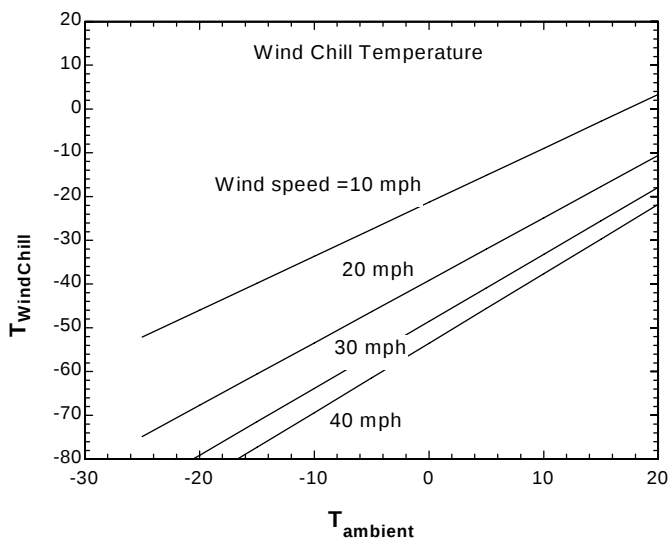
V=20}

V_use=max(V,4)

T_equiv=91.4-(91.4-T_ambient)*(0.475 - 0.0203*V_use + 0.304*sqrt(V_use))

"The parametric table was used to generate the plot, Fill in values for T_ambient and V (use Alter Values under Tables menu) then use F3 to solve table. Plot the first 10 rows and then overlay the second ten, and so on. Place the text on the plot using Add Text under the Plot menu."

T _{equiv} [F]	T _{ambient} [F]	V [mph]
-52	-25	10
-46	-20	10
-40	-15	10
-34	-10	10
-27	-5	10
-21	0	10
-15	5	10
-9	10	10
-3	15	10
3	20	10
-75	-25	20
-68	-20	20
-61	-15	20
-53	-10	20
-46	-5	20
-39	0	20
-32	5	20
-25	10	20
-18	15	20
-11	20	20
-87	-25	30
-79	-20	30
-72	-15	30
-64	-10	30
-56	-5	30
-49	0	30
-41	5	30
-33	10	30
-26	15	30
-18	20	30
-93	-25	40
-85	-20	40
-77	-15	40
-69	-10	40
-61	-5	40
-54	0	40
-46	5	40
-38	10	40
-30	15	40
-22	20	40

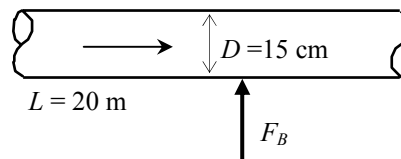


2-84 One section of the duct of an air-conditioning system is laid underwater. The upward force the water will exert on the duct is to be determined.

Assumptions 1 The diameter given is the outer diameter of the duct (or, the thickness of the duct material is negligible). 2 The weight of the duct and the air in is negligible.

Properties The density of air is given to be $\rho = 1.30 \text{ kg/m}^3$. We take the density of water to be 1000 kg/m^3 .

Analysis Noting that the weight of the duct and the air in it is negligible, the net upward force acting on the duct is the buoyancy force exerted by water. The volume of the underground section of the duct is



$$V = AL = (\pi D^2 / 4)L = [\pi(0.15 \text{ m})^2 / 4](20 \text{ m}) = 0.353 \text{ m}^3$$

Then the buoyancy force becomes

$$F_B = \rho g V = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.353 \text{ m}^3) \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) = \mathbf{3.46 \text{ kN}}$$

Discussion The upward force exerted by water on the duct is 3.46 kN, which is equivalent to the weight of a mass of 353 kg. Therefore, this force must be treated seriously.

2-85 A helium balloon tied to the ground carries 2 people. The acceleration of the balloon when it is first released is to be determined.

Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be $\rho = 1.16 \text{ kg/m}^3$. The density of helium gas is $1/7^{\text{th}}$ of this.

Analysis The buoyancy force acting on the balloon is

$$\begin{aligned} V_{\text{balloon}} &= 4\pi r^3/3 = 4\pi(5 \text{ m})^3/3 = 523.6 \text{ m}^3 \\ F_B &= \rho_{\text{air}} g V_{\text{balloon}} \\ &= (1.16 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(523.6 \text{ m}^3) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) = 5958 \text{ N} \end{aligned}$$

The total mass is

$$\begin{aligned} m_{\text{He}} &= \rho_{\text{He}} V = \left(\frac{1.16}{7} \text{ kg/m}^3 \right) (523.6 \text{ m}^3) = 86.8 \text{ kg} \\ m_{\text{total}} &= m_{\text{He}} + m_{\text{people}} = 86.8 + 2 \times 70 = 226.8 \text{ kg} \end{aligned}$$

The total weight is

$$W = m_{\text{total}} g = (226.8 \text{ kg})(9.81 \text{ m/s}^2) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) = 2225 \text{ N}$$

Thus the net force acting on the balloon is

$$F_{\text{net}} = F_B - W = 5958 - 2225 = 3733 \text{ N}$$

Then the acceleration becomes

$$a = \frac{F_{\text{net}}}{m_{\text{total}}} = \frac{3733 \text{ N}}{226.8 \text{ kg}} \left(\frac{1 \text{ kg} \cdot \text{m/s}^2}{1 \text{ N}} \right) = \mathbf{16.5 \text{ m/s}^2}$$



2-86 EES Problem 2-85 is reconsidered. The effect of the number of people carried in the balloon on acceleration is to be investigated. Acceleration is to be plotted against the number of people, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

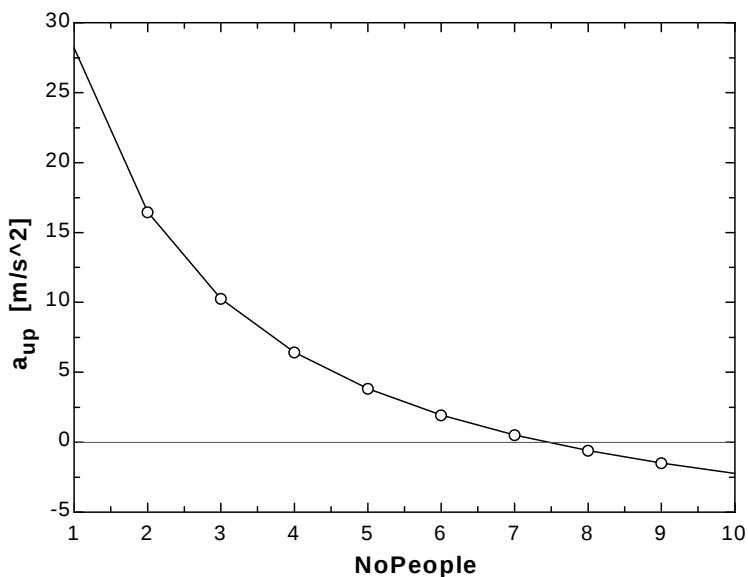
"Given Data:"

rho_air=1.16"[kg/m^3]" "density of air"
 g=9.807"[m/s^2]"
 d_balloon=10"[m]"
 m_1person=70"[kg]"
 {NoPeople = 2} "Data supplied in Parametric Table"

"Calculated values:"

rho_He=rho_air/7"[kg/m^3]" "density of helium"
 r_balloon=d_balloon/2"[m]"
 V_balloon=4*pi*r_balloon^3/3"[m^3]"
 m_people=NoPeople*m_1person"[kg]"
 m_He=rho_He*V_balloon"[kg]"
 m_total=m_He+m_people"[kg]"
 "The total weight of balloon and people is:"
 W_total=m_total*g"[N]"
 "The buoyancy force acting on the balloon, F_b, is equal to the weight of the air displaced by the balloon."
 F_b=rho_air*V_balloon*g"[N]"
 "From the free body diagram of the balloon, the balancing vertical forces must equal the product of the total mass and the vertical acceleration:"
 F_b - W_total=m_total*a_up

A_{up} [m/s ²]	NoPeople
28.19	1
16.46	2
10.26	3
6.434	4
3.831	5
1.947	6
0.5204	7
-0.5973	8
-1.497	9
-2.236	10



2-87 A balloon is filled with helium gas. The maximum amount of load the balloon can carry is to be determined.

Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be $\rho = 1.16 \text{ kg/m}^3$. The density of helium gas is 1/7th of this.

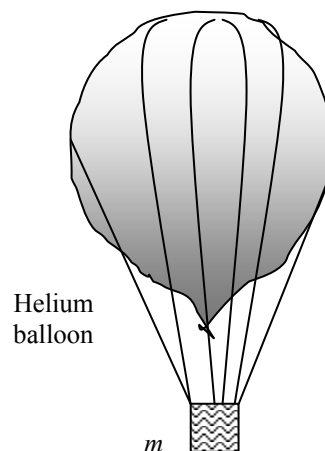
Analysis In the limiting case, the net force acting on the balloon will be zero. That is, the buoyancy force and the weight will balance each other:

$$W = mg = F_B$$

$$m_{\text{total}} = \frac{F_B}{g} = \frac{5958 \text{ N}}{9.81 \text{ m/s}^2} = 607.3 \text{ kg}$$

Thus,

$$m_{\text{people}} = m_{\text{total}} - m_{\text{He}} = 607.3 - 86.8 = \mathbf{520.5 \text{ kg}}$$



2-88E The pressure in a steam boiler is given in kgf/cm^2 . It is to be expressed in psi, kPa, atm, and bars.

Analysis We note that $1 \text{ atm} = 1.03323 \text{ kgf/cm}^2$, $1 \text{ atm} = 14.696 \text{ psi}$, $1 \text{ atm} = 101.325 \text{ kPa}$, and $1 \text{ atm} = 1.01325 \text{ bar}$ (inner cover page of text). Then the desired conversions become:

In atm:
$$P = (92 \text{ kgf/cm}^2) \left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2} \right) = \mathbf{89.04 \text{ atm}}$$

In psi:
$$P = (92 \text{ kgf/cm}^2) \left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2} \right) \left(\frac{14.696 \text{ psi}}{1 \text{ atm}} \right) = \mathbf{1309 \text{ psi}}$$

In kPa:
$$P = (92 \text{ kgf/cm}^2) \left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2} \right) \left(\frac{101.325 \text{ kPa}}{1 \text{ atm}} \right) = \mathbf{9022 \text{ kPa}}$$

In bars:
$$P = (92 \text{ kgf/cm}^2) \left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2} \right) \left(\frac{1.01325 \text{ bar}}{1 \text{ atm}} \right) = \mathbf{90.22 \text{ bar}}$$

Discussion Note that the units atm, kgf/cm^2 , and bar are almost identical to each other.

2-89 A 10-m high cylindrical container is filled with equal volumes of water and oil. The pressure difference between the top and the bottom of the container is to be determined.

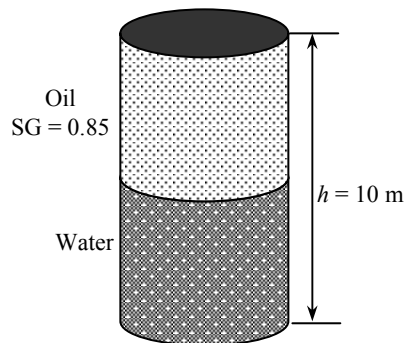
Properties The density of water is given to be $\rho = 1000 \text{ kg/m}^3$. The specific gravity of oil is given to be 0.85.

Analysis The density of the oil is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{\text{H}_2\text{O}} = (0.85)(1000 \text{ kg/m}^3) = 850 \text{ kg/m}^3$$

The pressure difference between the top and the bottom of the cylinder is the sum of the pressure differences across the two fluids,

$$\begin{aligned} \Delta P_{\text{total}} &= \Delta P_{\text{oil}} + \Delta P_{\text{water}} = (\rho g h)_{\text{oil}} + (\rho g h)_{\text{water}} \\ &= \left[(850 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(5 \text{ m}) + (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(5 \text{ m}) \right] \left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2} \right) \\ &= \mathbf{90.7 \text{ kPa}} \end{aligned}$$



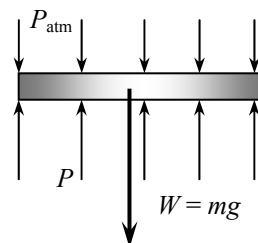
2-90 The pressure of a gas contained in a vertical piston-cylinder device is measured to be 250 kPa. The mass of the piston is to be determined.

Assumptions There is no friction between the piston and the cylinder.

Analysis Drawing the free body diagram of the piston and balancing the vertical forces yield

$$\begin{aligned} W &= PA - P_{\text{atm}}A \\ mg &= (P - P_{\text{atm}})A \\ (m)(9.81 \text{ m/s}^2) &= (250 - 100 \text{ kPa})(30 \times 10^{-4} \text{ m}^2) \left(\frac{1000 \text{ kg/m} \cdot \text{s}^2}{1 \text{ kPa}} \right) \end{aligned}$$

It yields $m = \mathbf{45.9 \text{ kg}}$

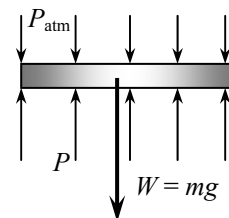


2-91 The gage pressure in a pressure cooker is maintained constant at 100 kPa by a petcock. The mass of the petcock is to be determined.

Assumptions There is no blockage of the pressure release valve.

Analysis Atmospheric pressure is acting on all surfaces of the petcock, which balances itself out. Therefore, it can be disregarded in calculations if we use the gage pressure as the cooker pressure. A force balance on the petcock ($\Sigma F_y = 0$) yields

$$\begin{aligned}
 W &= P_{\text{gage}} A \\
 m &= \frac{P_{\text{gage}} A}{g} = \frac{(100 \text{ kPa})(4 \times 10^{-6} \text{ m}^2)}{9.81 \text{ m/s}^2} \left(\frac{1000 \text{ kg/m} \cdot \text{s}^2}{1 \text{ kPa}} \right) \\
 &= \mathbf{0.0408 \text{ kg}}
 \end{aligned}$$



2-92 A glass tube open to the atmosphere is attached to a water pipe, and the pressure at the bottom of the tube is measured. It is to be determined how high the water will rise in the tube.

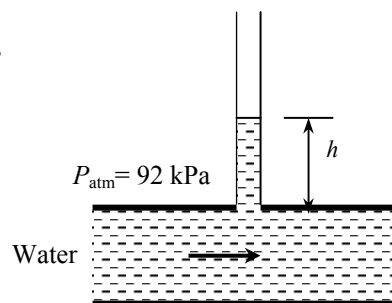
Properties The density of water is given to be $\rho = 1000 \text{ kg/m}^3$.

Analysis The pressure at the bottom of the tube can be expressed as

$$P = P_{\text{atm}} + (\rho g h)_{\text{tube}}$$

Solving for h ,

$$\begin{aligned}
 h &= \frac{P - P_{\text{atm}}}{\rho g} \\
 &= \frac{(115 - 92) \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \left(\frac{1 \text{ kg} \cdot \text{m/s}^2}{1 \text{ N}} \right) \left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}} \right) \\
 &= \mathbf{2.34 \text{ m}}
 \end{aligned}$$



2-93 The average atmospheric pressure is given as $P_{\text{atm}} = 101.325(1 - 0.02256z)^{5.256}$ where z is the altitude in km. The atmospheric pressures at various locations are to be determined.

Analysis The atmospheric pressures at various locations are obtained by substituting the altitude z values in km into the relation

$$P_{\text{atm}} = 101.325(1 - 0.02256z)^{5.256}$$

Atlanta: $(z = 0.306 \text{ km}): P_{\text{atm}} = 101.325(1 - 0.02256 \times 0.306)^{5.256} = \mathbf{97.7 \text{ kPa}}$

Denver: $(z = 1.610 \text{ km}): P_{\text{atm}} = 101.325(1 - 0.02256 \times 1.610)^{5.256} = \mathbf{83.4 \text{ kPa}}$

M. City: $(z = 2.309 \text{ km}): P_{\text{atm}} = 101.325(1 - 0.02256 \times 2.309)^{5.256} = \mathbf{76.5 \text{ kPa}}$

Mt. Ev.: $(z = 8.848 \text{ km}): P_{\text{atm}} = 101.325(1 - 0.02256 \times 8.848)^{5.256} = \mathbf{31.4 \text{ kPa}}$

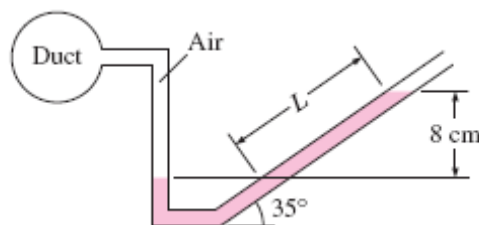
2-94 The air pressure in a duct is measured by an inclined manometer. For a given vertical level difference, the gage pressure in the duct and the length of the differential fluid column are to be determined.

Assumptions The manometer fluid is an incompressible substance.

Properties The density of the liquid is given to be $\rho = 0.81 \text{ kg/L} = 810 \text{ kg/m}^3$.

Analysis The gage pressure in the duct is determined from

$$\begin{aligned} P_{\text{gage}} &= P_{\text{abs}} - P_{\text{atm}} = \rho gh \\ &= (810 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.08 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ Pa}}{1 \text{ N/m}^2} \right) \\ &= \mathbf{636 \text{ Pa}} \end{aligned}$$



The length of the differential fluid column is

$$L = h / \sin \theta = (8 \text{ cm}) / \sin 35^\circ = \mathbf{13.9 \text{ cm}}$$

Discussion Note that the length of the differential fluid column is extended considerably by inclining the manometer arm for better readability.

2-95E Equal volumes of water and oil are poured into a U-tube from different arms, and the oil side is pressurized until the contact surface of the two fluids moves to the bottom and the liquid levels in both arms become the same. The excess pressure applied on the oil side is to be determined.

Assumptions 1 Both water and oil are incompressible substances. 2 Oil does not mix with water. 3 The cross-sectional area of the U-tube is constant.

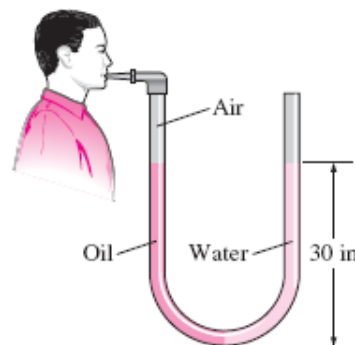
Properties The density of oil is given to be $\rho_{\text{oil}} = 49.3 \text{ lbfm/ft}^3$. We take the density of water to be $\rho_{\text{w}} = 62.4 \text{ lbfm/ft}^3$.

Analysis Noting that the pressure of both the water and the oil is the same at the contact surface, the pressure at this surface can be expressed as

$$P_{\text{contact}} = P_{\text{blow}} + \rho_{\text{a}} gh_{\text{a}} = P_{\text{atm}} + \rho_{\text{w}} gh_{\text{w}}$$

Noting that $h_{\text{a}} = h_{\text{w}}$ and rearranging,

$$\begin{aligned} P_{\text{gage,blow}} &= P_{\text{blow}} - P_{\text{atm}} = (\rho_{\text{w}} - \rho_{\text{oil}}) gh \\ &= (62.4 - 49.3 \text{ lbfm/ft}^3)(32.2 \text{ ft/s}^2)(30/12 \text{ ft}) \left(\frac{1 \text{ lbf}}{32.2 \text{ lbfm} \cdot \text{ft/s}^2} \right) \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2} \right) \\ &= \mathbf{0.227 \text{ psi}} \end{aligned}$$



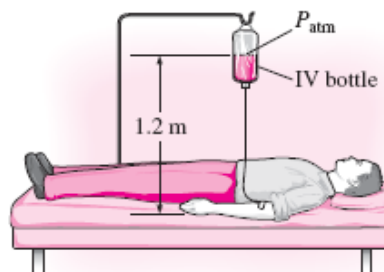
Discussion When the person stops blowing, the oil will rise and some water will flow into the right arm. It can be shown that when the curvature effects of the tube are disregarded, the differential height of water will be 23.7 in to balance 30-in of oil.

2-96 It is given that an IV fluid and the blood pressures balance each other when the bottle is at a certain height, and a certain gage pressure at the arm level is needed for sufficient flow rate. The gage pressure of the blood and elevation of the bottle required to maintain flow at the desired rate are to be determined.

Assumptions 1 The IV fluid is incompressible. 2 The IV bottle is open to the atmosphere.

Properties The density of the IV fluid is given to be $\rho = 1020 \text{ kg/m}^3$.

Analysis (a) Noting that the IV fluid and the blood pressures balance each other when the bottle is 1.2 m above the arm level, the gage pressure of the blood in the arm is simply equal to the gage pressure of the IV fluid at a depth of 1.2 m,



$$\begin{aligned} P_{\text{gage, arm}} &= P_{\text{abs}} - P_{\text{atm}} = \rho g h_{\text{arm-bottle}} \\ &= (1020 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(1.20 \text{ m}) \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) \\ &= \mathbf{12.0 \text{ kPa}} \end{aligned}$$

(b) To provide a gage pressure of 20 kPa at the arm level, the height of the bottle from the arm level is again determined from $P_{\text{gage, arm}} = \rho g h_{\text{arm-bottle}}$ to be

$$\begin{aligned} h_{\text{arm-bottle}} &= \frac{P_{\text{gage, arm}}}{\rho g} \\ &= \frac{20 \text{ kPa}}{(1020 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} \left(\frac{1000 \text{ kg} \cdot \text{m/s}^2}{1 \text{ kN}} \right) \left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}} \right) = \mathbf{2.0 \text{ m}} \end{aligned}$$

Discussion Note that the height of the reservoir can be used to control flow rates in gravity driven flows. When there is flow, the pressure drop in the tube due to friction should also be considered. This will result in raising the bottle a little higher to overcome pressure drop.

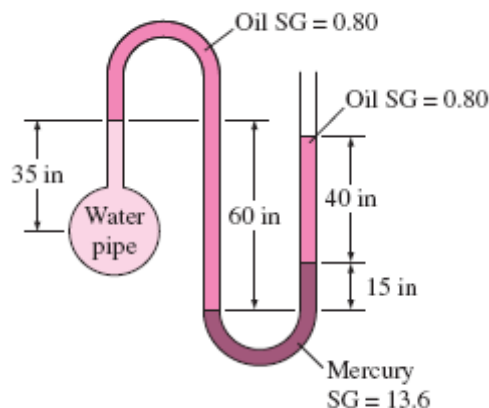
2-97E A water pipe is connected to a double-U manometer whose free arm is open to the atmosphere. The absolute pressure at the center of the pipe is to be determined.

Assumptions 1 All the liquids are incompressible.

2 The solubility of the liquids in each other is negligible.

Properties The specific gravities of mercury and oil are given to be 13.6 and 0.80, respectively. We take the density of water to be $\rho_w = 62.4 \text{ lbm/ft}^3$.

Analysis Starting with the pressure at the center of the water pipe, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives



$$P_{\text{water pipe}} - \rho_{\text{water}} g h_{\text{water}} + \rho_{\text{oil}} g h_{\text{oil}} - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{oil}} g h_{\text{oil}} = P_{\text{atm}}$$

Solving for $P_{\text{water pipe}}$,

$$P_{\text{water pipe}} = P_{\text{atm}} + \rho_{\text{water}} g (h_{\text{water}} - SG_{\text{oil}} h_{\text{oil}} + SG_{\text{Hg}} h_{\text{Hg}} + SG_{\text{oil}} h_{\text{oil}})$$

Substituting,

$$\begin{aligned} P_{\text{water pipe}} &= 14.2 \text{ psia} + (62.4 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)[(35/12 \text{ ft}) - 0.8(60/12 \text{ ft}) + 13.6(15/12 \text{ ft}) \\ &\quad + 0.8(40/12 \text{ ft})] \times \left(\frac{1 \text{ lbf}}{32.2 \text{ lbm} \cdot \text{ft/s}^2} \right) \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2} \right) \\ &= \mathbf{22.3 \text{ psia}} \end{aligned}$$

Therefore, the absolute pressure in the water pipe is 22.3 psia.

Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

2-98 The temperature of the atmosphere varies with altitude z as $T = T_0 - \beta z$, while the gravitational acceleration varies by $g(z) = g_0 / (1 + z / 6,370,320)^2$. Relations for the variation of pressure in atmosphere are to be obtained (a) by ignoring and (b) by considering the variation of g with altitude.

Assumptions The air in the troposphere behaves as an ideal gas.

Analysis (a) Pressure change across a differential fluid layer of thickness dz in the vertical z direction is

$$dP = -\rho g dz$$

From the ideal gas relation, the air density can be expressed as $\rho = \frac{P}{RT} = \frac{P}{R(T_0 - \beta z)}$. Then,

$$dP = -\frac{P}{R(T_0 - \beta z)} g dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^P \frac{dP}{P} = -\int_0^z \frac{g dz}{R(T_0 - \beta z)}$$

Performing the integrations,

$$\ln \frac{P}{P_0} = \frac{g}{R\beta} \ln \frac{T_0 - \beta z}{T_0}$$

Rearranging, the desired relation for atmospheric pressure for the case of constant g becomes

$$P = P_0 \left(1 - \frac{\beta z}{T_0} \right)^{\frac{g}{R\beta}}$$

(b) When the variation of g with altitude is considered, the procedure remains the same but the expressions become more complicated,

$$dP = -\frac{P}{R(T_0 - \beta z)} \frac{g_0}{(1 + z / 6,370,320)^2} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^P \frac{dP}{P} = -\int_0^z \frac{g_0 dz}{R(T_0 - \beta z)(1 + z / 6,370,320)^2}$$

Performing the integrations,

$$\ln P \Big|_{P_0}^P = \frac{g_0}{R\beta} \left[\frac{1}{(1 + kT_0 / \beta)(1 + kz)} - \frac{1}{(1 + kT_0 / \beta)^2} \ln \frac{1 + kz}{T_0 - \beta z} \right]_0^z$$

where $R = 287 \text{ J/kg} \cdot \text{K} = 287 \text{ m}^2/\text{s}^2 \cdot \text{K}$ is the gas constant of air. After some manipulations, we obtain

$$P = P_0 \exp \left[-\frac{g_0}{R(\beta + kT_0)} \left(\frac{1}{1 + 1/kz} + \frac{1}{1 + kT_0 / \beta} \ln \frac{1 + kz}{1 - \beta z / T_0} \right) \right]$$

where $T_0 = 288.15 \text{ K}$, $\beta = 0.0065 \text{ K/m}$, $g_0 = 9.807 \text{ m/s}^2$, $k = 1/6,370,320 \text{ m}^{-1}$, and z is the elevation in m..

Discussion When performing the integration in part (b), the following expression from integral tables is used, together with a transformation of variable $x = T_0 - \beta z$,

$$\int \frac{dx}{x(a + bx)^2} = \frac{1}{a(a + bx)} - \frac{1}{a^2} \ln \frac{a + bx}{x}$$

Also, for $z = 11,000 \text{ m}$, for example, the relations in (a) and (b) give 22.62 and 22.69 kPa, respectively.

2-99 The variation of pressure with density in a thick gas layer is given. A relation is to be obtained for pressure as a function of elevation z .

Assumptions The property relation $P = C\rho^n$ is valid over the entire region considered.

Analysis The pressure change across a differential fluid layer of thickness dz in the vertical z direction is given as,

$$dP = -\rho g dz$$

Also, the relation $P = C\rho^n$ can be expressed as $C = P / \rho^n = P_0 / \rho_0^n$, and thus

$$\rho = \rho_0 (P / P_0)^{1/n}$$

Substituting,

$$dP = -g\rho_0 (P / P_0)^{1/n} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0 = C\rho_0^n$ to $z = z$ where $P = P$,

$$\int_{P_0}^P (P / P_0)^{-1/n} dP = -\rho_0 g \int_0^z dz$$

Performing the integrations.

$$P_0 \frac{(P / P_0)^{-1/n+1}}{-1/n+1} \Big|_{P_0}^P = -\rho_0 g z \quad \rightarrow \quad \left(\frac{P}{P_0} \right)^{(n-1)/n} - 1 = -\frac{n-1}{n} \frac{\rho_0 g z}{P_0}$$

Solving for P ,

$$P = P_0 \left(1 - \frac{n-1}{n} \frac{\rho_0 g z}{P_0} \right)^{n/(n-1)}$$

which is the desired relation.

Discussion The final result could be expressed in various forms. The form given is very convenient for calculations as it facilitates unit cancellations and reduces the chance of error.

2-100 A pressure transducer is used to measure pressure by generating analogue signals, and it is to be calibrated by measuring both the pressure and the electric current simultaneously for various settings, and the results are tabulated. A calibration curve in the form of $P = aI + b$ is to be obtained, and the pressure corresponding to a signal of 10 mA is to be calculated.

Assumptions Mercury is an incompressible liquid.

Properties The specific gravity of mercury is given to be 13.56, and thus its density is $13,560 \text{ kg/m}^3$.

Analysis For a given differential height, the pressure can be calculated from

$$P = \rho g \Delta h$$

For $\Delta h = 28.0 \text{ mm} = 0.0280 \text{ m}$, for example,

$$P = 13.56(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.0280 \text{ m}) \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) = 3.75 \text{ kPa}$$

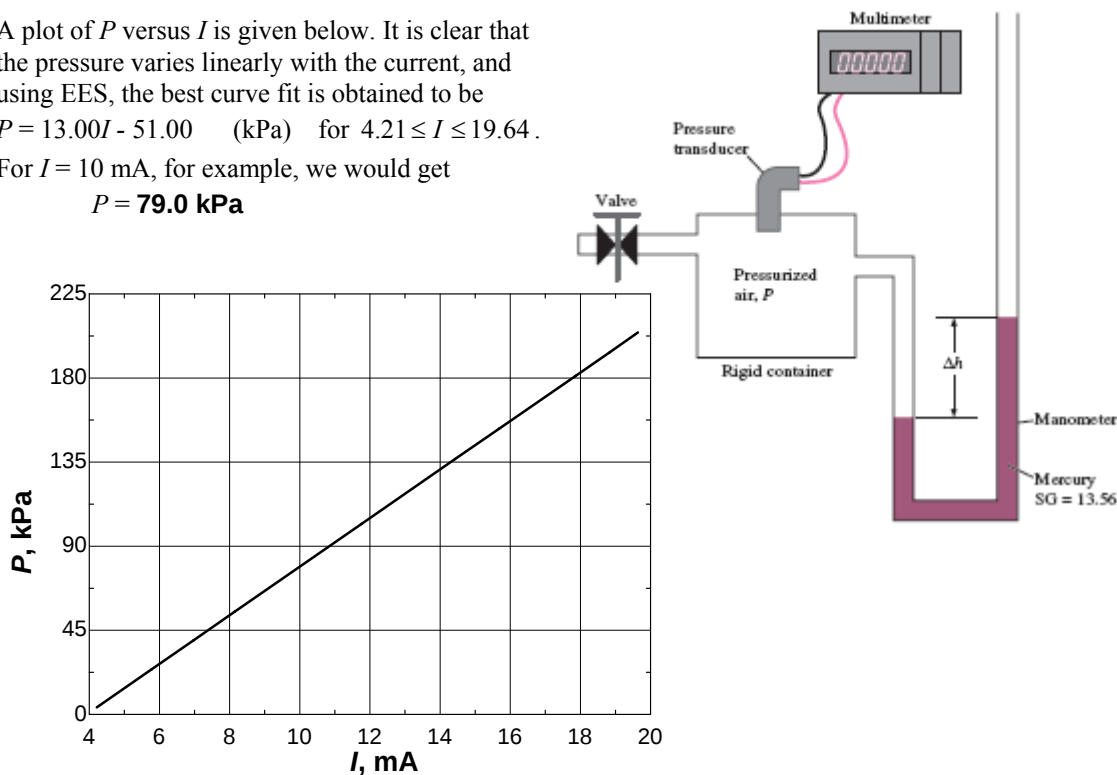
Repeating the calculations and tabulating, we have

$\Delta h(\text{mm})$	28.0	181.5	297.8	413.1	765.9	1027	1149	1362	1458	1536
$P(\text{kPa})$	3.73	24.14	39.61	54.95	101.9	136.6	152.8	181.2	193.9	204.3
$I(\text{mA})$	4.21	5.78	6.97	8.15	11.76	14.43	15.68	17.86	18.84	19.64

A plot of P versus I is given below. It is clear that the pressure varies linearly with the current, and using EES, the best curve fit is obtained to be $P = 13.00I - 51.00$ (kPa) for $4.21 \leq I \leq 19.64$.

For $I = 10 \text{ mA}$, for example, we would get

$$P = \mathbf{79.0 \text{ kPa}}$$



Discussion Note that the calibration relation is valid in the specified range of currents or pressures.

2-101 Design and Essay Problems



Solutions Manual

for

Introduction to Thermodynamics and Heat Transfer**Yunus A. Cengel****2nd Edition, 2008****Chapter 3****ENERGY, ENERGY TRANSFER, AND
GENERAL ENERGY ANALYSIS****PROPRIETARY AND CONFIDENTIAL**

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Forms of Energy

3-1C Initially, the rock possesses potential energy relative to the bottom of the sea. As the rock falls, this potential energy is converted into kinetic energy. Part of this kinetic energy is converted to thermal energy as a result of frictional heating due to air resistance, which is transferred to the air and the rock. Same thing happens in water. Assuming the impact velocity of the rock at the sea bottom is negligible, the entire potential energy of the rock is converted to thermal energy in water and air.

3-2C Hydrogen is also a fuel, since it can be burned, but it is not an energy source since there are no hydrogen reserves in the world. Hydrogen can be obtained from water by using another energy source, such as solar or nuclear energy, and then the hydrogen obtained can be used as a fuel to power cars or generators. Therefore, it is more proper to view hydrogen as an energy carrier than an energy source.

3-3C The *macroscopic* forms of energy are those a system possesses as a whole with respect to some outside reference frame. The *microscopic* forms of energy, on the other hand, are those related to the molecular structure of a system and the degree of the molecular activity, and are independent of outside reference frames.

3-4C The sum of all forms of the energy a system possesses is called *total energy*. In the absence of magnetic, electrical and surface tension effects, the total energy of a system consists of the kinetic, potential, and internal energies.

3-5C Thermal energy is the sensible and latent forms of internal energy, and it is referred to as heat in daily life.

3-6C The *mechanical energy* is the form of energy that can be converted to mechanical work completely and directly by a mechanical device such as a propeller. It differs from thermal energy in that thermal energy cannot be converted to work directly and completely. The forms of mechanical energy of a fluid stream are kinetic, potential, and flow energies.

3-7E The specific kinetic energy of a mass whose velocity is given is to be determined.

Analysis According to the definition of the specific kinetic energy,

$$ke = \frac{V^2}{2} = \frac{(100 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = \mathbf{0.200 \text{ Btu/lbm}}$$

3-8 The specific kinetic energy of a mass whose velocity is given is to be determined.

Analysis Substitution of the given data into the expression for the specific kinetic energy gives

$$ke = \frac{V^2}{2} = \frac{(30 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{0.45 \text{ kJ/kg}}$$

3-9E The total potential energy of an object that is below a reference level is to be determined.

Analysis Substituting the given data into the potential energy expression gives

$$PE = mgz = (100 \text{ lbm})(31.7 \text{ ft/s}^2)(-20 \text{ ft}) \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = \mathbf{-2.53 \text{ Btu}}$$

3-10 The specific potential energy of an object is to be determined.

Analysis The specific potential energy is given by

$$pe = gz = (9.8 \text{ m/s}^2)(50 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{0.49 \text{ kJ/kg}}$$

3-11 The total potential energy of an object is to be determined.

Analysis Substituting the given data into the potential energy expression gives

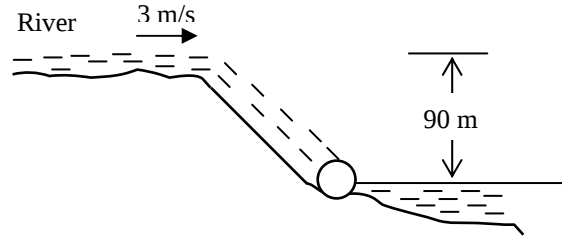
$$PE = mgz = (100 \text{ kg})(9.8 \text{ m/s}^2)(20 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{19.6 \text{ kJ}}$$

3-12 A river is flowing at a specified velocity, flow rate, and elevation. The total mechanical energy of the river water per unit mass, and the power generation potential of the entire river are to be determined.

Assumptions 1 The elevation given is the elevation of the free surface of the river. 2 The velocity given is the average velocity. 3 The mechanical energy of water at the turbine exit is negligible.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis Noting that the sum of the flow energy and the potential energy is constant for a given fluid body, we can take the elevation of the entire river water to be the elevation of the free surface, and ignore the flow energy. Then the total mechanical energy of the river water per unit mass becomes



$$e_{\text{mech}} = pe + ke = gh + \frac{V^2}{2} = \left((9.81 \text{ m/s}^2)(90 \text{ m}) + \frac{(3 \text{ m/s})^2}{2} \right) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{0.887 \text{ kJ/kg}}$$

The power generation potential of the river water is obtained by multiplying the total mechanical energy by the mass flow rate,

$$\dot{m} = \rho \dot{V} = (1000 \text{ kg/m}^3)(500 \text{ m}^3/\text{s}) = 500,000 \text{ kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (500,000 \text{ kg/s})(0.887 \text{ kJ/kg}) = 444,000 \text{ kW} = \mathbf{444 \text{ MW}}$$

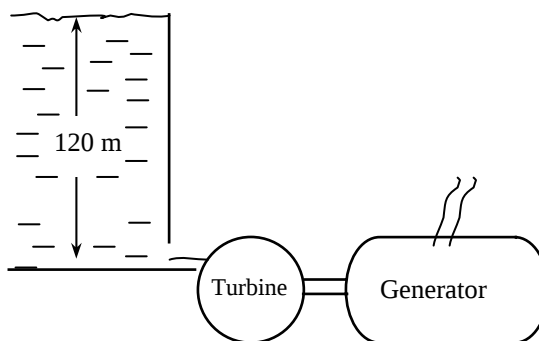
Therefore, 444 MW of power can be generated from this river as it discharges into the lake if its power potential can be recovered completely.

Discussion Note that the kinetic energy of water is negligible compared to the potential energy, and it can be ignored in the analysis. Also, the power output of an actual turbine will be less than 444 MW because of losses and inefficiencies.

3-13 A hydraulic turbine-generator is to generate electricity from the water of a large reservoir. The power generation potential is to be determined.

Assumptions 1 The elevation of the reservoir remains constant. 2 The mechanical energy of water at the turbine exit is negligible.

Analysis The total mechanical energy water in a reservoir possesses is equivalent to the potential energy of water at the free surface, and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate.



$$e_{\text{mech}} = pe = gz = (9.81 \text{ m/s}^2)(120 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 1.177 \text{ kJ/kg}$$

Then the power generation potential becomes

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (1500 \text{ kg/s})(1.177 \text{ kJ/kg}) \left(\frac{1 \text{ kW}}{1 \text{ kJ/s}} \right) = \mathbf{1766 \text{ kW}}$$

Therefore, the reservoir has the potential to generate 1766 kW of power.

Discussion This problem can also be solved by considering a point at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

3-14 Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass and the power generation potential are to be determined.

Assumptions The wind is blowing steadily at a constant uniform velocity.

Properties The density of air is given to be $\rho = 1.25 \text{ kg/m}^3$.

Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

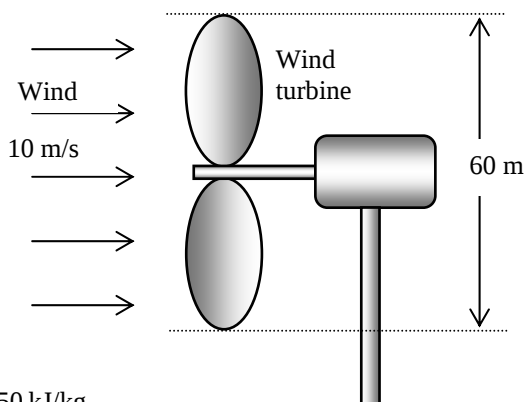
$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(10 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.050 \text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V \frac{\pi D^2}{4} = (1.25 \text{ kg/m}^3)(10 \text{ m/s}) \frac{\pi (60 \text{ m})^2}{4} = 35,340 \text{ kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (35,340 \text{ kg/s})(0.050 \text{ kJ/kg}) = \mathbf{1770 \text{ kW}}$$

Therefore, 1770 kW of actual power can be generated by this wind turbine at the stated conditions.

Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.



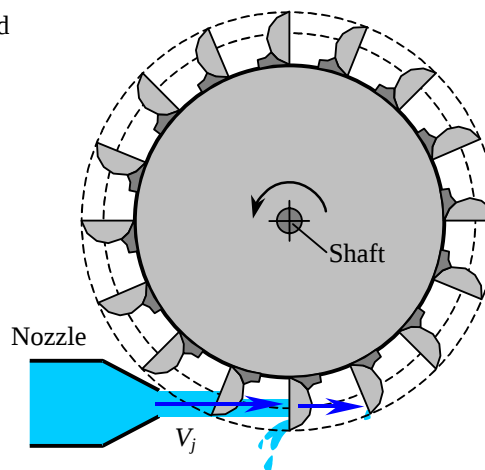
3-15 A water jet strikes the buckets located on the perimeter of a wheel at a specified velocity and flow rate. The power generation potential of this system is to be determined.

Assumptions Water jet flows steadily at the specified speed and flow rate.

Analysis Kinetic energy is the only form of harvestable mechanical energy the water jet possesses, and it can be converted to work entirely. Therefore, the power potential of the water jet is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(60 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 1.8 \text{ kJ/kg}$$

$$\begin{aligned} \dot{W}_{\text{max}} &= \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} \\ &= (120 \text{ kg/s})(1.8 \text{ kJ/kg}) \left(\frac{1 \text{ kW}}{1 \text{ kJ/s}} \right) = \mathbf{216 \text{ kW}} \end{aligned}$$



Therefore, 216 kW of power can be generated by this water jet at the stated conditions.

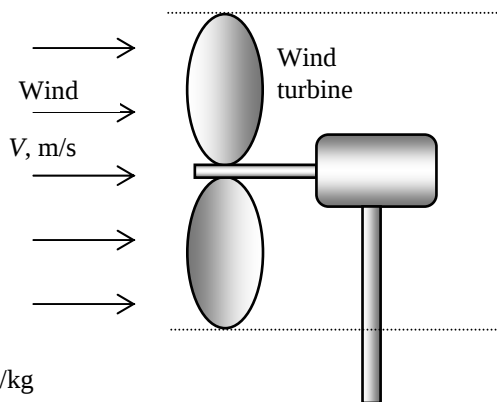
Discussion An actual hydroelectric turbine (such as the Pelton wheel) can convert over 90% of this potential to actual electric power.

3-16 Two sites with specified wind data are being considered for wind power generation. The site better suited for wind power generation is to be determined.

Assumptions 1 The wind is blowing steadily at specified velocity during specified times. 2 The wind power generation is negligible during other times.

Properties We take the density of air to be $\rho = 1.25 \text{ kg/m}^3$ (it does not affect the final answer).

Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate. Considering a unit flow area ($A = 1 \text{ m}^2$), the maximum wind power and power generation becomes



$$e_{\text{mech},1} = ke_1 = \frac{V_1^2}{2} = \frac{(7 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.0245 \text{ kJ/kg}$$

$$e_{\text{mech},2} = ke_2 = \frac{V_2^2}{2} = \frac{(10 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.050 \text{ kJ/kg}$$

$$\dot{W}_{\text{max},1} = \dot{E}_{\text{mech},1} = \dot{m}_1 e_{\text{mech},1} = \rho V_1 A k e_1 = (1.25 \text{ kg/m}^3)(7 \text{ m/s})(1 \text{ m}^2)(0.0245 \text{ kJ/kg}) = 0.2144 \text{ kW}$$

$$\dot{W}_{\text{max},2} = \dot{E}_{\text{mech},2} = \dot{m}_2 e_{\text{mech},2} = \rho V_2 A k e_2 = (1.25 \text{ kg/m}^3)(10 \text{ m/s})(1 \text{ m}^2)(0.050 \text{ kJ/kg}) = 0.625 \text{ kW}$$

since $1 \text{ kW} = 1 \text{ kJ/s}$. Then the maximum electric power generations per year become

$$E_{\text{max},1} = \dot{W}_{\text{max},1} \Delta t_1 = (0.2144 \text{ kW})(3000 \text{ h/yr}) = \mathbf{643 \text{ kWh/yr}} \text{ (per } \text{m}^2 \text{ flow area)}$$

$$E_{\text{max},2} = \dot{W}_{\text{max},2} \Delta t_2 = (0.625 \text{ kW})(2000 \text{ h/yr}) = \mathbf{1250 \text{ kWh/yr}} \text{ (per } \text{m}^2 \text{ flow area)}$$

Therefore, **second site** is a better one for wind generation.

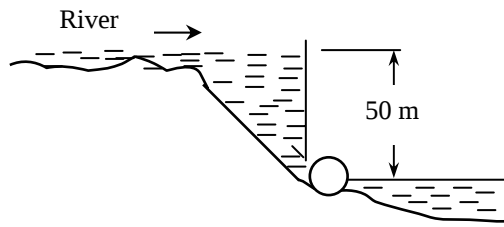
Discussion Note the power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the average wind velocity is the primary consideration in wind power generation decisions.

3-17 A river flowing steadily at a specified flow rate is considered for hydroelectric power generation by collecting the water in a dam. For a specified water height, the power generation potential is to be determined.

Assumptions 1 The elevation given is the elevation of the free surface of the river. 2 The mechanical energy of water at the turbine exit is negligible.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis The total mechanical energy the water in a dam possesses is equivalent to the potential energy of water at the free surface of the dam (relative to free surface of discharge water), and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate.



$$e_{\text{mech}} = pe = gz = (9.81 \text{ m/s}^2)(50 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.4905 \text{ kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho \dot{V} = (1000 \text{ kg/m}^3)(240 \text{ m}^3/\text{s}) = 240,000 \text{ kg/s}$$

Then the power generation potential becomes

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (240,000 \text{ kg/s})(0.4905 \text{ kJ/kg}) \left(\frac{1 \text{ MW}}{1000 \text{ kJ/s}} \right) = \mathbf{118 \text{ MW}}$$

Therefore, 118 MW of power can be generated from this river if its power potential can be recovered completely.

Discussion Note that the power output of an actual turbine will be less than 118 MW because of losses and inefficiencies.

3-18 A person with his suitcase goes up to the 10th floor in an elevator. The part of the energy of the elevator stored in the suitcase is to be determined.

Assumptions 1 The vibrational effects in the elevator are negligible.

Analysis The energy stored in the suitcase is stored in the form of potential energy, which is mgz . Therefore,

$$\Delta E_{\text{suitcase}} = \Delta PE = mg\Delta z = (30 \text{ kg})(9.81 \text{ m/s}^2)(35 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{10.3 \text{ kJ}}$$

Therefore, the suitcase on 10th floor has 10.3 kJ more energy compared to an identical suitcase on the lobby level.

Discussion Noting that 1 kWh = 3600 kJ, the energy transferred to the suitcase is $10.3/3600 = 0.0029$ kWh, which is very small.

Energy Transfer by Heat and Work

3-19C Energy can cross the boundaries of a closed system in two forms: heat and work.

3-20C The form of energy that crosses the boundary of a closed system because of a temperature difference is heat; all other forms are work.

3-21C An adiabatic process is a process during which there is no heat transfer. A system that does not exchange any heat with its surroundings is an adiabatic system.

3-22C Point functions depend on the state only whereas the path functions depend on the path followed during a process. Properties of substances are point functions, heat and work are path functions.

3-23C The caloric theory is based on the assumption that heat is a fluid-like substance called the "caloric" which is a massless, colorless, odorless substance. It was abandoned in the middle of the nineteenth century after it was shown that there is no such thing as the caloric.

3-24C (a) The car's radiator transfers heat from the hot engine cooling fluid to the cooler air. No work interaction occurs in the radiator.

(b) The hot engine transfers heat to cooling fluid and ambient air while delivering work to the transmission.

(c) The warm tires transfer heat to the cooler air and to some degree to the cooler road while no work is produced. No work is produced since there is no motion of the forces acting at the interface between the tire and road.

(d) There is minor amount of heat transfer between the tires and road. Presuming that the tires are hotter than the road, the heat transfer is from the tires to the road. There is no work exchange associated with the road since it cannot move.

(e) Heat is being added to the atmospheric air by the hotter components of the car. Work is being done on the air as it passes over and through the car.

3-25C When the length of the spring is changed by applying a force to it, the interaction is a work interaction since it involves a force acting through a displacement. A heat interaction is required to change the temperature (and, hence, length) of the spring.

3-26C (a) From the perspective of the contents, heat must be removed in order to reduce and maintain the content's temperature. Heat is also being added to the contents from the room air since the room air is hotter than the contents.

(b) Considering the system formed by the refrigerator box when the doors are closed, there are three interactions, electrical work and two heat transfers. There is a transfer of heat from the room air to the refrigerator through its walls. There is also a transfer of heat from the hot portions of the refrigerator (i.e., back of the compressor where condenser is placed) system to the room air. Finally, electrical work is being added to the refrigerator through the refrigeration system.

(c) Heat is transferred through the walls of the room from the warm room air to the cold winter air. Electrical work is being done on the room through the electrical wiring leading into the room.

3-27C (a) As one types on the keyboard, electrical signals are produced and transmitted to the processing unit. Simultaneously, the temperature of the electrical parts is increased slightly. The work done on the keys when they are depressed is work done on the system (i.e., keyboard). The flow of electrical current (with its voltage drop) does work on the keyboard. Since the temperature of the electrical parts of the keyboard is somewhat higher than that of the surrounding air, there is a transfer of heat from the keyboard to the surrounding air.

(b) The monitor is powered by the electrical current supplied to it. This current (and voltage drop) is work done on the system (i.e., monitor). The temperatures of the electrical parts of the monitor are higher than that of the surrounding air. Hence there is a heat transfer to the surroundings.

(c) The processing unit is like the monitor in that electrical work is done on it while it transfers heat to the surroundings.

(d) The entire unit then has electrical work done on it, and mechanical work done on it to depress the keys. It also transfers heat from all its electrical parts to the surroundings.

3-28 The power produced by an electrical motor is to be expressed in different units.

Analysis Using appropriate conversion factors, we obtain

$$(a) \quad \dot{W} = (10 \text{ W}) \left(\frac{1 \text{ J/s}}{1 \text{ W}} \right) \left(\frac{1 \text{ N} \cdot \text{m}}{1 \text{ J}} \right) = \mathbf{10 \text{ N} \cdot \text{m/s}}$$

$$(b) \quad \dot{W} = (10 \text{ W}) \left(\frac{1 \text{ J/s}}{1 \text{ W}} \right) \left(\frac{1 \text{ N} \cdot \text{m}}{1 \text{ J}} \right) \left(\frac{1 \text{ kg} \cdot \text{m/s}^2}{1 \text{ N}} \right) = \mathbf{10 \text{ kg} \cdot \text{m}^2/\text{s}^3}$$

3-29E The power produced by a model aircraft engine is to be expressed in different units.

Analysis Using appropriate conversion factors, we obtain

$$(a) \quad \dot{W} = (10 \text{ W}) \left(\frac{1 \text{ Btu/s}}{1055.056 \text{ W}} \right) \left(\frac{778.169 \text{ lbf} \cdot \text{ft/s}}{1 \text{ Btu/s}} \right) = \mathbf{7.38 \text{ lbf} \cdot \text{ft/s}}$$

$$(b) \quad \dot{W} = (10 \text{ W}) \left(\frac{1 \text{ hp}}{745.7 \text{ W}} \right) = \mathbf{0.0134 \text{ hp}}$$

Mechanical Forms of Work

3-30C The work done is the same, but the power is different.

3-31 A car is accelerated from rest to 100 km/h. The work needed to achieve this is to be determined.

Analysis The work needed to accelerate a body the change in kinetic energy of the body,

$$W_a = \frac{1}{2} m(V_2^2 - V_1^2) = \frac{1}{2} (800 \text{ kg}) \left(\left(\frac{100,000 \text{ m}}{3600 \text{ s}} \right)^2 - 0 \right) \left(\frac{1 \text{ kJ}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2} \right) = \mathbf{309 \text{ kJ}}$$

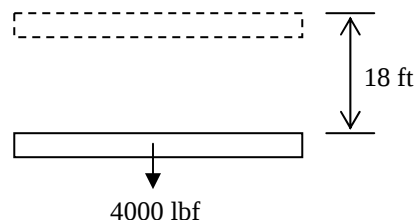
3-32E A construction crane lifting a concrete beam is considered. The amount of work is to be determined considering (a) the beam and (b) the crane as the system.

Analysis (a) The work is done on the beam and it is determined from

$$\begin{aligned} W &= mg\Delta z = (2 \times 2000 \text{ lbm})(32.174 \text{ ft/s}^2) \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm} \cdot \text{ft/s}^2} \right) (18 \text{ ft}) \\ &= \mathbf{72,000 \text{ lbf} \cdot \text{ft}} \\ &= (72,000 \text{ lbf} \cdot \text{ft}) \left(\frac{1 \text{ Btu}}{778.169 \text{ lbf} \cdot \text{ft}} \right) = \mathbf{92.5 \text{ Btu}} \end{aligned}$$

(b) Since the crane must produce the same amount of work as is required to lift the beam, the work done by the crane is

$$W = \mathbf{72,000 \text{ lbf} \cdot \text{ft} = 92.5 \text{ Btu}}$$



3-33 A man is pushing a cart with its contents up a ramp that is inclined at an angle of 20° from the horizontal. The work needed to move along this ramp is to be determined considering (a) the man and (b) the cart and its contents as the system.

Analysis (a) Considering the man as the system, letting l be the displacement along the ramp, and letting θ be the inclination angle of the ramp,

$$W = Fl \sin \theta = mgl \sin \theta = (100 + 100 \text{ kg})(9.8 \text{ m/s}^2)(100 \text{ m}) \sin(20^\circ) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{67.0 \text{ kJ}}$$

This is work that the man must do to raise the weight of the cart and contents, plus his own weight, a distance of $l \sin \theta$.

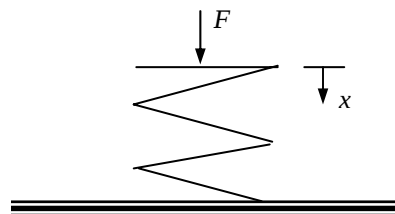
(b) Applying the same logic to the cart and its contents gives

$$W = Fl \sin \theta = mgl \sin \theta = (100 \text{ kg})(9.8 \text{ m/s}^2)(100 \text{ m}) \sin(20^\circ) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{33.5 \text{ kJ}}$$

3-34E The work required to compress a spring is to be determined.

Analysis Since there is no preload, $F = kx$. Substituting this into the work expression gives

$$\begin{aligned} W &= \int_1^2 F ds = \int_1^2 kx dx = k \int_1^2 x dx = \frac{k}{2} (x_2^2 - x_1^2) \\ &= \frac{200 \text{ lbf/in}}{2} [(1 \text{ in})^2 - 0^2] \left(\frac{1 \text{ ft}}{12 \text{ in}} \right) = \mathbf{8.33 \text{ lbf} \cdot \text{ft}} \\ &= (8.33 \text{ lbf} \cdot \text{ft}) \left(\frac{1 \text{ Btu}}{778.169 \text{ lbf} \cdot \text{ft}} \right) = \mathbf{0.0107 \text{ Btu}} \end{aligned}$$



3-35 As a spherical ammonia vapor bubble rises in liquid ammonia, its diameter increases. The amount of work produced by this bubble is to be determined.

Assumptions **1** The bubble is treated as a spherical bubble. **2** The surface tension coefficient is taken constant.

Analysis Executing the work integral for a constant surface tension coefficient gives

$$\begin{aligned} W &= \sigma \int_1^2 dA = \sigma (A_2 - A_1) = \sigma 4\pi (r_2^2 - r_1^2) \\ &= 4\pi (0.02 \text{ N/m}) [(0.015 \text{ m})^2 - (0.005 \text{ m})^2] \\ &= 5.03 \times 10^{-5} \text{ N} \cdot \text{m} \\ &= (5.03 \times 10^{-5} \text{ N} \cdot \text{m}) \left(\frac{1 \text{ kJ}}{1000 \text{ N} \cdot \text{m}} \right) = \mathbf{5.03 \times 10^{-8} \text{ kJ}} \end{aligned}$$

3-36 The work required to stretch a steel rod in a specified length is to be determined.

Assumptions The Young's modulus does not change as the rod is stretched.

Analysis The original volume of the rod is

$$V_0 = \frac{\pi D^2}{4} L = \frac{\pi (0.005 \text{ m})^2}{4} (10 \text{ m}) = 1.963 \times 10^{-4} \text{ m}^3$$

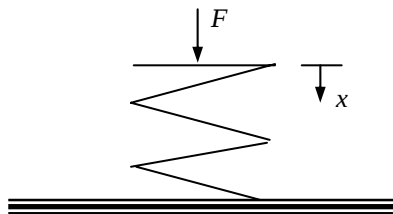
The work required to stretch the rod 3 cm is

$$\begin{aligned} W &= \frac{V_0 E}{2} (\varepsilon_2^2 - \varepsilon_1^2) \\ &= \frac{(1.963 \times 10^{-4} \text{ m}^3)(21 \times 10^4 \text{ kN/m}^2)}{2} [(0.03 \text{ m})^2 - 0^2] \\ &= 0.01855 \text{ kN} \cdot \text{m} = \mathbf{0.01855 \text{ kJ}} \end{aligned}$$

3-37E The work required to compress a spring is to be determined.

Analysis The force at any point during the deflection of the spring is given by $F = F_0 + kx$, where F_0 is the initial force and x is the deflection as measured from the point where the initial force occurred. From the perspective of the spring, this force acts in the direction opposite to that in which the spring is deflected. Then,

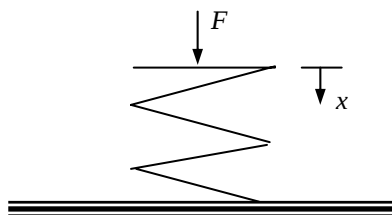
$$\begin{aligned}
 W &= \int_1^2 F ds = \int_1^2 (F_0 + kx) dx \\
 &= F_0(x_2 - x_1) + \frac{k}{2}(x_2^2 - x_1^2) \\
 &= (100 \text{ lbf})(1 - 0) \text{ in} + \frac{200 \text{ lbf/in}}{2}(1^2 - 0^2) \text{ in}^2 \\
 &= 200 \text{ lbf} \cdot \text{in} \\
 &= (200 \text{ lbf} \cdot \text{in}) \left(\frac{1 \text{ Btu}}{778.169 \text{ lbf} \cdot \text{ft}} \right) \left(\frac{1 \text{ ft}}{12 \text{ in}} \right) = \mathbf{0.0214 \text{ Btu}}
 \end{aligned}$$



3-38 The work required to compress a spring is to be determined.

Analysis Since there is no preload, $F = kx$. Substituting this into the work expression gives

$$\begin{aligned}
 W &= \int_1^2 F ds = \int_1^2 kx dx = k \int_1^2 x dx = \frac{k}{2}(x_2^2 - x_1^2) \\
 &= \frac{300 \text{ kN/m}}{2} [(0.03 \text{ m})^2 - 0^2] \\
 &= 0.135 \text{ kN} \cdot \text{m} \\
 &= (0.135 \text{ kN} \cdot \text{m}) \left(\frac{1 \text{ kJ}}{1 \text{ kN} \cdot \text{m}} \right) = \mathbf{0.135 \text{ kJ}}
 \end{aligned}$$



3-39 A ski lift is operating steadily at 10 km/h. The power required to operate and also to accelerate this ski lift from rest to the operating speed are to be determined.

Assumptions 1 Air drag and friction are negligible. 2 The average mass of each loaded chair is 250 kg. 3 The mass of chairs is small relative to the mass of people, and thus the contribution of returning empty chairs to the motion is disregarded (this provides a safety factor).

Analysis The lift is 1000 m long and the chairs are spaced 20 m apart. Thus at any given time there are $1000/20 = 50$ chairs being lifted. Considering that the mass of each chair is 250 kg, the load of the lift at any given time is

$$\text{Load} = (50 \text{ chairs})(250 \text{ kg/chair}) = 12,500 \text{ kg}$$

Neglecting the work done on the system by the returning empty chairs, the work needed to raise this mass by 200 m is

$$W_g = mg(z_2 - z_1) = (12,500 \text{ kg})(9.81 \text{ m/s}^2)(200 \text{ m}) \left(\frac{1 \text{ kJ}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2} \right) = 24,525 \text{ kJ}$$

At 10 km/h, it will take

$$\Delta t = \frac{\text{distance}}{\text{velocity}} = \frac{1 \text{ km}}{10 \text{ km/h}} = 0.1 \text{ h} = 360 \text{ s}$$

to do this work. Thus the power needed is

$$\dot{W}_g = \frac{W_g}{\Delta t} = \frac{24,525 \text{ kJ}}{360 \text{ s}} = \mathbf{68.1 \text{ kW}}$$

The velocity of the lift during steady operation, and the acceleration during start up are

$$V = (10 \text{ km/h}) \left(\frac{1 \text{ m/s}}{3.6 \text{ km/h}} \right) = 2.778 \text{ m/s}$$

$$a = \frac{\Delta V}{\Delta t} = \frac{2.778 \text{ m/s} - 0}{5 \text{ s}} = 0.556 \text{ m/s}^2$$

During acceleration, the power needed is

$$\dot{W}_a = \frac{1}{2} m(V_2^2 - V_1^2) / \Delta t = \frac{1}{2} (12,500 \text{ kg}) \left((2.778 \text{ m/s})^2 - 0 \right) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) / (5 \text{ s}) = 9.6 \text{ kW}$$

Assuming the power applied is constant, the acceleration will also be constant and the vertical distance traveled during acceleration will be

$$h = \frac{1}{2} at^2 \sin \alpha = \frac{1}{2} at^2 \frac{200 \text{ m}}{1000 \text{ m}} = \frac{1}{2} (0.556 \text{ m/s}^2)(5 \text{ s})^2 (0.2) = 1.39 \text{ m}$$

and

$$\dot{W}_g = mg(z_2 - z_1) / \Delta t = (12,500 \text{ kg})(9.81 \text{ m/s}^2)(1.39 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2} \right) / (5 \text{ s}) = 34.1 \text{ kW}$$

Thus,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 9.6 + 34.1 = \mathbf{43.7 \text{ kW}}$$

3-40 A car is to climb a hill in 10 s. The power needed is to be determined for three different cases.

Assumptions Air drag, friction, and rolling resistance are negligible.

Analysis The total power required for each case is the sum of the rates of changes in potential and kinetic energies. That is,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g$$

(a) $\dot{W}_a = 0$ since the velocity is constant. Also, the vertical rise is $h = (100 \text{ m})(\sin 30^\circ) = 50 \text{ m}$. Thus,

$$\dot{W}_g = mg(z_2 - z_1) / \Delta t = (2000 \text{ kg})(9.81 \text{ m/s}^2)(50 \text{ m}) \left(\frac{1 \text{ kJ}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2} \right) / (10 \text{ s}) = 98.1 \text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 0 + 98.1 = \mathbf{98.1 \text{ kW}}$

(b) The power needed to accelerate is

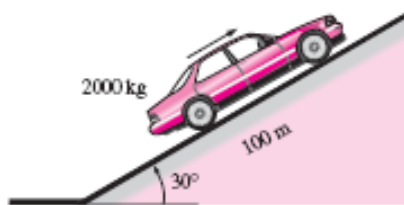
$$\dot{W}_a = \frac{1}{2} m(V_2^2 - V_1^2) / \Delta t = \frac{1}{2} (2000 \text{ kg}) \left[(30 \text{ m/s})^2 - 0 \right] \left(\frac{1 \text{ kJ}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2} \right) / (10 \text{ s}) = 90 \text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 90 + 98.1 = \mathbf{188.1 \text{ kW}}$

(c) The power needed to decelerate is

$$\dot{W}_a = \frac{1}{2} m(V_2^2 - V_1^2) / \Delta t = \frac{1}{2} (2000 \text{ kg}) \left[(5 \text{ m/s})^2 - (35 \text{ m/s})^2 \right] \left(\frac{1 \text{ kJ}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2} \right) / (10 \text{ s}) = -120 \text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = -120 + 98.1 = \mathbf{-21.9 \text{ kW}}$ (braking power)



3-41 A damaged car is being towed by a truck. The extra power needed is to be determined for three different cases.

Assumptions Air drag, friction, and rolling resistance are negligible.

Analysis The total power required for each case is the sum of the rates of changes in potential and kinetic energies. That is,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g$$

(a) Zero.

(b) $\dot{W}_a = 0$. Thus,

$$\begin{aligned}\dot{W}_{\text{total}} &= \dot{W}_g = mg(z_2 - z_1) / \Delta t = mg \frac{\Delta z}{\Delta t} = mgV_z = mgV \sin 30^\circ \\ &= (1200 \text{ kg})(9.81 \text{ m/s}^2) \left(\frac{50,000 \text{ m}}{3600 \text{ s}} \right) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) (0.5) = \mathbf{81.7 \text{ kW}}\end{aligned}$$



(c) $\dot{W}_g = 0$. Thus,

$$\dot{W}_{\text{total}} = \dot{W}_a = \frac{1}{2} m(V_2^2 - V_1^2) / \Delta t = \frac{1}{2} (1200 \text{ kg}) \left(\left(\frac{90,000 \text{ m}}{3600 \text{ s}} \right)^2 - 0 \right) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) / (12 \text{ s}) = \mathbf{31.3 \text{ kW}}$$

The First Law of Thermodynamics

3-42C No. This is the case for adiabatic systems only.

3-43C Warmer. Because energy is added to the room air in the form of electrical work.

3-44C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass transport.

3-45 The high rolling resistance tires of a car are replaced by low rolling resistance ones. For a specified unit fuel cost, the money saved by switching to low resistance tires is to be determined.

Assumptions 1 The low rolling resistance tires deliver 2 mpg over all velocities. 2 The car is driven 15,000 miles per year.

Analysis The annual amount of fuel consumed by this car on high- and low-rolling resistance tires are

$$\text{Annual Fuel Consumption}_{\text{High}} = \frac{\text{Miles driven per year}}{\text{Miles per gallon}} = \frac{15,000 \text{ miles/year}}{25 \text{ miles/gal}} = 600 \text{ gal/year}$$

$$\text{Annual Fuel Consumption}_{\text{Low}} = \frac{\text{Miles driven per year}}{\text{Miles per gallon}} = \frac{15,000 \text{ miles/year}}{27 \text{ miles/gal}} = 555.5 \text{ gal/year}$$

Then the fuel and money saved per year become

$$\begin{aligned} \text{Fuel Savings} &= \text{Annual Fuel Consumption}_{\text{High}} - \text{Annual Fuel Consumption}_{\text{Low}} \\ &= 600 \text{ gal/year} - 555.5 \text{ gal/year} = 44.5 \text{ gal/year} \end{aligned}$$

$$\text{Cost savings} = (\text{Fuel savings})(\text{Unit cost of fuel}) = (44.5 \text{ gal/year})(\$2.20/\text{gal}) = \mathbf{\$97.9/\text{year}}$$

Discussion A typical tire lasts about 3 years, and thus the low rolling resistance tires have the potential to save about \$300 to the car owner over the life of the tires, which is comparable to the installation cost of the tires.

3-46 The specific energy change of a system which is accelerated is to be determined.

Analysis Since the only property that changes for this system is the velocity, only the kinetic energy will change. The change in the specific energy is

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(30 \text{ m/s})^2 - (0 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{0.45 \text{ kJ/kg}}$$

3-47 The specific energy change of a system which is raised is to be determined.

Analysis Since the only property that changes for this system is the elevation, only the potential energy will change. The change in the specific energy is then

$$\Delta pe = g(z_2 - z_1) = (9.8 \text{ m/s}^2)(100 - 0) \text{ m} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{0.98 \text{ kJ/kg}}$$

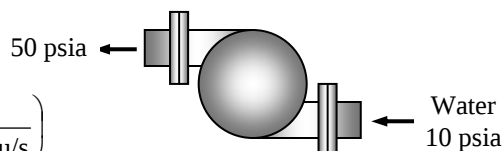
3-48E A water pump increases water pressure. The power input is to be determined.

Analysis The power input is determined from

$$\dot{W} = \dot{V}(P_2 - P_1)$$

$$= (1.2 \text{ ft}^3/\text{s})(50 - 10) \text{ psia} \left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3} \right) \left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}} \right)$$

$$= \mathbf{12.6 \text{ hp}}$$



The water temperature at the inlet does not have any significant effect on the required power.

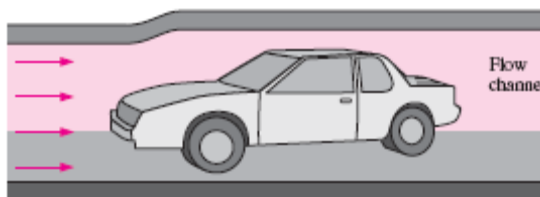
3-49 An automobile moving at a given velocity is considered. The power required to move the car and the area of the effective flow channel behind the car are to be determined.

Analysis The absolute pressure of the air is

$$P = (750 \text{ mm Hg}) \left(\frac{0.1333 \text{ kPa}}{1 \text{ mm Hg}} \right) = 99.98 \text{ kPa}$$

and the specific volume of the air is

$$\nu = \frac{RT}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(303 \text{ K})}{99.98 \text{ kPa}} = 0.8698 \text{ m}^3/\text{kg}$$



The mass flow rate through the control volume is

$$\dot{m} = \frac{A_1 V_1}{\nu} = \frac{(3 \text{ m}^2)(90/3.6 \text{ m/s})}{0.8698 \text{ m}^3/\text{kg}} = 86.23 \text{ kg/s}$$

The power requirement is

$$\dot{W} = \dot{m} \frac{V_1^2 - V_2^2}{2} = (86.23 \text{ kg/s}) \frac{(90/3.6 \text{ m/s})^2 - (82/3.6 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{4.578 \text{ kW}}$$

The outlet area is

$$\dot{m} = \frac{A_2 V_2}{\nu} \longrightarrow A_2 = \frac{\dot{m} \nu}{V_2} = \frac{(86.23 \text{ kg/s})(0.8698 \text{ m}^3/\text{kg})}{(82/3.6 \text{ m/s})} = \mathbf{3.29 \text{ m}^2}$$

3-50 A classroom is to be air-conditioned using window air-conditioning units. The cooling load is due to people, lights, and heat transfer through the walls and the windows. The number of 5-kW window air conditioning units required is to be determined.

Assumptions There are no heat dissipating equipment (such as computers, TVs, or ranges) in the room.

Analysis The total cooling load of the room is determined from

$$\dot{Q}_{\text{cooling}} = \dot{Q}_{\text{lights}} + \dot{Q}_{\text{people}} + \dot{Q}_{\text{heat gain}}$$

where

$$\dot{Q}_{\text{lights}} = 10 \times 100 \text{ W} = 1 \text{ kW}$$

$$\dot{Q}_{\text{people}} = 40 \times 360 \text{ kJ/h} = 4 \text{ kW}$$

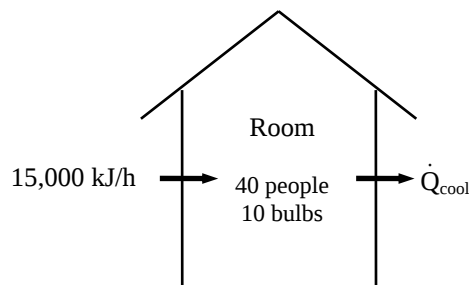
$$\dot{Q}_{\text{heat gain}} = 15,000 \text{ kJ/h} = 4.17 \text{ kW}$$

Substituting,

$$\dot{Q}_{\text{cooling}} = 1 + 4 + 4.17 = 9.17 \text{ kW}$$

Thus the number of air-conditioning units required is

$$\frac{9.17 \text{ kW}}{5 \text{ kW/unit}} = 1.83 \longrightarrow \mathbf{2 \text{ units}}$$



3-51 The lighting energy consumption of a storage room is to be reduced by installing motion sensors. The amount of energy and money that will be saved as well as the simple payback period are to be determined.

Assumptions The electrical energy consumed by the ballasts is negligible.

Analysis The plant operates 12 hours a day, and thus currently the lights are on for the entire 12 hour period. The motion sensors installed will keep the lights on for 3 hours, and off for the remaining 9 hours every day. This corresponds to a total of $9 \times 365 = 3285$ off hours per year. Disregarding the ballast factor, the annual energy and cost savings become

Energy Savings = (Number of lamps)(Lamp wattage)(Reduction of annual operating hours)

$$= (24 \text{ lamps})(60 \text{ W/lamp})(3285 \text{ hours/year})$$

$$= \mathbf{4730 \text{ kWh/year}}$$

Cost Savings = (Energy Savings)(Unit cost of energy)

$$= (4730 \text{ kWh/year})(\$0.08/\text{kWh})$$

$$= \mathbf{\$378/\text{year}}$$

The implementation cost of this measure is the sum of the purchase price of the sensor plus the labor,

$$\text{Implementation Cost} = \text{Material} + \text{Labor} = \$32 + \$40 = \$72$$

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$72}{\$378/\text{year}} = \mathbf{0.19 \text{ year}} \text{ (2.3 months)}$$

Therefore, the motion sensor will pay for itself in about 2 months.



3-52 The classrooms and faculty offices of a university campus are not occupied an average of 4 hours a day, but the lights are kept on. The amounts of electricity and money the campus will save per year if the lights are turned off during unoccupied periods are to be determined.

Analysis The total electric power consumed by the lights in the classrooms and faculty offices is

$$\dot{E}_{\text{lighting, classroom}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (200 \times 12 \times 110 \text{ W}) = 264,000 = 264 \text{ kW}$$

$$\dot{E}_{\text{lighting, offices}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (400 \times 6 \times 110 \text{ W}) = 264,000 = 264 \text{ kW}$$

$$\dot{E}_{\text{lighting, total}} = \dot{E}_{\text{lighting, classroom}} + \dot{E}_{\text{lighting, offices}} = 264 + 264 = 528 \text{ kW}$$

Noting that the campus is open 240 days a year, the total number of unoccupied work hours per year is

$$\text{Unoccupied hours} = (4 \text{ hours/day})(240 \text{ days/year}) = 960 \text{ h/yr}$$

Then the amount of electrical energy consumed per year during unoccupied work period and its cost are

$$\text{Energy savings} = (\dot{E}_{\text{lighting, total}})(\text{Unoccupied hours}) = (528 \text{ kW})(960 \text{ h/yr}) = 506,880 \text{ kWh}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy}) = (506,880 \text{ kWh/yr})(\$0.082/\text{kWh}) = \mathbf{\$41,564/\text{yr}}$$

Discussion Note that simple conservation measures can result in significant energy and cost savings.

3-53 A room contains a light bulb, a TV set, a refrigerator, and an iron. The rate of increase of the energy content of the room when all of these electric devices are on is to be determined.

Assumptions **1** The room is well sealed, and heat loss from the room is negligible. **2** All the appliances are kept on.

Analysis Taking the room as the system, the rate form of the energy balance can be written as

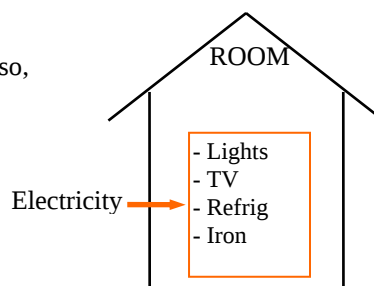
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\frac{dE_{\text{system}}}{dt}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \rightarrow \frac{dE_{\text{room}}}{dt} = \dot{E}_{\text{in}}$$

since no energy is leaving the room in any form, and thus $\dot{E}_{\text{out}} = 0$. Also,

$$\begin{aligned} \dot{E}_{\text{in}} &= \dot{E}_{\text{lights}} + \dot{E}_{\text{TV}} + \dot{E}_{\text{refrig}} + \dot{E}_{\text{iron}} \\ &= 100 + 110 + 200 + 1000 \text{ W} \\ &= 1410 \text{ W} \end{aligned}$$

Substituting, the rate of increase in the energy content of the room becomes

$$\frac{dE_{\text{room}}}{dt} = \dot{E}_{\text{in}} = \mathbf{1410 \text{ W}}$$



Discussion Note that some appliances such as refrigerators and irons operate intermittently, switching on and off as controlled by a thermostat. Therefore, the rate of energy transfer to the room, in general, will be less.

3-54 A fan is to accelerate quiescent air to a specified velocity at a specified flow rate. The minimum power that must be supplied to the fan is to be determined.

Assumptions The fan operates steadily.

Properties The density of air is given to be $\rho = 1.18 \text{ kg/m}^3$.

Analysis A fan transmits the mechanical energy of the shaft (shaft power) to mechanical energy of air (kinetic energy). For a control volume that encloses the fan, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}} / dt}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{e^0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

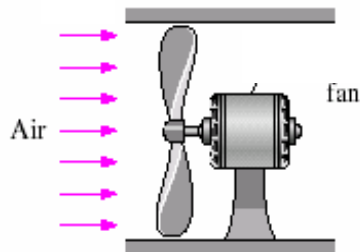
$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$

where

$$\dot{m}_{\text{air}} = \rho \dot{V} = (1.18 \text{ kg/m}^3)(4 \text{ m}^3/\text{s}) = 4.72 \text{ kg/s}$$

Substituting, the minimum power input required is determined

$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (4.72 \text{ kg/s}) \frac{(10 \text{ m/s})^2}{2} \left(\frac{1 \text{ J/kg}}{1 \text{ m}^2/\text{s}^2} \right) = 236 \text{ J/s} = \mathbf{236 \text{ W}}$$



Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher because of the losses associated with the conversion of mechanical shaft energy to kinetic energy of air.

3-55E A fan accelerates air to a specified velocity in a square duct. The minimum electric power that must be supplied to the fan motor is to be determined.

Assumptions 1 The fan operates steadily. 2 There are no conversion losses.

Properties The density of air is given to be $\rho = 0.075 \text{ lbm/ft}^3$.

Analysis A fan motor converts electrical energy to mechanical shaft energy, and the fan transmits the mechanical energy of the shaft (shaft power) to mechanical energy of air (kinetic energy). For a control volume that encloses the fan-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}} / dt}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{p0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{elect, in}} = \dot{m}_{\text{air}} \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$

where

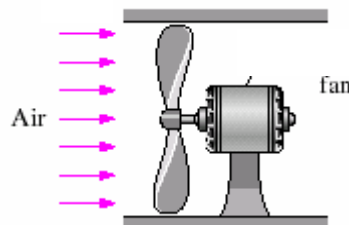
$$\dot{m}_{\text{air}} = \rho VA = (0.075 \text{ lbm/ft}^3)(3 \times 3 \text{ ft}^2)(22 \text{ ft/s}) = 14.85 \text{ lbm/s}$$

Substituting, the minimum power input required is determined to be

$$\dot{W}_{in} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (14.85 \text{ lbm/s}) \frac{(22 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = 0.1435 \text{ Btu/s} = \mathbf{151 \text{ W}}$$

since $1 \text{ Btu} = 1.055 \text{ kJ}$ and $1 \text{ kJ/s} = 1000 \text{ W}$.

Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-kinetic energy of air.



3-56 CD EES A gasoline pump raises the pressure to a specified value while consuming electric power at a specified rate. The maximum volume flow rate of gasoline is to be determined.

Assumptions 1 The gasoline pump operates steadily. 2 The changes in kinetic and potential energies across the pump are negligible.

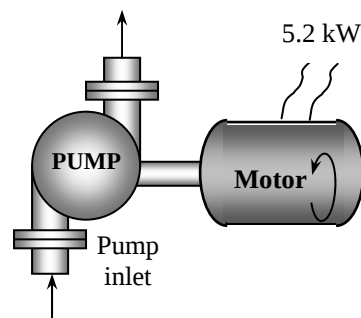
Analysis For a control volume that encloses the pump-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}}/dt}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\approx 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}(P\nu)_1 = \dot{m}(P\nu)_2 \rightarrow \dot{W}_{\text{in}} = \dot{m}(P_2 - P_1)\nu = \dot{\nu} \Delta P$$

since $\dot{m} = \dot{\nu}/\nu$ and the changes in kinetic and potential energies of gasoline are negligible, Solving for volume flow rate and substituting, the maximum flow rate is determined to be

$$\dot{\nu}_{\text{max}} = \frac{\dot{W}_{\text{in}}}{\Delta P} = \frac{5.2 \text{ kJ/s}}{5 \text{ kPa}} \left(\frac{1 \text{ kPa} \cdot \text{m}^3}{1 \text{ kJ}} \right) = \mathbf{1.04 \text{ m}^3/\text{s}}$$



Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the volume flow rate will be less because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-flow energy.

3-57 An inclined escalator is to move a certain number of people upstairs at a constant velocity. The minimum power required to drive this escalator is to be determined.

Assumptions **1** Air drag and friction are negligible. **2** The average mass of each person is 75 kg. **3** The escalator operates steadily, with no acceleration or breaking. **4** The mass of escalator itself is negligible.

Analysis At design conditions, the total mass moved by the escalator at any given time is

$$\text{Mass} = (30 \text{ persons})(75 \text{ kg/person}) = 2250 \text{ kg}$$

The vertical component of escalator velocity is

$$V_{\text{vert}} = V \sin 45^\circ = (0.8 \text{ m/s}) \sin 45^\circ$$

Under stated assumptions, the power supplied is used to increase the potential energy of people. Taking the people on elevator as the closed system, the energy balance in the rate form can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}}/dt}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = dE_{\text{sys}}/dt \cong \frac{\Delta E_{\text{sys}}}{\Delta t}$$

$$\dot{W}_{\text{in}} = \frac{\Delta PE}{\Delta t} = \frac{mg\Delta z}{\Delta t} = mgV_{\text{vert}}$$

That is, under stated assumptions, the power input to the escalator must be equal to the rate of increase of the potential energy of people. Substituting, the required power input becomes

$$\dot{W}_{\text{in}} = mgV_{\text{vert}} = (2250 \text{ kg})(9.81 \text{ m/s}^2)(0.8 \text{ m/s}) \sin 45^\circ \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 12.5 \text{ kJ/s} = \mathbf{12.5 \text{ kW}}$$

When the escalator velocity is doubled to $V = 1.6 \text{ m/s}$, the power needed to drive the escalator becomes

$$\dot{W}_{\text{in}} = mgV_{\text{vert}} = (2250 \text{ kg})(9.81 \text{ m/s}^2)(1.6 \text{ m/s}) \sin 45^\circ \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 25.0 \text{ kJ/s} = \mathbf{25.0 \text{ kW}}$$

Discussion Note that the power needed to drive an escalator is proportional to the escalator velocity.

Energy Conversion Efficiencies

3-58C *Mechanical efficiency* is defined as the ratio of the mechanical energy output to the mechanical energy input. A mechanical efficiency of 100% for a hydraulic turbine means that the entire mechanical energy of the fluid is converted to mechanical (shaft) work.

3-59C The *combined pump-motor efficiency* of a pump/motor system is defined as the ratio of the increase in the mechanical energy of the fluid to the electrical power consumption of the motor,

$$\eta_{\text{pump-motor}} = \eta_{\text{pump}} \eta_{\text{motor}} = \frac{\dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}}}{\dot{W}_{\text{elect,in}}} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{\dot{W}_{\text{pump}}}{\dot{W}_{\text{elect,in}}}$$

The combined pump-motor efficiency cannot be greater than either of the pump or motor efficiency since both pump and motor efficiencies are less than 1, and the product of two numbers that are less than one is less than either of the numbers.

3-60C The turbine efficiency, generator efficiency, and *combined turbine-generator efficiency* are defined as follows:

$$\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy extracted from the fluid}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|}$$

$$\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}$$

$$\eta_{\text{turbine-gen}} = \eta_{\text{turbine}} \eta_{\text{generator}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{E}_{\text{mech,in}} - \dot{E}_{\text{mech,out}}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|}$$

3-61C No, the combined pump-motor efficiency cannot be greater than either of the pump efficiency or the motor efficiency. This is because $\eta_{\text{pump-motor}} = \eta_{\text{pump}} \eta_{\text{motor}}$, and both η_{pump} and η_{motor} are less than one, and a number gets smaller when multiplied by a number smaller than one.

3-62 A hooded electric open burner and a gas burner are considered. The amount of the electrical energy used directly for cooking and the cost of energy per “utilized” kWh are to be determined.

Analysis The efficiency of the electric heater is given to be 73 percent. Therefore, a burner that consumes 3-kW of electrical energy will supply

$$\eta_{\text{gas}} = 38\%$$

$$\eta_{\text{electric}} = 73\%$$

$$\dot{Q}_{\text{utilized}} = (\text{Energy input}) \times (\text{Efficiency}) = (3 \text{ kW})(0.73) = \mathbf{2.19 \text{ kW}}$$

of useful energy. The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$0.07/\text{kWh}}{0.73} = \mathbf{\$0.096/\text{kWh}}$$

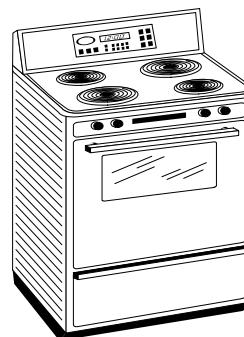
Noting that the efficiency of a gas burner is 38 percent, the energy input to a gas burner that supplies utilized energy at the same rate (2.19 kW) is

$$\dot{Q}_{\text{input, gas}} = \frac{\dot{Q}_{\text{utilized}}}{\text{Efficiency}} = \frac{2.19 \text{ kW}}{0.38} = \mathbf{5.76 \text{ kW}} \quad (= 19,660 \text{ Btu/h})$$

since 1 kW = 3412 Btu/h. Therefore, a gas burner should have a rating of at least 19,660 Btu/h to perform as well as the electric unit.

Noting that 1 therm = 29.3 kWh, the unit cost of utilized energy in the case of gas burner is determined the same way to be

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$1.20/(29.3 \text{ kWh})}{0.38} = \mathbf{\$0.108/\text{kWh}}$$



3-63 A worn out standard motor is replaced by a high efficiency one. The reduction in the internal heat gain due to the higher efficiency under full load conditions is to be determined.

Assumptions 1 The motor and the equipment driven by the motor are in the same room. **2** The motor operates at full load so that $f_{\text{load}} = 1$.

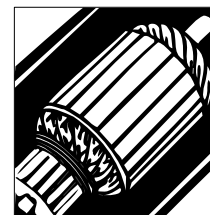
Analysis The heat generated by a motor is due to its inefficiency, and the difference between the heat generated by two motors that deliver the same shaft power is simply the difference between the electric power drawn by the motors,

$$\dot{W}_{\text{in, electric, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746 \text{ W}) / 0.91 = 61,484 \text{ W}$$

$$\dot{W}_{\text{in, electric, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746 \text{ W}) / 0.954 = 58,648 \text{ W}$$

Then the reduction in heat generation becomes

$$\dot{Q}_{\text{reduction}} = \dot{W}_{\text{in, electric, standard}} - \dot{W}_{\text{in, electric, efficient}} = 61,484 - 58,648 = \mathbf{2836 \text{ W}}$$



3-64 An electric car is powered by an electric motor mounted in the engine compartment. The rate of heat supply by the motor to the engine compartment at full load conditions is to be determined.

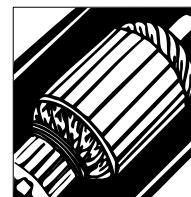
Assumptions The motor operates at full load so that the load factor is 1.

Analysis The heat generated by a motor is due to its inefficiency, and is equal to the difference between the electrical energy it consumes and the shaft power it delivers,

$$\begin{aligned}\dot{W}_{\text{in, electric}} &= \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (90 \text{ hp}) / 0.91 = 98.90 \text{ hp} \\ \dot{Q}_{\text{generation}} &= \dot{W}_{\text{in, electric}} - \dot{W}_{\text{shaft out}} = 98.90 - 90 = 8.90 \text{ hp} = \mathbf{6.64 \text{ kW}}\end{aligned}$$

since 1 hp = 0.746 kW.

Discussion Note that the electrical energy not converted to mechanical power is converted to heat.



3-65 A worn out standard motor is to be replaced by a high efficiency one. The amount of electrical energy and money savings as a result of installing the high efficiency motor instead of the standard one as well as the simple payback period are to be determined.

Assumptions The load factor of the motor remains constant at 0.75.

Analysis The electric power drawn by each motor and their difference can be expressed as

$$\begin{aligned}\dot{W}_{\text{electric in, standard}} &= \dot{W}_{\text{shaft}} / \eta_{\text{standard}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{standard}} \\ \dot{W}_{\text{electric in, efficient}} &= \dot{W}_{\text{shaft}} / \eta_{\text{efficient}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{efficient}} \\ \text{Power savings} &= \dot{W}_{\text{electric in, standard}} - \dot{W}_{\text{electric in, efficient}} \\ &= (\text{Power rating})(\text{Load factor})[1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}}]\end{aligned}$$

where η_{standard} is the efficiency of the standard motor, and $\eta_{\text{efficient}}$ is the efficiency of the comparable high efficiency motor. Then the annual energy and cost savings associated with the installation of the high efficiency motor are determined to be

$$\begin{aligned}\text{Energy Savings} &= (\text{Power savings})(\text{Operating Hours}) \\ &= (\text{Power Rating})(\text{Operating Hours})(\text{Load Factor})(1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}}) \\ &= (75 \text{ hp})(0.746 \text{ kW/hp})(4,368 \text{ hours/year})(0.75)(1/0.91 - 1/0.954) \\ &= \mathbf{9,290 \text{ kWh/year}}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= (\text{Energy savings})(\text{Unit cost of energy}) \\ &= (9,290 \text{ kWh/year})(\$0.08/\text{kWh}) \\ &= \mathbf{\$743/\text{year}}\end{aligned}$$

The implementation cost of this measure consists of the excess cost the high efficiency motor over the standard one. That is,

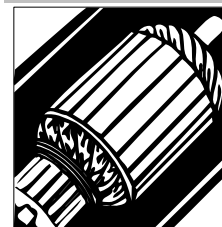
$$\text{Implementation Cost} = \text{Cost differential} = \$5,520 - \$5,449 = \$71$$

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$71}{\$743/\text{year}} = \mathbf{0.096 \text{ year}} \text{ (or 1.1 months)}$$

Therefore, the high-efficiency motor will pay for its cost differential in about one month.

$$\begin{aligned}\eta_{\text{old}} &= 91.0\% \\ \eta_{\text{new}} &= 95.4\%\end{aligned}$$



3-66E The combustion efficiency of a furnace is raised from 0.7 to 0.8 by tuning it up. The annual energy and cost savings as a result of tuning up the boiler are to be determined.

Assumptions The boiler operates at full load while operating.

Analysis The heat output of boiler is related to the fuel energy input to the boiler by

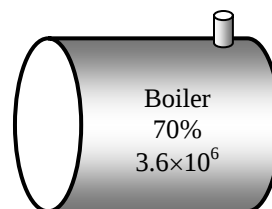
$$\text{Boiler output} = (\text{Boiler input})(\text{Combustion efficiency}) \quad \text{or} \quad \dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} \eta_{\text{furnace}}$$

The current rate of heat input to the boiler is given to be $\dot{Q}_{\text{in, current}} = 3.6 \times 10^6 \text{ Btu/h}$.

Then the rate of useful heat output of the boiler becomes

$$\dot{Q}_{\text{out}} = (\dot{Q}_{\text{in}} \eta_{\text{furnace}})_{\text{current}} = (3.6 \times 10^6 \text{ Btu/h})(0.7) = 2.52 \times 10^6 \text{ Btu/h}$$

The boiler must supply useful heat at the same rate after the tune up. Therefore, the rate of heat input to the boiler after the tune up and the rate of energy savings become



$$\dot{Q}_{\text{in, new}} = \dot{Q}_{\text{out}} / \eta_{\text{furnace, new}} = (2.52 \times 10^6 \text{ Btu/h}) / 0.8 = 3.15 \times 10^6 \text{ Btu/h}$$

$$\dot{Q}_{\text{in, saved}} = \dot{Q}_{\text{in, current}} - \dot{Q}_{\text{in, new}} = 3.6 \times 10^6 - 3.15 \times 10^6 = 0.45 \times 10^6 \text{ Btu/h}$$

Then the annual energy and cost savings associated with tuning up the boiler become

$$\text{Energy Savings} = \dot{Q}_{\text{in, saved}} (\text{Operation hours})$$

$$= (0.45 \times 10^6 \text{ Btu/h})(1500 \text{ h/year}) = \mathbf{675 \times 10^6 \text{ Btu/yr}}$$

$$\text{Cost Savings} = (\text{Energy Savings})(\text{Unit cost of energy})$$

$$= (675 \times 10^6 \text{ Btu/yr})(\$4.35 \text{ per } 10^6 \text{ Btu}) = \mathbf{\$2936/\text{year}}$$

Discussion Notice that tuning up the boiler will save \$2936 a year, which is a significant amount. The implementation cost of this measure is negligible if the adjustment can be made by in-house personnel. Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the boiler twice a year.

3-67E EES Problem 3-66E is reconsidered. The effects of the unit cost of energy and combustion efficiency on the annual energy used and the cost savings as the efficiency varies from 0.6 to 0.9 and the unit cost varies from \$4 to \$6 per million Btu are the investigated. The annual energy saved and the cost savings are to be plotted against the efficiency for unit costs of \$4, \$5, and \$6 per million Btu.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns:"

eta_boiler_current = 0.7

eta_boiler_new = 0.8

Q_dot_in_current = 3.6E+6 "[Btu/h]"

DELTA_t = 1500 "[h/year]"

UnitCost_energy = 5E-6 "[dollars/Btu]"

"Analysis: The heat output of boiler is related to the fuel energy input to the boiler by

Boiler output = (Boiler input)(Combustion efficiency)

Then the rate of useful heat output of the boiler becomes"

Q_dot_out=Q_dot_in_current*eta_boiler_current "[Btu/h]"

"The boiler must supply useful heat at the same rate after the tune up.

Therefore, the rate of heat input to the boiler after the tune up

and the rate of energy savings become "

Q_dot_in_new=Q_dot_out/eta_boiler_new "[Btu/h]"

Q_dot_in_saved=Q_dot_in_current - Q_dot_in_new "[Btu/h]"

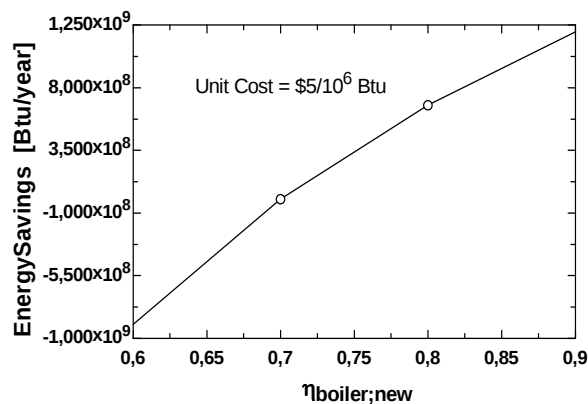
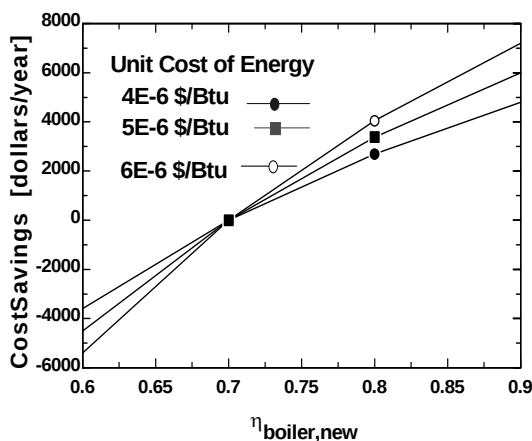
"Then the annual energy and cost savings associated with tuning up the boiler become"

EnergySavings =Q_dot_in_saved*DELTA_t "[Btu/year]"

CostSavings = EnergySavings*UnitCost_energy "[dollars/year]"

"Discussion Notice that tuning up the boiler will save \$2936 a year, which is a significant amount. The implementation cost of this measure is negligible if the adjustment can be made by in-house personnel. Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the boiler twice a year. "

CostSavings [dollars/year]	EnergySavings [Btu/year]	$\eta_{\text{boiler,new}}$
-4500	-9.000E+08	0.6
0	0	0.7
3375	6.750E+08	0.8
6000	1.200E+09	0.9



3-68 Several people are working out in an exercise room. The rate of heat gain from people and the equipment is to be determined.

Assumptions The average rate of heat dissipated by people in an exercise room is 525 W.

Analysis The 8 weight lifting machines do not have any motors, and thus they do not contribute to the internal heat gain directly. The usage factors of the motors of the treadmills are taken to be unity since they are used constantly during peak periods. Noting that 1 hp = 746 W, the total heat generated by the motors is

$$\begin{aligned}\dot{Q}_{\text{motors}} &= (\text{No. of motors}) \times \dot{W}_{\text{motor}} \times f_{\text{load}} \times f_{\text{usage}} / \eta_{\text{motor}} \\ &= 4 \times (2.5 \times 746 \text{ W}) \times 0.70 \times 1.0 / 0.77 = 6782 \text{ W}\end{aligned}$$

The heat gain from 14 people is

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 14 \times (525 \text{ W}) = 7350 \text{ W}$$

Then the total rate of heat gain of the exercise room during peak period becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{motors}} + \dot{Q}_{\text{people}} = 6782 + 7350 = \mathbf{14,132 \text{ W}}$$



3-69 A classroom has a specified number of students, instructors, and fluorescent light bulbs. The rate of internal heat generation in this classroom is to be determined.

Assumptions 1 There is a mix of men, women, and children in the classroom. 2 The amount of light (and thus energy) leaving the room through the windows is negligible.

Properties The average rate of heat generation from people seated in a room/office is given to be 100 W.

Analysis The amount of heat dissipated by the lamps is equal to the amount of electrical energy consumed by the lamps, including the 10% additional electricity consumed by the ballasts. Therefore,

$$\begin{aligned}\dot{Q}_{\text{lighting}} &= (\text{Energy consumed per lamp}) \times (\text{No. of lamps}) \\ &= (40 \text{ W})(1.1)(18) = 792 \text{ W}\end{aligned}$$

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 56 \times (100 \text{ W}) = 5600 \text{ W}$$

Then the total rate of heat gain (or the internal heat load) of the classroom from the lights and people become

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{lighting}} + \dot{Q}_{\text{people}} = 792 + 5600 = \mathbf{6392 \text{ W}}$$



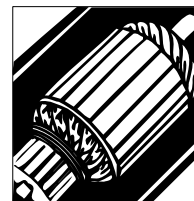
3-70 A room is cooled by circulating chilled water through a heat exchanger, and the air is circulated through the heat exchanger by a fan. The contribution of the fan-motor assembly to the cooling load of the room is to be determined.

Assumptions The fan motor operates at full load so that $f_{\text{load}} = 1$.

Analysis The entire electrical energy consumed by the motor, including the shaft power delivered to the fan, is eventually dissipated as heat. Therefore, the contribution of the fan-motor assembly to the cooling load of the room is equal to the electrical energy it consumes,

$$\begin{aligned}\dot{Q}_{\text{internal generation}} &= \dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} \\ &= (0.25 \text{ hp}) / 0.54 = 0.463 \text{ hp} = \mathbf{345 \text{ W}}\end{aligned}$$

since $1 \text{ hp} = 746 \text{ W}$.



3-71 A hydraulic turbine-generator is generating electricity from the water of a large reservoir. The combined turbine-generator efficiency and the turbine efficiency are to be determined.

Assumptions 1 The elevation of the reservoir remains constant. 2 The mechanical energy of water at the turbine exit is negligible.

Analysis We take the free surface of the reservoir to be point 1 and the turbine exit to be point 2. We also take the turbine exit as the reference level ($z_2 = 0$), and thus the potential energy at points 1 and 2 are $pe_1 = gz_1$ and $pe_2 = 0$. The flow energy P/ρ at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$). Further, the kinetic energy at both points is zero ($ke_1 = ke_2 = 0$) since the water at point 1 is essentially motionless, and the kinetic energy of water at turbine exit is assumed to be negligible. The potential energy of water at point 1 is

$$pe_1 = gz_1 = (9.81 \text{ m/s}^2)(70 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.687 \text{ kJ/kg}$$

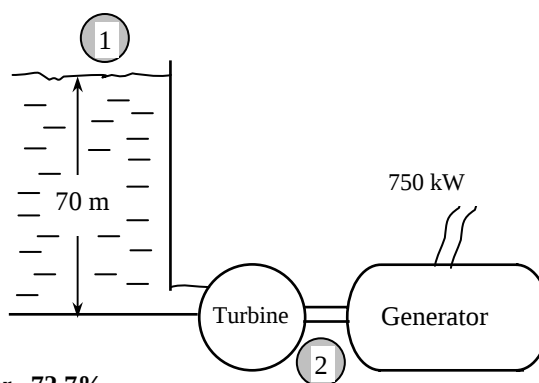
Then the rate at which the mechanical energy of the fluid is supplied to the turbine become

$$\begin{aligned} |\Delta \dot{E}_{\text{mech, fluid}}| &= \dot{m}(e_{\text{mech, in}} - e_{\text{mech, out}}) = \dot{m}(pe_1 - 0) \\ &= \dot{m}pe_1 \\ &= (1500 \text{ kg/s})(0.687 \text{ kJ/kg}) \\ &= 1031 \text{ kW} \end{aligned}$$

The combined turbine-generator and the turbine efficiency are determined from their definitions,

$$\eta_{\text{turbine-gen}} = \frac{\dot{W}_{\text{elect, out}}}{|\Delta \dot{E}_{\text{mech, fluid}}|} = \frac{750 \text{ kW}}{1031 \text{ kW}} = 0.727 \quad \text{or} \quad 72.7\%$$

$$\eta_{\text{turbine}} = \frac{\dot{W}_{\text{shaft, out}}}{|\Delta \dot{E}_{\text{mech, fluid}}|} = \frac{800 \text{ kW}}{1031 \text{ kW}} = 0.776 \quad \text{or} \quad 77.6\%$$



Therefore, the reservoir supplies 1031 kW of mechanical energy to the turbine, which converts 800 kW of it to shaft work that drives the generator, which generates 750 kW of electric power.

Discussion This problem can also be solved by taking point 1 to be at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

3-72 Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass, the power generation potential, and the actual electric power generation are to be determined.

Assumptions 1 The wind is blowing steadily at a constant uniform velocity. 2 The efficiency of the wind turbine is independent of the wind speed.

Properties The density of air is given to be $\rho = 1.25 \text{ kg/m}^3$.

Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(12 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.072 \text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V \frac{\pi D^2}{4} = (1.25 \text{ kg/m}^3)(12 \text{ m/s}) \frac{\pi (50 \text{ m})^2}{4} = 29,450 \text{ kg/s}$$

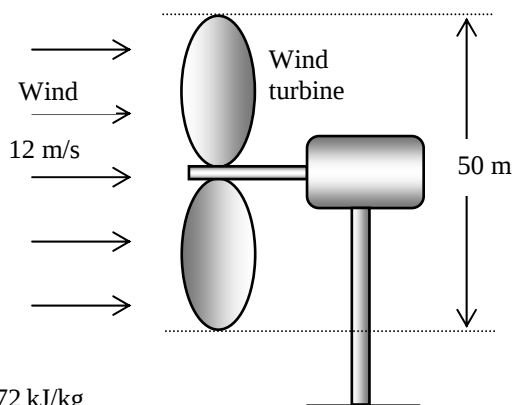
$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (29,450 \text{ kg/s})(0.072 \text{ kJ/kg}) = \mathbf{2121 \text{ kW}}$$

The actual electric power generation is determined by multiplying the power generation potential by the efficiency,

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}} \dot{W}_{\text{max}} = (0.30)(2121 \text{ kW}) = \mathbf{636 \text{ kW}}$$

Therefore, 636 kW of actual power can be generated by this wind turbine at the stated conditions.

Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.



3-73 EES Problem 3-72 is reconsidered. The effect of wind velocity and the blade span diameter on wind power generation as the velocity varies from 5 m/s to 20 m/s in increments of 5 m/s, and the diameter varies from 20 m to 80 m in increments of 20 m is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

D1=20 [m]

D2=40 [m]

D3=60 [m]

D4=80 [m]

Eta=0.30

rho=1.25 [kg/m^3]

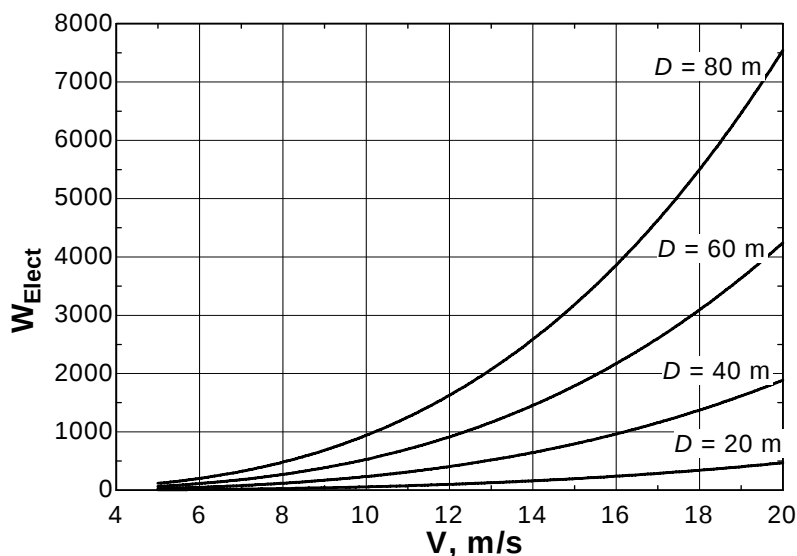
m1_dot=rho*V*(pi*D1^2/4); W1_Elect=Eta*m1_dot*(V^2/2)/1000 "kW"

m2_dot=rho*V*(pi*D2^2/4); W2_Elect=Eta*m2_dot*(V^2/2)/1000 "kW"

m3_dot=rho*V*(pi*D3^2/4); W3_Elect=Eta*m3_dot*(V^2/2)/1000 "kW"

m4_dot=rho*V*(pi*D4^2/4); W4_Elect=Eta*m4_dot*(V^2/2)/1000 "kW"

D, m	V, m/s	m, kg/s	W _{elect} , kW
20	5	1,963	7
	10	3,927	59
	15	5,890	199
	20	7,854	471
40	5	7,854	29
	10	15,708	236
	15	23,562	795
	20	31,416	1885
60	5	17,671	66
	10	35,343	530
	15	53,014	1789
	20	70,686	4241
80	5	31,416	118
	10	62,832	942
	15	94,248	3181
	20	125,664	7540



3-74 A wind turbine produces 180 kW of power. The average velocity of the air and the conversion efficiency of the turbine are to be determined.

Assumptions The wind turbine operates steadily.

Properties The density of air is given to be 1.31 kg/m^3 .

Analysis (a) The blade diameter and the blade span area are

$$D = \frac{V_{\text{tip}}}{\pi \dot{n}} = \frac{(250 \text{ km/h}) \left(\frac{1 \text{ m/s}}{3.6 \text{ km/h}} \right)}{\pi (15 \text{ L/min}) \left(\frac{1 \text{ min}}{60 \text{ s}} \right)} = 88.42 \text{ m}$$

$$A = \frac{\pi D^2}{4} = \frac{\pi (88.42 \text{ m})^2}{4} = 6140 \text{ m}^2$$

Then the average velocity of air through the wind turbine becomes

$$V = \frac{\dot{m}}{\rho A} = \frac{42,000 \text{ kg/s}}{(1.31 \text{ kg/m}^3)(6140 \text{ m}^2)} = \mathbf{5.23 \text{ m/s}}$$

(b) The kinetic energy of the air flowing through the turbine is

$$\dot{\text{KE}} = \frac{1}{2} \dot{m} V^2 = \frac{1}{2} (42,000 \text{ kg/s})(5.23 \text{ m/s})^2 = 574.3 \text{ kW}$$

Then the conversion efficiency of the turbine becomes

$$\eta = \frac{\dot{W}}{\dot{\text{KE}}} = \frac{180 \text{ kW}}{574.3 \text{ kW}} = \mathbf{0.313 = 31.3\%}$$

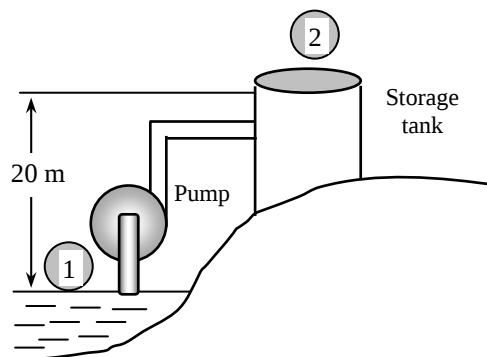
Discussion Note that about one-third of the kinetic energy of the wind is converted to power by the wind turbine, which is typical of actual turbines.

3-75 Water is pumped from a lake to a storage tank at a specified rate. The overall efficiency of the pump-motor unit and the pressure difference between the inlet and the exit of the pump are to be determined.

Assumptions 1 The elevations of the tank and the lake remain constant. 2 Frictional losses in the pipes are negligible. 3 The changes in kinetic energy are negligible. 4 The elevation difference across the pump is negligible.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis (a) We take the free surface of the lake to be point 1 and the free surfaces of the storage tank to be point 2. We also take the lake surface as the reference level ($z_1 = 0$), and thus the potential energy at points 1 and 2 are $pe_1 = 0$ and $pe_2 = gz_2$. The flow energy at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$). Further, the kinetic energy at both points is zero ($ke_1 = ke_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are



$$\dot{m} = \rho \dot{V} = (1000 \text{ kg/m}^3)(0.070 \text{ m}^3/\text{s}) = 70 \text{ kg/s}$$

$$pe_2 = gz_2 = (9.81 \text{ m/s}^2)(20 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.196 \text{ kJ/kg}$$

Then the rate of increase of the mechanical energy of water becomes

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}(pe_2 - 0) = \dot{m}pe_2 = (70 \text{ kg/s})(0.196 \text{ kJ/kg}) = 13.7 \text{ kW}$$

The overall efficiency of the combined pump-motor unit is determined from its definition,

$$\eta_{\text{pump-motor}} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{13.7 \text{ kW}}{20.4 \text{ kW}} = 0.672 \quad \text{or} \quad \mathbf{67.2\%}$$

(b) Now we consider the pump. The change in the mechanical energy of water as it flows through the pump consists of the change in the flow energy only since the elevation difference across the pump and the change in the kinetic energy are negligible. Also, this change must be equal to the useful mechanical energy supplied by the pump, which is 13.7 kW:

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m} \frac{P_2 - P_1}{\rho} = \dot{V} \Delta P$$

Solving for ΔP and substituting,

$$\Delta P = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{V}} = \frac{13.7 \text{ kJ/s}}{0.070 \text{ m}^3/\text{s}} \left(\frac{1 \text{ kPa} \cdot \text{m}^3}{1 \text{ kJ}} \right) = \mathbf{196 \text{ kPa}}$$

Therefore, the pump must boost the pressure of water by 196 kPa in order to raise its elevation by 20 m.

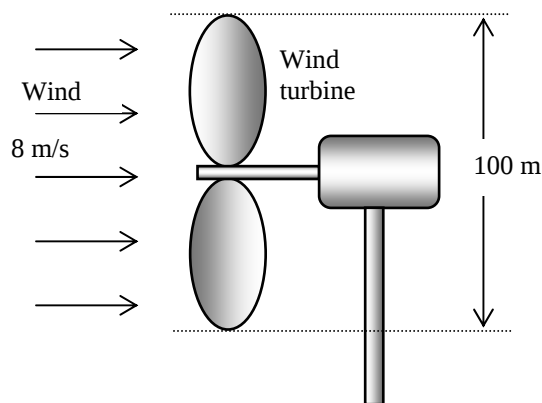
Discussion Note that only two-thirds of the electric energy consumed by the pump-motor is converted to the mechanical energy of water; the remaining one-third is wasted because of the inefficiencies of the pump and the motor.

3-76 A large wind turbine is installed at a location where the wind is blowing steadily at a certain velocity. The electric power generation, the daily electricity production, and the monetary value of this electricity are to be determined.

Assumptions 1 The wind is blowing steadily at a constant uniform velocity. 2 The efficiency of the wind turbine is independent of the wind speed.

Properties The density of air is given to be $\rho = 1.25 \text{ kg/m}^3$.

Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:



$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(8 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.032 \text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V \frac{\pi D^2}{4} = (1.25 \text{ kg/m}^3)(8 \text{ m/s}) \frac{\pi (100 \text{ m})^2}{4} = 78,540 \text{ kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (78,540 \text{ kg/s})(0.032 \text{ kJ/kg}) = 2513 \text{ kW}$$

The actual electric power generation is determined from

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}} \dot{W}_{\text{max}} = (0.32)(2513 \text{ kW}) = \mathbf{804.2 \text{ kW}}$$

Then the amount of electricity generated per day and its monetary value become

Amount of electricity = (Wind power)(Operating hours) = (804.2 kW)(24 h) = **19,300 kWh**

Revenues = (Amount of electricity)(Unit price) = (19,300 kWh)(\$0.06/kWh) = **\$1158** (per day)

Discussion Note that a single wind turbine can generate several thousand dollars worth of electricity every day at a reasonable cost, which explains the overwhelming popularity of wind turbines in recent years.

3-77E A water pump raises the pressure of water by a specified amount at a specified flow rate while consuming a known amount of electric power. The mechanical efficiency of the pump is to be determined.

Assumptions 1 The pump operates steadily. 2 The changes in velocity and elevation across the pump are negligible. 3 Water is incompressible.

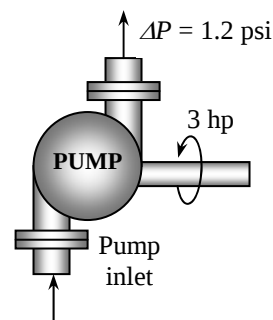
Analysis To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\begin{aligned}\Delta \dot{E}_{\text{mech, fluid}} &= \dot{m}(e_{\text{mech, out}} - e_{\text{mech, in}}) = \dot{m}[(P\boldsymbol{\nu})_2 - (P\boldsymbol{\nu})_1] = \dot{m}(P_2 - P_1)\boldsymbol{\nu} \\ &= \dot{\boldsymbol{\nu}}(P_2 - P_1) = (8 \text{ ft}^3/\text{s})(1.2 \text{ psi}) \left(\frac{1 \text{ Btu}}{5.404 \text{ psi} \cdot \text{ft}^3} \right) = 1.776 \text{ Btu/s} = 2.51 \text{ hp}\end{aligned}$$

since $1 \text{ hp} = 0.7068 \text{ Btu/s}$, $\dot{m} = \rho \dot{\boldsymbol{\nu}} = \dot{\boldsymbol{\nu}} / \boldsymbol{\nu}$, and there is no change in kinetic and potential energies of the fluid. Then the mechanical efficiency of the pump becomes

$$\eta_{\text{pump}} = \frac{\Delta \dot{E}_{\text{mech, fluid}}}{\dot{W}_{\text{pump, shaft}}} = \frac{2.51 \text{ hp}}{3 \text{ hp}} = 0.838 \quad \text{or} \quad \mathbf{83.8\%}$$

Discussion The overall efficiency of this pump will be lower than 83.8% because of the inefficiency of the electric motor that drives the pump.

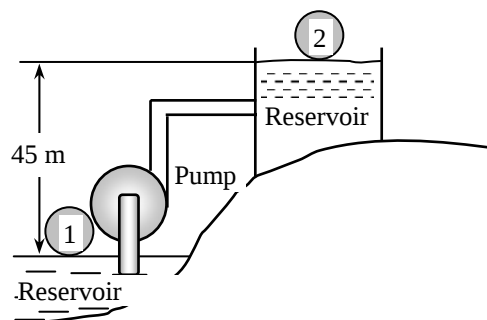


3-78 Water is pumped from a lower reservoir to a higher reservoir at a specified rate. For a specified shaft power input, the power that is converted to thermal energy is to be determined.

Assumptions 1 The pump operates steadily. 2 The elevations of the reservoirs remain constant. 3 The changes in kinetic energy are negligible.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,



$$\begin{aligned}\Delta \dot{E}_{\text{mech}} &= \dot{m} \Delta e_{\text{mech}} = \dot{m} \Delta pe = \dot{m} g \Delta z = \rho \dot{V} g \Delta z \\ &= (1000 \text{ kg/m}^3)(0.03 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(45 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kW}}{1000 \text{ N} \cdot \text{m/s}} \right) = 13.2 \text{ kW}\end{aligned}$$

Then the mechanical power lost because of frictional effects becomes

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump, in}} - \Delta \dot{E}_{\text{mech}} = 20 - 13.2 \text{ kW} = \mathbf{6.8 \text{ kW}}$$

Discussion The 6.8 kW of power is used to overcome the friction in the piping system. The effect of frictional losses in a pump is always to convert mechanical energy to an equivalent amount of thermal energy, which results in a slight rise in fluid temperature. Note that this pumping process could be accomplished by a 13.2 kW pump (rather than 20 kW) if there were no frictional losses in the system. In this ideal case, the pump would function as a turbine when the water is allowed to flow from the upper reservoir to the lower reservoir and extract 13.2 kW of power from the water.

3-79 The mass flow rate of water through the hydraulic turbines of a dam is to be determined.

Analysis The mass flow rate is determined from

$$\dot{W} = \dot{m} g (z_2 - z_1) \longrightarrow \dot{m} = \frac{\dot{W}}{g(z_2 - z_1)} = \frac{100,000 \text{ kJ/s}}{(9.8 \text{ m/s}^2)(206 - 0) \text{ m} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right)} = \mathbf{49,500 \text{ kg/s}}$$

3-80 A pump is pumping oil at a specified rate. The pressure rise of oil in the pump is measured, and the motor efficiency is specified. The mechanical efficiency of the pump is to be determined.

Assumptions 1 The flow is steady and incompressible. 2 The elevation difference across the pump is negligible.

Properties The density of oil is given to be $\rho = 860 \text{ kg/m}^3$.

Analysis Then the total mechanical energy of a fluid is the sum of the potential, flow, and kinetic energies, and is expressed per unit mass as $e_{\text{mech}} = gh + Pv + V^2/2$. To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta \dot{E}_{\text{mech, fluid}} = \dot{m}(e_{\text{mech, out}} - e_{\text{mech, in}}) = \dot{m} \left((Pv)_2 + \frac{V_2^2}{2} - (Pv)_1 - \frac{V_1^2}{2} \right) = \dot{V} \left((P_2 - P_1) + \rho \frac{V_2^2 - V_1^2}{2} \right)$$

since $\dot{m} = \rho \dot{V} = \dot{V}/v$, and there is no change in the potential energy of the fluid. Also,

$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.1 \text{ m}^3/\text{s}}{\pi (0.08 \text{ m})^2/4} = 19.9 \text{ m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.1 \text{ m}^3/\text{s}}{\pi (0.12 \text{ m})^2/4} = 8.84 \text{ m/s}$$

Substituting, the useful pumping power is determined to be

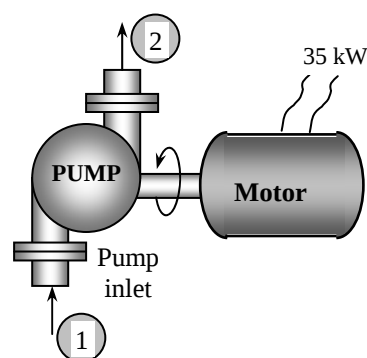
$$\begin{aligned} \dot{W}_{\text{pump, u}} &= \Delta \dot{E}_{\text{mech, fluid}} \\ &= (0.1 \text{ m}^3/\text{s}) \left(400 \text{ kN/m}^2 + (860 \text{ kg/m}^3) \frac{(8.84 \text{ m/s})^2 - (19.9 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \right) \left(\frac{1 \text{ kW}}{1 \text{ kN} \cdot \text{m/s}} \right) \\ &= 26.3 \text{ kW} \end{aligned}$$

Then the shaft power and the mechanical efficiency of the pump become

$$\dot{W}_{\text{pump, shaft}} = \eta_{\text{motor}} \dot{W}_{\text{electric}} = (0.90)(35 \text{ kW}) = 31.5 \text{ kW}$$

$$\eta_{\text{pump}} = \frac{\dot{W}_{\text{pump, u}}}{\dot{W}_{\text{pump, shaft}}} = \frac{26.3 \text{ kW}}{31.5 \text{ kW}} = 0.836 = \mathbf{83.6\%}$$

Discussion The overall efficiency of this pump/motor unit is the product of the mechanical and motor efficiencies, which is $0.9 \times 0.836 = 0.75$.



3-81E Water is pumped from a lake to a nearby pool by a pump with specified power and efficiency. The mechanical power used to overcome frictional effects is to be determined.

Assumptions 1 The flow is steady and incompressible. 2 The elevation difference between the lake and the free surface of the pool is constant. 3 The average flow velocity is constant since pipe diameter is constant.

Properties We take the density of water to be $\rho = 62.4 \text{ lbm/ft}^3$.

Analysis The useful mechanical pumping power delivered to water is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}} \dot{W}_{\text{pump}} = (0.73)(12 \text{ hp}) = 8.76 \text{ hp}$$

The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,

$$\Delta \dot{E}_{\text{mech}} = \dot{m} \Delta e_{\text{mech}} = \dot{m} \Delta pe = \dot{m} g \Delta z = \rho \dot{V} g \Delta z$$

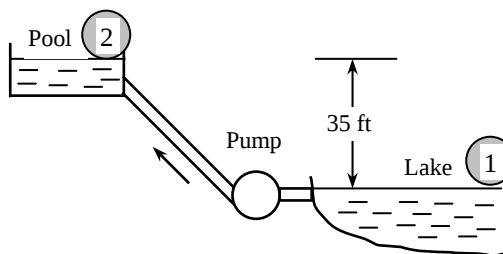
Substituting, the rate of change of mechanical energy of water becomes

$$\Delta \dot{E}_{\text{mech}} = (62.4 \text{ lbm/ft}^3)(1.2 \text{ ft}^3/\text{s})(32.2 \text{ ft/s}^2)(35 \text{ ft}) \left(\frac{1 \text{ lbf}}{32.2 \text{ lbm} \cdot \text{ft/s}^2} \right) \left(\frac{1 \text{ hp}}{550 \text{ lbf} \cdot \text{ft/s}} \right) = 4.76 \text{ hp}$$

Then the mechanical power lost in piping because of frictional effects becomes

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump,u}} - \Delta \dot{E}_{\text{mech}} = 8.76 - 4.76 \text{ hp} = \mathbf{4.0 \text{ hp}}$$

Discussion Note that the pump must supply to the water an additional useful mechanical power of 4.0 hp to overcome the frictional losses in pipes.



Energy and Environment

3-82C Energy conversion pollutes the soil, the water, and the air, and the environmental pollution is a serious threat to vegetation, wild life, and human health. The emissions emitted during the combustion of fossil fuels are responsible for smog, acid rain, and global warming and climate change. The primary chemicals that pollute the air are hydrocarbons (HC, also referred to as volatile organic compounds, VOC), nitrogen oxides (NO_x), and carbon monoxide (CO). The primary source of these pollutants is the motor vehicles.

3-83C Smog is the brown haze that builds up in a large stagnant air mass, and hangs over populated areas on calm hot summer days. Smog is made up mostly of ground-level ozone (O₃), but it also contains numerous other chemicals, including carbon monoxide (CO), particulate matter such as soot and dust, volatile organic compounds (VOC) such as benzene, butane, and other hydrocarbons. Ground-level ozone is formed when hydrocarbons and nitrogen oxides react in the presence of sunlight in hot calm days. Ozone irritates eyes and damage the air sacs in the lungs where oxygen and carbon dioxide are exchanged, causing eventual hardening of this soft and spongy tissue. It also causes shortness of breath, wheezing, fatigue, headaches, nausea, and aggravate respiratory problems such as asthma.

3-84C Fossil fuels include small amounts of sulfur. The sulfur in the fuel reacts with oxygen to form sulfur dioxide (SO₂), which is an air pollutant. The sulfur oxides and nitric oxides react with water vapor and other chemicals high in the atmosphere in the presence of sunlight to form sulfuric and nitric acids. The acids formed usually dissolve in the suspended water droplets in clouds or fog. These acid-laden droplets are washed from the air on to the soil by rain or snow. This is known as *acid rain*. It is called “rain” since it comes down with rain droplets.

As a result of acid rain, many lakes and rivers in industrial areas have become too acidic for fish to grow. Forests in those areas also experience a slow death due to absorbing the acids through their leaves, needles, and roots. Even marble structures deteriorate due to acid rain.

3-85C Carbon dioxide (CO₂), water vapor, and trace amounts of some other gases such as methane and nitrogen oxides act like a blanket and keep the earth warm at night by blocking the heat radiated from the earth. This is known as the *greenhouse effect*. The greenhouse effect makes life on earth possible by keeping the earth warm. But excessive amounts of these gases disturb the delicate balance by trapping too much energy, which causes the average temperature of the earth to rise and the climate at some localities to change. These undesirable consequences of the greenhouse effect are referred to as *global warming* or *global climate change*. The greenhouse effect can be reduced by reducing the net production of CO₂ by consuming less energy (for example, by buying energy efficient cars and appliances) and planting trees.

3-86C Carbon monoxide, which is a colorless, odorless, poisonous gas that deprives the body's organs from getting enough oxygen by binding with the red blood cells that would otherwise carry oxygen. At low levels, carbon monoxide decreases the amount of oxygen supplied to the brain and other organs and muscles, slows body reactions and reflexes, and impairs judgment. It poses a serious threat to people with heart disease because of the fragile condition of the circulatory system and to fetuses because of the oxygen needs of the developing brain. At high levels, it can be fatal, as evidenced by numerous deaths caused by cars that are warmed up in closed garages or by exhaust gases leaking into the cars.

3-87E A person trades in his Ford Taurus for a Ford Explorer. The extra amount of CO_2 emitted by the Explorer within 5 years is to be determined.

Assumptions The Explorer is assumed to use 940 gallons of gasoline a year compared to 715 gallons for Taurus.

Analysis The extra amount of gasoline the Explorer will use within 5 years is

$$\begin{aligned}\text{Extra Gasoline} &= (\text{Extra per year})(\text{No. of years}) \\ &= (940 - 715 \text{ gal/yr})(5 \text{ yr}) \\ &= 1125 \text{ gal} \\ \text{Extra CO}_2 \text{ produced} &= (\text{Extra gallons of gasoline used})(\text{CO}_2 \text{ emission per gallon}) \\ &= (1125 \text{ gal})(19.7 \text{ lbm/gal}) \\ &= \mathbf{22,163 \text{ lbm CO}_2}\end{aligned}$$

Discussion Note that the car we choose to drive has a significant effect on the amount of greenhouse gases produced.

3-88 A power plant that burns natural gas produces 0.59 kg of carbon dioxide (CO_2) per kWh. The amount of CO_2 production that is due to the refrigerators in a city is to be determined.

Assumptions The city uses electricity produced by a natural gas power plant.

Properties 0.59 kg of CO_2 is produced per kWh of electricity generated (given).

Analysis Noting that there are 200,000 households in the city and each household consumes 700 kWh of electricity for refrigeration, the total amount of CO_2 produced is

$$\begin{aligned}\text{Amount of CO}_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of CO}_2 \text{ per kWh}) \\ &= (200,000 \text{ household})(700 \text{ kWh/year household})(0.59 \text{ kg/kWh}) \\ &= 8.26 \times 10^7 \text{ CO}_2 \text{ kg/year} \\ &= \mathbf{82,600 \text{ CO}_2 \text{ ton/year}}\end{aligned}$$

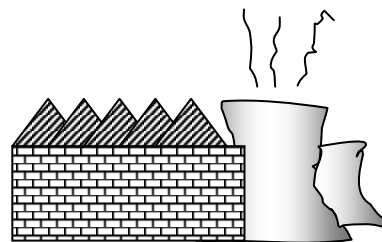
Therefore, the refrigerators in this city are responsible for the production of 82,600 tons of CO_2 .

3-89 A power plant that burns coal, produces 1.1 kg of carbon dioxide (CO₂) per kWh. The amount of CO₂ production that is due to the refrigerators in a city is to be determined.

Assumptions The city uses electricity produced by a coal power plant.

Properties 1.1 kg of CO₂ is produced per kWh of electricity generated (given).

Analysis Noting that there are 200,000 households in the city and each household consumes 700 kWh of electricity for refrigeration, the total amount of CO₂ produced is



$$\begin{aligned}
 \text{Amount of CO}_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of CO}_2 \text{ per kWh}) \\
 &= (200,000 \text{ household})(700 \text{ kWh/household})(1.1 \text{ kg/kWh}) \\
 &= 15.4 \times 10^7 \text{ CO}_2 \text{ kg/year} \\
 &= \mathbf{154,000 \text{ CO}_2 \text{ ton/year}}
 \end{aligned}$$

Therefore, the refrigerators in this city are responsible for the production of 154,000 tons of CO₂.

3-90E A household uses fuel oil for heating, and electricity for other energy needs. Now the household reduces its energy use by 20%. The reduction in the CO₂ production this household is responsible for is to be determined.

Properties The amount of CO₂ produced is 1.54 lbm per kWh and 26.4 lbm per gallon of fuel oil (given).

Analysis Noting that this household consumes 11,000 kWh of electricity and 1500 gallons of fuel oil per year, the amount of CO₂ production this household is responsible for is

$$\begin{aligned}
 \text{Amount of CO}_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of CO}_2 \text{ per kWh}) \\
 &\quad + (\text{Amount of fuel oil consumed})(\text{Amount of CO}_2 \text{ per gallon}) \\
 &= (11,000 \text{ kWh/yr})(1.54 \text{ lbm/kWh}) + (1500 \text{ gal/yr})(26.4 \text{ lbm/gal}) \\
 &= 56,540 \text{ CO}_2 \text{ lbm/year}
 \end{aligned}$$

Then reducing the electricity and fuel oil usage by 15% will reduce the annual amount of CO₂ production by this household by

$$\begin{aligned}
 \text{Reduction in CO}_2 \text{ produced} &= (0.15)(\text{Current amount of CO}_2 \text{ production}) \\
 &= (0.15)(56,540 \text{ CO}_2 \text{ kg/year}) \\
 &= \mathbf{8481 \text{ CO}_2 \text{ lbm/year}}
 \end{aligned}$$

Therefore, any measure that saves energy also reduces the amount of pollution emitted to the environment.

3-91 A household has 2 cars, a natural gas furnace for heating, and uses electricity for other energy needs. The annual amount of NO_x emission to the atmosphere this household is responsible for is to be determined.

Properties The amount of NO_x produced is 7.1 g per kWh, 4.3 g per therm of natural gas, and 11 kg per car (given).

Analysis Noting that this household has 2 cars, consumes 1200 therms of natural gas, and 9,000 kWh of electricity per year, the amount of NO_x production this household is responsible for is



$$\begin{aligned}
 \text{Amount of } \text{NO}_x \text{ produced} &= (\text{No. of cars})(\text{Amount of } \text{NO}_x \text{ produced per car}) \\
 &\quad + (\text{Amount of electricity consumed})(\text{Amount of } \text{NO}_x \text{ per kWh}) \\
 &\quad + (\text{Amount of gas consumed})(\text{Amount of } \text{NO}_x \text{ per gallon}) \\
 &= (2 \text{ cars})(11 \text{ kg/car}) + (9000 \text{ kWh/yr})(0.0071 \text{ kg/kWh}) \\
 &\quad + (1200 \text{ therms/yr})(0.0043 \text{ kg/therm}) \\
 &= \mathbf{91.06 \text{ NO}_x \text{ kg/year}}
 \end{aligned}$$

Discussion Any measure that saves energy will also reduce the amount of pollution emitted to the atmosphere.

Review Problems

3-92 A decision is to be made between a cheaper but inefficient natural gas heater and an expensive but efficient natural gas heater for a house.

Assumptions The two heaters are comparable in all aspects other than the initial cost and efficiency.

Analysis Other things being equal, the logical choice is the heater that will cost less during its lifetime. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period.

The annual heating cost is given to be \$1200. Noting that the existing heater is 55% efficient, only 55% of that energy (and thus money) is delivered to the house, and the rest is wasted due to the inefficiency of the heater. Therefore, the monetary value of the heating load of the house is

$$\begin{aligned}\text{Cost of useful heat} &= (55\%)(\text{Current annual heating cost}) \\ &= 0.55 \times (\$1200/\text{yr}) = \$660/\text{yr}\end{aligned}$$

Gas Heater $\eta_1 = 82\%$ $\eta_2 = 95\%$

This is how much it would cost to heat this house with a heater that is 100% efficient. For heaters that are less efficient, the annual heating cost is determined by dividing \$660 by the efficiency:

82% heater: Annual cost of heating = (Cost of useful heat)/Efficiency = $(\$660/\text{yr})/0.82 = \$805/\text{yr}$

95% heater: Annual cost of heating = (Cost of useful heat)/Efficiency = $(\$660/\text{yr})/0.95 = \$695/\text{yr}$

Annual cost savings with the efficient heater = $805 - 695 = \$110$

Excess initial cost of the efficient heater = $2700 - 1600 = \$1100$

The simple payback period becomes

$$\text{Simple payback period} = \frac{\text{Excess initial cost}}{\text{Annual cost savings}} = \frac{\$1100}{\$110/\text{yr}} = \mathbf{10 \text{ years}}$$

Therefore, the more efficient heater will pay for the \$1100 cost differential in this case in 10 years, which is more than the 8-year limit. Therefore, the purchase of the cheaper and less efficient heater is a better buy in this case.

3-93 A wind turbine is rotating at 20 rpm under steady winds of 30 km/h. The power produced, the tip speed of the blade, and the revenue generated by the wind turbine per year are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The wind turbine operates continuously during the entire year at the specified conditions.

Properties The density of air is given to be $\rho = 1.20 \text{ kg/m}^3$.

Analysis (a) The blade span area and the mass flow rate of air through the turbine are

$$A = \pi D^2 / 4 = \pi (80 \text{ m})^2 / 4 = 5027 \text{ m}^2$$

$$V = (30 \text{ km/h}) \left(\frac{1000 \text{ m}}{1 \text{ km}} \right) \left(\frac{1 \text{ h}}{3600 \text{ s}} \right) = 8.333 \text{ m/s}$$

$$\dot{m} = \rho AV = (1.2 \text{ kg/m}^3)(5027 \text{ m}^2)(8.333 \text{ m/s}) = 50,270 \text{ kg/s}$$

Noting that the kinetic energy of a unit mass is $V^2/2$ and the wind turbine captures 35% of this energy, the power generated by this wind turbine becomes

$$\dot{W} = \eta \left(\frac{1}{2} \dot{m} V^2 \right) = (0.35) \frac{1}{2} (50,270 \text{ kg/s})(8.333 \text{ m/s})^2 \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{610.9 \text{ kW}}$$

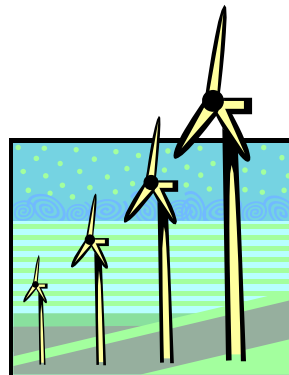
(b) Noting that the tip of blade travels a distance of πD per revolution, the tip velocity of the turbine blade for an rpm of n becomes

$$V_{\text{tip}} = \pi D n = \pi (80 \text{ m})(20 / \text{min}) = 5027 \text{ m/min} = 83.8 \text{ m/s} = \mathbf{302 \text{ km/h}}$$

(c) The amount of electricity produced and the revenue generated per year are

$$\begin{aligned} \text{Electricity produced} &= \dot{W} \Delta t = (610.9 \text{ kW})(365 \times 24 \text{ h/year}) \\ &= 5.351 \times 10^6 \text{ kWh/year} \end{aligned}$$

$$\begin{aligned} \text{Revenue generated} &= (\text{Electricity produced})(\text{Unit price}) = (5.351 \times 10^6 \text{ kWh/year})(\$0.06/\text{kWh}) \\ &= \mathbf{\$321,100/\text{year}} \end{aligned}$$



3-94 A wind turbine is rotating at 20 rpm under steady winds of 25 km/h. The power produced, the tip speed of the blade, and the revenue generated by the wind turbine per year are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The wind turbine operates continuously during the entire year at the specified conditions.

Properties The density of air is given to be $\rho = 1.20 \text{ kg/m}^3$.

Analysis (a) The blade span area and the mass flow rate of air through the turbine are

$$A = \pi D^2 / 4 = \pi (80 \text{ m})^2 / 4 = 5027 \text{ m}^2$$

$$V = (25 \text{ km/h}) \left(\frac{1000 \text{ m}}{1 \text{ km}} \right) \left(\frac{1 \text{ h}}{3600 \text{ s}} \right) = 6.944 \text{ m/s}$$

$$\dot{m} = \rho AV = (1.2 \text{ kg/m}^3)(5027 \text{ m}^2)(6.944 \text{ m/s}) = 41,891 \text{ kg/s}$$

Noting that the kinetic energy of a unit mass is $V^2/2$ and the wind turbine captures 35% of this energy, the power generated by this wind turbine becomes

$$\dot{W} = \eta \left(\frac{1}{2} \dot{m} V^2 \right) = (0.35) \frac{1}{2} (41,891 \text{ kg/s})(6.944 \text{ m/s})^2 \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{353.5 \text{ kW}}$$

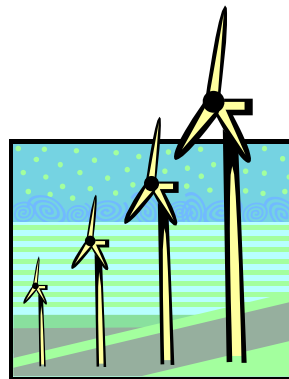
(b) Noting that the tip of blade travels a distance of πD per revolution, the tip velocity of the turbine blade for an rpm of $\dot{n}$ becomes

$$V_{\text{tip}} = \pi D \dot{n} = \pi (80 \text{ m})(20 / \text{min}) = 5027 \text{ m/min} = 83.8 \text{ m/s} = \mathbf{302 \text{ km/h}}$$

(c) The amount of electricity produced and the revenue generated per year are

$$\begin{aligned} \text{Electricity produced} &= \dot{W} \Delta t = (353.5 \text{ kW})(365 \times 24 \text{ h/year}) \\ &= 3,096,660 \text{ kWh/year} \end{aligned}$$

$$\begin{aligned} \text{Revenue generated} &= (\text{Electricity produced})(\text{Unit price}) = (3,096,660 \text{ kWh/year})(\$0.06/\text{kWh}) \\ &= \mathbf{\$185,800/\text{year}} \end{aligned}$$



3-95E The energy contents, unit costs, and typical conversion efficiencies of various energy sources for use in water heaters are given. The lowest cost energy source is to be determined.

Assumptions The differences in installation costs of different water heaters are not considered.

Properties The energy contents, unit costs, and typical conversion efficiencies of different systems are given in the problem statement.

Analysis The unit cost of each Btu of useful energy supplied to the water heater by each system can be determined from

$$\text{Unit cost of useful energy} = \frac{\text{Unit cost of energy supplied}}{\text{Conversion efficiency}}$$

Substituting,

$$\text{Natural gas heater:} \quad \text{Unit cost of useful energy} = \frac{\$0.012/\text{ft}^3}{0.55} \left(\frac{1 \text{ ft}^3}{1025 \text{ Btu}} \right) = \$21.3 \times 10^{-6} / \text{Btu}$$

$$\text{Heating by oil heater:} \quad \text{Unit cost of useful energy} = \frac{\$1.15/\text{gal}}{0.55} \left(\frac{1 \text{ gal}}{138,700 \text{ Btu}} \right) = \$15.1 \times 10^{-6} / \text{Btu}$$

$$\text{Electric heater:} \quad \text{Unit cost of useful energy} = \frac{\$0.084/\text{kWh}}{0.90} \left(\frac{1 \text{ kWh}}{3412 \text{ Btu}} \right) = \$27.4 \times 10^{-6} / \text{Btu}$$

Therefore, the lowest cost energy source for hot water heaters in this case is **oil**.

3-96 A home owner is considering three different heating systems for heating his house. The system with the lowest energy cost is to be determined.

Assumptions The differences in installation costs of different heating systems are not considered.

Properties The energy contents, unit costs, and typical conversion efficiencies of different systems are given in the problem statement.

Analysis The unit cost of each Btu of useful energy supplied to the house by each system can be determined from

$$\text{Unit cost of useful energy} = \frac{\text{Unit cost of energy supplied}}{\text{Conversion efficiency}}$$

Substituting,

$$\text{Natural gas heater:} \quad \text{Unit cost of useful energy} = \frac{\$1.24/\text{therm}}{0.87} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = \$13.5 \times 10^{-6} / \text{kJ}$$

$$\text{Heating oil heater:} \quad \text{Unit cost of useful energy} = \frac{\$1.25/\text{gal}}{0.87} \left(\frac{1 \text{ gal}}{138,500 \text{ kJ}} \right) = \$10.4 \times 10^{-6} / \text{kJ}$$

$$\text{Electric heater:} \quad \text{Unit cost of useful energy} = \frac{\$0.09/\text{kWh}}{1.0} \left(\frac{1 \text{ kWh}}{3600 \text{ kJ}} \right) = \$25.0 \times 10^{-6} / \text{kJ}$$

Therefore, the system with the lowest energy cost for heating the house is the **heating oil heater**.

3-97 The heating and cooling costs of a poorly insulated house can be reduced by up to 30 percent by adding adequate insulation. The time it will take for the added insulation to pay for itself from the energy it saves is to be determined.

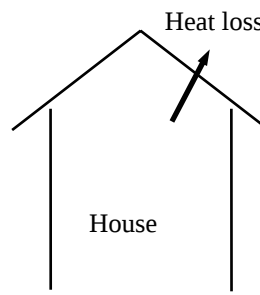
Assumptions It is given that the annual energy usage of a house is \$1200 a year, and 46% of it is used for heating and cooling. The cost of added insulation is given to be \$200.

Analysis The amount of money that would be saved per year is determined directly from

$$\text{Money saved} = (\$1200/\text{year})(0.46)(0.30) = \$166/\text{yr}$$

Then the simple payback period becomes

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$200}{\$166/\text{yr}} = \mathbf{1.2 \text{ yr}}$$



Therefore, the proposed measure will pay for itself in less than one and a half year.

3-98 Caulking and weather-stripping doors and windows to reduce air leaks can reduce the energy use of a house by up to 10 percent. The time it will take for the caulking and weather-stripping to pay for itself from the energy it saves is to be determined.

Assumptions It is given that the annual energy usage of a house is \$1100 a year, and the cost of caulking and weather-stripping a house is \$50.

Analysis The amount of money that would be saved per year is determined directly from

$$\text{Money saved} = (\$1100/\text{year})(0.10) = \$110/\text{yr}$$

Then the simple payback period becomes

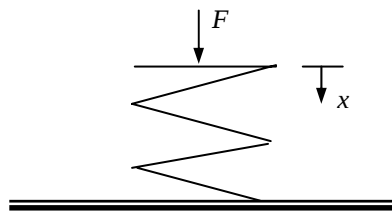
$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$50}{\$110/\text{yr}} = \mathbf{0.45 \text{ yr}}$$

Therefore, the proposed measure will pay for itself in less than half a year.

3-99 The work required to compress a spring is to be determined.

Analysis (a) With the preload, $F = F_0 + kx$. Substituting this into the work integral gives

$$\begin{aligned} W &= \int_1^2 F ds = \int_1^2 (kx + F_0) dx \\ &= \frac{k}{2} (x_2^2 - x_1^2) + F_0 (x_2 - x_1) \\ &= \frac{300 \text{ N/cm}}{2} [(1 \text{ cm})^2 - 0^2] + (100 \text{ N})[(1 \text{ cm}) - 0] \\ &= 250 \text{ N} \cdot \text{cm} = 2.50 \text{ N} \cdot \text{m} = 2.50 \text{ J} = \mathbf{0.0025 \text{ kJ}} \end{aligned}$$

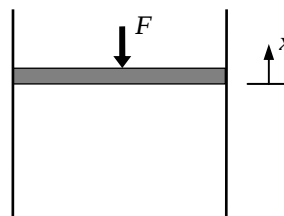


3-100 The work required to compress a gas in a gas spring is to be determined.

Assumptions All forces except that generated by the gas spring will be neglected.

Analysis When the expression given in the problem statement is substituted into the work integral relation, and advantage is taken of the fact that the force and displacement vectors are collinear, the result is

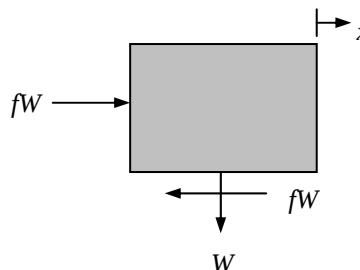
$$\begin{aligned}
 W &= \int_1^2 F ds = \int_1^2 \frac{\text{Constant}}{x^k} dx \\
 &= \frac{\text{Constant}}{1-k} (x_2^{1-k} - x_1^{1-k}) \\
 &= \frac{1000 \text{ N} \cdot \text{m}^{1.3}}{1-1.3} [(0.3 \text{ m})^{-0.3} - (0.1 \text{ m})^{-0.3}] \\
 &= 1867 \text{ N} \cdot \text{m} = 1867 \text{ J} = \mathbf{1.87 \text{ kJ}}
 \end{aligned}$$



3-101E A man pushes a block along a horizontal plane. The work required to move the block is to be determined considering (a) the man and (b) the block as the system.

Analysis The work applied to the block to overcome the friction is found by using the work integral,

$$\begin{aligned}
 W &= \int_1^2 F ds = \int_1^2 fW (x_2 - x_1) \\
 &= (0.2)(100 \text{ lbf})(100 \text{ ft}) \\
 &= 2000 \text{ lbf} \cdot \text{ft} \\
 &= (2000 \text{ lbf} \cdot \text{ft}) \left(\frac{1 \text{ Btu}}{778.169 \text{ lbf} \cdot \text{ft}} \right) = \mathbf{2.57 \text{ Btu}}
 \end{aligned}$$



The man must then produce the amount of work

$$W = \mathbf{2.57 \text{ Btu}}$$

3-102E The power required to pump a specified rate of water to a specified elevation is to be determined.

Properties The density of water is taken to be 62.4 lbm/ft^3 (Table A-3E).

Analysis The required power is determined from

$$\begin{aligned}
 \dot{W} &= mg(z_2 - z_1) = \rho \dot{V} g(z_2 - z_1) \\
 &= (62.4 \text{ lbm/ft}^3)(200 \text{ gal/min}) \left(\frac{35.315 \text{ ft}^3/\text{s}}{15,850 \text{ gal/min}} \right) (32.174 \text{ ft/s}^2)(300 \text{ ft}) \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm} \cdot \text{ft/s}^2} \right) \\
 &= 8342 \text{ lbf} \cdot \text{ft/s} = (8342 \text{ lbf} \cdot \text{ft/s}) \left(\frac{1 \text{ kW}}{737.56 \text{ lbf} \cdot \text{ft/s}} \right) = \mathbf{11.3 \text{ kW}}
 \end{aligned}$$

3-103 The power that could be produced by a water wheel is to be determined.

Properties The density of water is taken to be $1000 \text{ m}^3/\text{kg}$ (Table A-3).

Analysis The power production is determined from

$$\begin{aligned}\dot{W} &= \dot{m}g(z_2 - z_1) = \rho \dot{V}g(z_2 - z_1) \\ &= (1000 \text{ kg/m}^3)(0.400/60 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(10 \text{ m})\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = \mathbf{0.654 \text{ kW}}\end{aligned}$$

3-104 The flow of air through a flow channel is considered. The diameter of the wind channel downstream from the rotor and the power produced by the windmill are to be determined.

Analysis The specific volume of the air is

$$\nu = \frac{RT}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})}{100 \text{ kPa}} = 0.8409 \text{ m}^3/\text{kg}$$

The diameter of the wind channel downstream from the rotor is

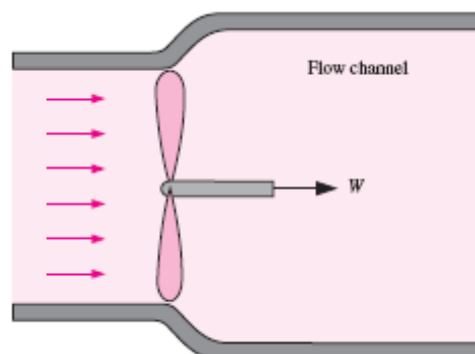
$$\begin{aligned}A_1 V_1 &= A_2 V_2 \longrightarrow (\pi D_1^2 / 4) V_1 = (\pi D_2^2 / 4) V_2 \\ \longrightarrow D_2 &= D_1 \sqrt{\frac{V_1}{V_2}} = (7 \text{ m}) \sqrt{\frac{10 \text{ m/s}}{9 \text{ m/s}}} = \mathbf{7.38 \text{ m}}\end{aligned}$$

The mass flow rate through the wind mill is

$$\dot{m} = \frac{A_1 V_1}{\nu} = \frac{\pi (7 \text{ m})^2 (10 \text{ m/s})}{4(0.8409 \text{ m}^3/\text{kg})} = 457.7 \text{ kg/s}$$

The power produced is then

$$\dot{W} = \dot{m} \frac{V_1^2 - V_2^2}{2} = (457.7 \text{ kg/s}) \frac{(10 \text{ m/s})^2 - (9 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{4.35 \text{ kW}}$$

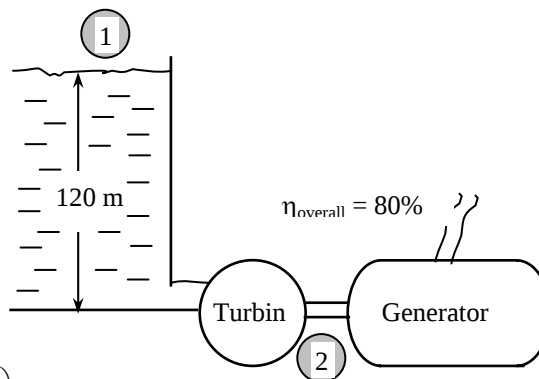


3-105 The available head, flow rate, and efficiency of a hydroelectric turbine are given. The electric power output is to be determined.

Assumptions 1 The flow is steady. 2 Water levels at the reservoir and the discharge site remain constant. 3 Frictional losses in piping are negligible.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3 = 1 \text{ kg/L}$.

Analysis The total mechanical energy the water in a dam possesses is equivalent to the potential energy of water at the free surface of the dam (relative to free surface of discharge water), and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate.



$$e_{\text{mech}} = pe = gz = (9.81 \text{ m/s}^2)(120 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 1.177 \text{ kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho \dot{V} = (1000 \text{ kg/m}^3)(100 \text{ m}^3/\text{s}) = 100,000 \text{ kg/s}$$

Then the maximum and actual electric power generation become

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (100,000 \text{ kg/s})(1.177 \text{ kJ/kg}) \left(\frac{1 \text{ MW}}{1000 \text{ kJ/s}} \right) = 117.7 \text{ MW}$$

$$\dot{W}_{\text{electric}} = \eta_{\text{overall}} \dot{W}_{\text{max}} = 0.80(117.7 \text{ MW}) = \mathbf{94.2 \text{ MW}}$$

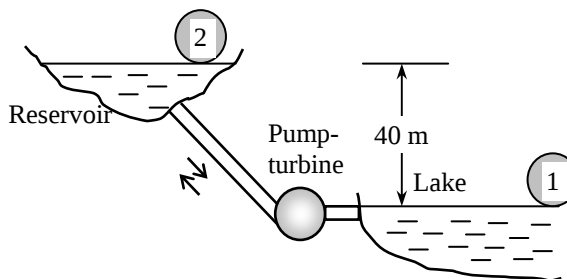
Discussion Note that the power generation would increase by more than 1 MW for each percentage point improvement in the efficiency of the turbine-generator unit.

3-106 An entrepreneur is to build a large reservoir above the lake level, and pump water from the lake to the reservoir at night using cheap power, and let the water flow from the reservoir back to the lake during the day, producing power. The potential revenue this system can generate per year is to be determined.

Assumptions 1 The flow in each direction is steady and incompressible. 2 The elevation difference between the lake and the reservoir can be taken to be constant, and the elevation change of reservoir during charging and discharging is disregarded. 3 Frictional losses in piping are negligible. 4 The system operates every day of the year for 10 hours in each mode.

Properties We take the density of water to be $\rho = 1000 \text{ kg/m}^3$.

Analysis The total mechanical energy of water in an upper reservoir relative to water in a lower reservoir is equivalent to the potential energy of water at the free surface of this reservoir relative to free surface of the lower reservoir. Therefore, the power potential of water is its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate. This also represents the minimum power required to pump water from the lower reservoir to the higher reservoir.



$$\begin{aligned}\dot{W}_{\max, \text{ turbine}} &= \dot{W}_{\min, \text{ pump}} = \dot{W}_{\text{ideal}} = \Delta \dot{E}_{\text{mech}} = \dot{m} \Delta e_{\text{mech}} = \dot{m} \Delta pe = \dot{m} g \Delta z = \rho \dot{V} g \Delta z \\ &= (1000 \text{ kg/m}^3)(2 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(40 \text{ m}) \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kW}}{1000 \text{ N} \cdot \text{m/s}} \right) \\ &= 784.8 \text{ kW}\end{aligned}$$

The actual pump and turbine electric powers are

$$\begin{aligned}\dot{W}_{\text{pump, elect}} &= \frac{\dot{W}_{\text{ideal}}}{\eta_{\text{pump-motor}}} = \frac{784.8 \text{ kW}}{0.75} = 1046 \text{ kW} \\ \dot{W}_{\text{turbine}} &= \eta_{\text{turbine-gen}} \dot{W}_{\text{ideal}} = 0.75(784.8 \text{ kW}) = 588.6 \text{ kW}\end{aligned}$$

Then the power consumption cost of the pump, the revenue generated by the turbine, and the net income (revenue minus cost) per year become

$$\text{Cost} = \dot{W}_{\text{pump, elect}} \Delta t \times \text{Unit price} = (1046 \text{ kW})(365 \times 10 \text{ h/year})(\$0.03/\text{kWh}) = \$114,500/\text{year}$$

$$\text{Revenue} = \dot{W}_{\text{turbine}} \Delta t \times \text{Unit price} = (588.6 \text{ kW})(365 \times 10 \text{ h/year})(\$0.08/\text{kWh}) = \$171,900/\text{year}$$

$$\text{Net income} = \text{Revenue} - \text{Cost} = 171,900 - 114,500 = \mathbf{\$57,400/\text{year}}$$

Discussion It appears that this pump-turbine system has a potential to generate net revenues of about \$57,000 per year. A decision on such a system will depend on the initial cost of the system, its life, the operating and maintenance costs, the interest rate, and the length of the contract period, among other things.

3-107 A diesel engine burning light diesel fuel that contains sulfur is considered. The rate of sulfur that ends up in the exhaust and the rate of sulfurous acid given off to the environment are to be determined.

Assumptions 1 All of the sulfur in the fuel ends up in the exhaust. 2 For one kmol of sulfur in the exhaust, one kmol of sulfurous acid is added to the environment.

Properties The molar mass of sulfur is 32 kg/kmol.

Analysis The mass flow rates of fuel and the sulfur in the exhaust are

$$\dot{m}_{\text{fuel}} = \frac{\dot{m}_{\text{air}}}{\text{AF}} = \frac{(336 \text{ kg air/h})}{(18 \text{ kg air/kg fuel})} = 18.67 \text{ kg fuel/h}$$

$$\dot{m}_{\text{Sulfur}} = (750 \times 10^{-6}) \dot{m}_{\text{fuel}} = (750 \times 10^{-6})(18.67 \text{ kg/h}) = \mathbf{0.014 \text{ kg/h}}$$

The rate of sulfurous acid given off to the environment is

$$\dot{m}_{\text{H}_2\text{SO}_3} = \frac{M_{\text{H}_2\text{SO}_3}}{M_{\text{Sulfur}}} \dot{m}_{\text{Sulfur}} = \frac{2 \times 1 + 32 + 3 \times 16}{32} (0.014 \text{ kg/h}) = \mathbf{0.036 \text{ kg/h}}$$

Discussion This problem shows why the sulfur percentage in diesel fuel must be below certain value to satisfy regulations.

3-108 Lead is a very toxic engine emission. Leaded gasoline contains lead that ends up in the exhaust. The amount of lead put out to the atmosphere per year for a given city is to be determined.

Assumptions 35% of lead is exhausted to the environment.

Analysis The gasoline consumption and the lead emission are

$$\text{Gasoline Consumption} = (10,000 \text{ cars})(15,000 \text{ km/car} \cdot \text{year})(10 \text{ L}/100 \text{ km}) = 1.5 \times 10^7 \text{ L/year}$$

$$\begin{aligned} \text{Lead Emission} &= (\text{Gasoline Consumption}) m_{\text{lead}} f_{\text{lead}} \\ &= (1.5 \times 10^7 \text{ L/year})(0.15 \times 10^{-3} \text{ kg/L})(0.35) \\ &= \mathbf{788 \text{ kg/year}} \end{aligned}$$

Discussion Note that a huge amount of lead emission is avoided by the use of unleaded gasoline.

3-109 ... 3-115 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 4
PROPERTIES OF PURE
SUBSTANCES

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Pure Substances, Phase Change Processes, Property Diagrams

4-1C Yes, since the chemical composition throughout the tank remain the same.

4-2C A liquid that is about to vaporize is saturated liquid; otherwise it is compressed liquid.

4-3C A vapor that is about to condense is saturated vapor; otherwise it is superheated vapor.

4-4C No.

4-5C The temperature will also increase since the boiling or saturation temperature of a pure substance depends on pressure.

4-6C Because one cannot be varied while holding the other constant. In other words, when one changes, so does the other one.

4-7C At critical point the saturated liquid and the saturated vapor states are identical. At triple point the three phases of a pure substance coexist in equilibrium.

4-8C Yes.

4-9C Case (c) when the pan is covered with a heavy lid. Because the heavier the lid, the greater the pressure in the pan, and thus the greater the cooking temperature.

4-10C At supercritical pressures, there is no distinct phase change process. The liquid uniformly and gradually expands into a vapor. At subcritical pressures, there is always a distinct surface between the phases.

Property Tables

4-11C A perfectly fitting pot and its lid often stick after cooking as a result of the vacuum created inside as the temperature and thus the corresponding saturation pressure inside the pan drops. An easy way of removing the lid is to reheat the food. When the temperature rises to boiling level, the pressure rises to atmospheric value and thus the lid will come right off.

4-12C The molar mass of gasoline (C_8H_{18}) is 114 kg/kmol, which is much larger than the molar mass of air that is 29 kg/kmol. Therefore, the gasoline vapor will settle down instead of rising even if it is at a much higher temperature than the surrounding air. As a result, the warm mixture of air and gasoline on top of an open gasoline will most likely settle down instead of rising in a cooler environment

4-13C Ice can be made by evacuating the air in a water tank. During evacuation, vapor is also thrown out, and thus the vapor pressure in the tank drops, causing a difference between the vapor pressures at the water surface and in the tank. This pressure difference is the driving force of vaporization, and forces the liquid to evaporate. But the liquid must absorb the heat of vaporization before it can vaporize, and it absorbs it from the liquid and the air in the neighborhood, causing the temperature in the tank to drop. The process continues until water starts freezing. The process can be made more efficient by insulating the tank well so that the entire heat of vaporization comes essentially from the water.

4-14C Yes. Otherwise we can create energy by alternately vaporizing and condensing a substance.

4-15C No. Because in the thermodynamic analysis we deal with the changes in properties; and the changes are independent of the selected reference state.

4-16C The term h_{fg} represents the amount of energy needed to vaporize a unit mass of saturated liquid at a specified temperature or pressure. It can be determined from $h_{fg} = h_g - h_f$.

4-17C Yes; the higher the temperature the lower the h_{fg} value.

4-18C Quality is the fraction of vapor in a saturated liquid-vapor mixture. It has no meaning in the superheated vapor region.

4-19C Completely vaporizing 1 kg of saturated liquid at 1 atm pressure since the higher the pressure, the lower the h_{fg} .

4-20C Yes. It decreases with increasing pressure and becomes zero at the critical pressure.

4-21C No. Quality is a mass ratio, and it is not identical to the volume ratio.

4-22C The compressed liquid can be approximated as a saturated liquid at the given temperature. Thus $\nu_{T,P} \cong \nu_{f@T}$.

4-23 CD EES Complete the following table for H_2O :

$T, ^\circ\text{C}$	P, kPa	$v, \text{m}^3/\text{kg}$	Phase description
50	12.352	4.16	Saturated mixture
120.21	200	0.8858	Saturated vapor
250	400	0.5952	Superheated vapor
110	600	0.001051	Compressed liquid

4-24 EES Problem 4-23 is reconsidered. The missing properties of water are to be determined using EES, and the solution is to be repeated for refrigerant-134a, refrigerant-22, and ammonia.

Analysis The problem is solved using EES, and the solution is given below.

\$Warning off

{ \$Arrays off }

Procedure Find(Fluid\$, Prop1\$, Prop2\$, Value1, Value2: T, p, h, s, v, u, x, State\$)

"Due to the very general nature of this problem, a large number of 'if-then-else' statements are necessary."

If Prop1\$='Temperature, C' Then

 T=Value1

 If Prop2\$='Temperature, C' then Call Error('Both properties cannot be Temperature, T=xxx°F2', T)

 if Prop2\$='Pressure, kPa' then

 p=value2

 h=enthalpy(Fluid\$, T=T, P=p)

 s=entropy(Fluid\$, T=T, P=p)

 v=volume(Fluid\$, T=T, P=p)

 u=intenergy(Fluid\$, T=T, P=p)

 x=quality(Fluid\$, T=T, P=p)

 endif

 if Prop2\$='Enthalpy, kJ/kg' then

 h=value2

 p=Pressure(Fluid\$, T=T, h=h)

 s=entropy(Fluid\$, T=T, h=h)

 v=volume(Fluid\$, T=T, h=h)

 u=intenergy(Fluid\$, T=T, h=h)

 x=quality(Fluid\$, T=T, h=h)

 endif

 if Prop2\$='Entropy, kJ/kg-K' then

 s=value2

 p=Pressure(Fluid\$, T=T, s=s)

 h=enthalpy(Fluid\$, T=T, s=s)

 v=volume(Fluid\$, T=T, s=s)

 u=intenergy(Fluid\$, T=T, s=s)

 x=quality(Fluid\$, T=T, s=s)

 endif

 if Prop2\$='Volume, m³/kg' then

 v=value2

 p=Pressure(Fluid\$, T=T, v=v)

 h=enthalpy(Fluid\$, T=T, v=v)

 s=entropy(Fluid\$, T=T, v=v)

 u=intenergy(Fluid\$, T=T, v=v)

 x=quality(Fluid\$, T=T, v=v)

 endif

 if Prop2\$='Internal Energy, kJ/kg' then

 u=value2

 p=Pressure(Fluid\$, T=T, u=u)

 h=enthalpy(Fluid\$, T=T, u=u)

 s=entropy(Fluid\$, T=T, u=u)

 v=volume(Fluid\$, T=T, s=s)

 x=quality(Fluid\$, T=T, u=u)

 endif

 if Prop2\$='Quality' then

```

    x=value2
    p=Pressure(Fluid$,T=T,x=x)
    h=enthalpy(Fluid$,T=T,x=x)
    s=entropy(Fluid$,T=T,x=x)
    v=volume(Fluid$,T=T,x=x)
    u=IntEnergy(Fluid$,T=T,x=x)
  endif
Endif
If Prop1$='Pressure, kPa' Then
  p=Value1
  If Prop2$='Pressure, kPa' then Call Error('Both properties cannot be Pressure, p=xxxF2',p)
  if Prop2$='Temperature, C' then
    T=value2
    h=enthalpy(Fluid$,T=T,P=p)
    s=entropy(Fluid$,T=T,P=p)
    v=volume(Fluid$,T=T,P=p)
    u=intenergy(Fluid$,T=T,P=p)
    x=quality(Fluid$,T=T,P=p)
  endif
  if Prop2$='Enthalpy, kJ/kg' then
    h=value2
    T=Temperature(Fluid$,p=p,h=h)
    s=entropy(Fluid$,p=p,h=h)
    v=volume(Fluid$,p=p,h=h)
    u=intenergy(Fluid$,p=p,h=h)
    x=quality(Fluid$,p=p,h=h)
  endif
  if Prop2$='Entropy, kJ/kg-K' then
    s=value2
    T=Temperature(Fluid$,p=p,s=s)
    h=enthalpy(Fluid$,p=p,s=s)
    v=volume(Fluid$,p=p,s=s)
    u=intenergy(Fluid$,p=p,s=s)
    x=quality(Fluid$,p=p,s=s)
  endif
  if Prop2$='Volume, m^3/kg' then
    v=value2
    T=Temperature(Fluid$,p=p,v=v)
    h=enthalpy(Fluid$,p=p,v=v)
    s=entropy(Fluid$,p=p,v=v)
    u=intenergy(Fluid$,p=p,v=v)
    x=quality(Fluid$,p=p,v=v)
  endif
  if Prop2$='Internal Energy, kJ/kg' then
    u=value2
    T=Temperature(Fluid$,p=p,u=u)
    h=enthalpy(Fluid$,p=p,u=u)
    s=entropy(Fluid$,p=p,u=u)
    v=volume(Fluid$,p=p,s=s)
    x=quality(Fluid$,p=p,u=u)
  endif
  if Prop2$='Quality' then
    x=value2
    T=Temperature(Fluid$,p=p,x=x)
    h=enthalpy(Fluid$,p=p,x=x)
    s=entropy(Fluid$,p=p,x=x)
  endif

```

```

        v=volume(Fluid$,p=p,x=x)
        u=IntEnergy(Fluid$,p=p,x=x)
    endif
Endif
If Prop1$='Enthalpy, kJ/kg' Then
    h=Value1
    If Prop2$='Enthalpy, kJ/kg' then Call Error('Both properties cannot be Enthalpy, h=xxxF2',h)
    if Prop2$='Pressure, kPa' then
        p=value2
        T=Temperature(Fluid$,h=h,P=p)
        s=entropy(Fluid$,h=h,P=p)
        v=volume(Fluid$,h=h,P=p)
        u=intenergy(Fluid$,h=h,P=p)
        x=quality(Fluid$,h=h,P=p)
    endif
    if Prop2$='Temperature, C' then
        T=value2
        p=Pressure(Fluid$,T=T,h=h)
        s=entropy(Fluid$,T=T,h=h)
        v=volume(Fluid$,T=T,h=h)
        u=intenergy(Fluid$,T=T,h=h)
        x=quality(Fluid$,T=T,h=h)
    endif
    if Prop2$='Entropy, kJ/kg-K' then
        s=value2
        p=Pressure(Fluid$,h=h,s=s)
        T=Temperature(Fluid$,h=h,s=s)
        v=volume(Fluid$,h=h,s=s)
        u=intenergy(Fluid$,h=h,s=s)
        x=quality(Fluid$,h=h,s=s)
    endif
    if Prop2$='Volume, m^3/kg' then
        v=value2
        p=Pressure(Fluid$,h=h,v=v)
        T=Temperature(Fluid$,h=h,v=v)
        s=entropy(Fluid$,h=h,v=v)
        u=intenergy(Fluid$,h=h,v=v)
        x=quality(Fluid$,h=h,v=v)
    endif
    if Prop2$='Internal Energy, kJ/kg' then
        u=value2
        p=Pressure(Fluid$,h=h,u=u)
        T=Temperature(Fluid$,h=h,u=u)
        s=entropy(Fluid$,h=h,u=u)
        v=volume(Fluid$,h=h,s=s)
        x=quality(Fluid$,h=h,u=u)
    endif
    if Prop2$='Quality' then
        x=value2
        p=Pressure(Fluid$,h=h,x=x)
        T=Temperature(Fluid$,h=h,x=x)
        s=entropy(Fluid$,h=h,x=x)
        v=volume(Fluid$,h=h,x=x)
        u=IntEnergy(Fluid$,h=h,x=x)
    endif
endif
endif

```

```

If Prop1$='Entropy, kJ/kg-K' Then
  s=Value1
  If Prop2$='Entropy, kJ/kg-K' then Call Error('Both properties cannot be Entropy, h=xxxF2',s)
  if Prop2$='Pressure, kPa' then
    p=value2
    T=Temperature(Fluid$,s=s,P=p)
    h=enthalpy(Fluid$,s=s,P=p)
    v=volume(Fluid$,s=s,P=p)
    u=intenergy(Fluid$,s=s,P=p)
    x=quality(Fluid$,s=s,P=p)
  endif
  if Prop2$='Temperature, C' then
    T=value2
    p=Pressure(Fluid$,T=T,s=s)
    h=enthalpy(Fluid$,T=T,s=s)
    v=volume(Fluid$,T=T,s=s)
    u=intenergy(Fluid$,T=T,s=s)
    x=quality(Fluid$,T=T,s=s)
  endif
  if Prop2$='Enthalpy, kJ/kg' then
    h=value2
    p=Pressure(Fluid$,h=h,s=s)
    T=Temperature(Fluid$,h=h,s=s)
    v=volume(Fluid$,h=h,s=s)
    u=intenergy(Fluid$,h=h,s=s)
    x=quality(Fluid$,h=h,s=s)
  endif
  if Prop2$='Volume, m^3/kg' then
    v=value2
    p=Pressure(Fluid$,s=s,v=v)
    T=Temperature(Fluid$,s=s,v=v)
    h=enthalpy(Fluid$,s=s,v=v)
    u=intenergy(Fluid$,s=s,v=v)
    x=quality(Fluid$,s=s,v=v)
  endif
  if Prop2$='Internal Energy, kJ/kg' then
    u=value2
    p=Pressure(Fluid$,s=s,u=u)
    T=Temperature(Fluid$,s=s,u=u)
    h=enthalpy(Fluid$,s=s,u=u)
    v=volume(Fluid$,s=s,u=u)
    x=quality(Fluid$,s=s,u=u)
  endif
  if Prop2$='Quality' then
    x=value2
    p=Pressure(Fluid$,s=s,x=x)
    T=Temperature(Fluid$,s=s,x=x)
    h=enthalpy(Fluid$,s=s,x=x)
    v=volume(Fluid$,s=s,x=x)
    u=IntEnergy(Fluid$,s=s,x=x)
  endif
Endif
if x<0 then State$='in the compressed liquid region.'
if x>1 then State$='in the superheated region.'
If (x<1) and (X>0) then State$='in the two-phase region.'
If (x=1) then State$='a saturated vapor.'

```

if (x=0) then State\$='a saturated liquid.'

end

"Input from the diagram window"

{Fluid\$='Steam'

Prop1\$='Temperature'

Prop2\$='Pressure'

Value1=50

value2=101.3}

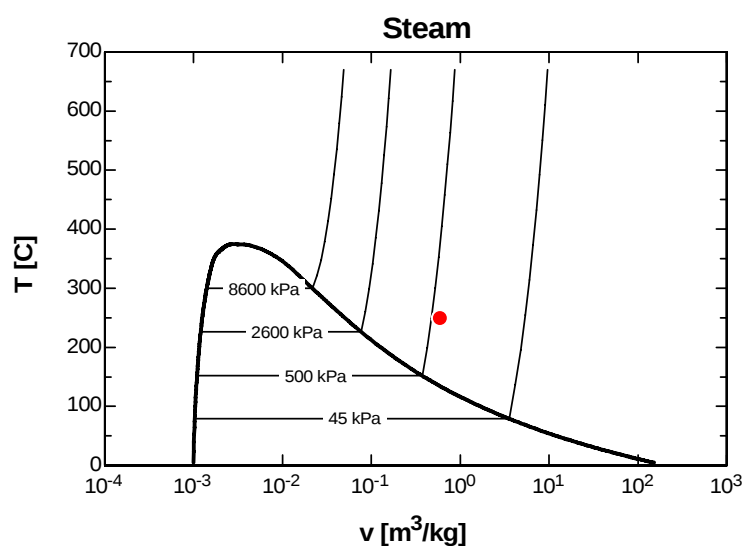
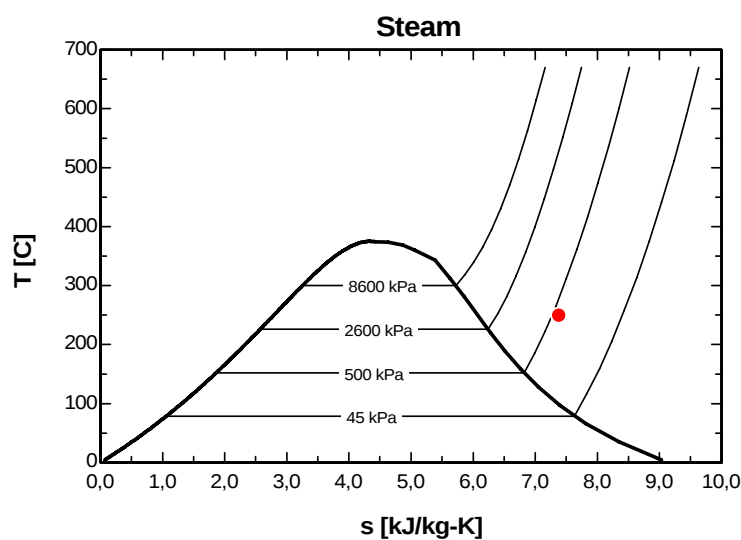
Call Find(Fluid\$,Prop1\$,Prop2\$,Value1,Value2:T,p,h,s,v,u,x,State\$)

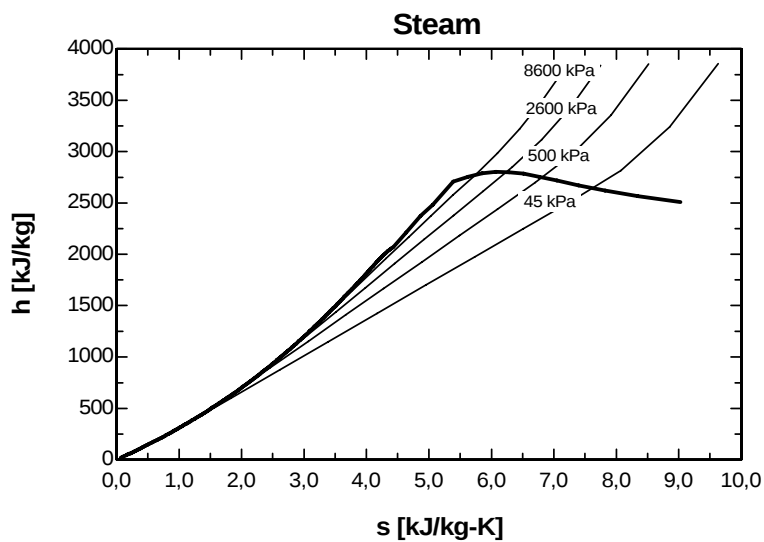
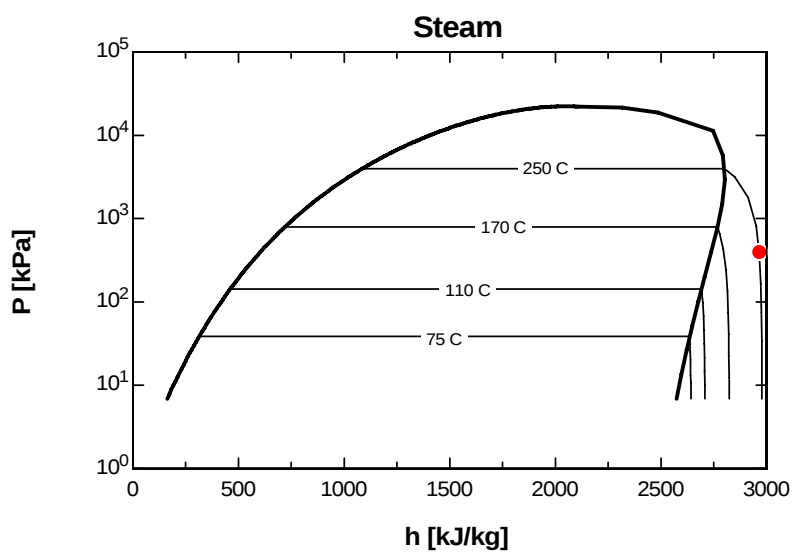
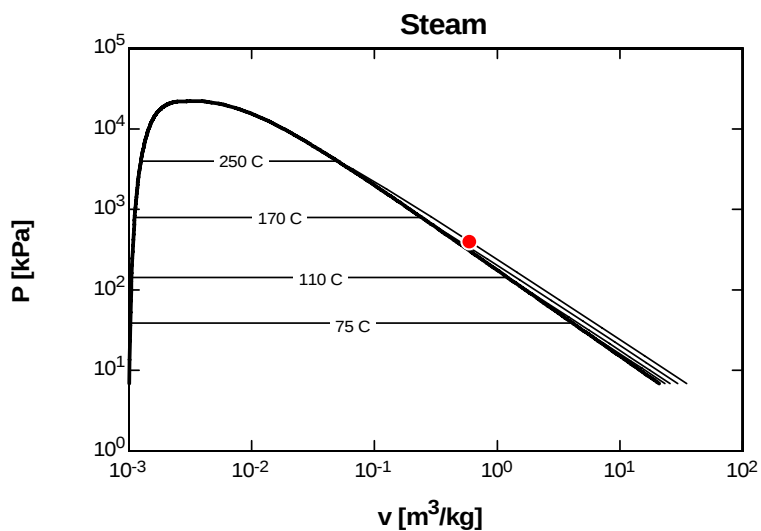
T[1]=T ; p[1]=p ; h[1]=h ; s[1]=s ; v[1]=v ; u[1]=u ; x[1]=x

"Array variables were used so the states can be plotted on property plots."

ARRAYS TABLE

h kJ/kg	P kPa	s kJ/kgK	T C	u kJ/kg	v m ³ /kg	x
2964.5	400	7.3804	250	2726.4	0.5952	100





4-25E Complete the following table for H_2O :

T, °F	P, psia	u, Btu / lbm	Phase description
300	67.03	782	<i>Saturated mixture</i>
267.22	40	236.02	Saturated liquid
500	120	1174.4	<i>Superheated vapor</i>
400	400	373.84	<i>Compressed liquid</i>

4-26E EES Problem 4-25E is reconsidered. The missing properties of water are to be determined using EES, and the solution is to be repeated for refrigerant-134a, refrigerant-22, and ammonia.

Analysis The problem is solved using EES, and the solution is given below.

"Given"

T[1]=300 [F]
u[1]=782 [Btu/lbm]
P[2]=40 [psia]
x[2]=0
T[3]=500 [F]
P[3]=120 [psia]
T[4]=400 [F]
P[4]=420 [psia]

"Analysis"

Fluid\$='steam_iapws'
P[1]=pressure(Fluid\$, T=T[1], u=u[1])
x[1]=quality(Fluid\$, T=T[1], u=u[1])
T[2]=temperature(Fluid\$, P=P[2], x=x[2])
u[2]=intenergy(Fluid\$, P=P[2], x=x[2])
u[3]=intenergy(Fluid\$, P=P[3], T=T[3])
x[3]=quality(Fluid\$, P=P[3], T=T[3])
u[4]=intenergy(Fluid\$, P=P[4], T=T[4])
x[4]=quality(Fluid\$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"

Solution for steam

T, °F	P, psia	x	u, Btu/lbm
300	67.028	0.6173	782
267.2	40	0	236
500	120	100	1174
400	400	-100	373.8

4-27 Complete the following table for H_2O :

T, °C	P, kPa	h, kJ / kg	x	Phase description
120.21	200	2045.8	0.7	Saturated mixture
140	361.53	1800	0.565	Saturated mixture
177.66	950	752.74	0.0	Saturated liquid
80	500	335.37	- - -	Compressed liquid
350.0	800	3162.2	- - -	Superheated vapor

4-28 Complete the following table for Refrigerant-134a:

T, °C	P, kPa	v , m ³ / kg	Phase description
-8	320	0.0007569	Compressed liquid
30	770.64	0.015	Saturated mixture
-12.73	180	0.11041	Saturated vapor
80	600	0.044710	Superheated vapor

4-29 Complete the following table for Refrigerant-134a:

T, °C	P, kPa	u, kJ / kg	Phase description
20	572.07	95	Saturated mixture
-12	185.37	35.78	Saturated liquid
86.24	400	300	Superheated vapor
8	600	62.26	Compressed liquid

4-30E Complete the following table for Refrigerant-134a:

T, °F	P, psia	h, Btu / lbm	x	Phase description
65.89	80	78	0.566	Saturated mixture
15	29.759	69.92	0.6	Saturated mixture
10	70	15.35	- - -	Compressed liquid
160	180	129.46	- - -	Superheated vapor
110	161.16	117.23	1.0	Saturated vapor

4-31 A piston-cylinder device contains R-134a at a specified state. Heat is transferred to R-134a. The final pressure, the volume change of the cylinder, and the enthalpy change are to be determined.

Analysis (a) The final pressure is equal to the initial pressure, which is determined from

$$P_2 = P_1 = P_{\text{atm}} + \frac{m_p g}{\pi D^2 / 4} = 88 \text{ kPa} + \frac{(12 \text{ kg})(9.81 \text{ m/s}^2)}{\pi (0.25 \text{ m})^2 / 4} \left(\frac{1 \text{ kN}}{1000 \text{ kg.m/s}^2} \right) = \mathbf{90.4 \text{ kPa}}$$

(b) The specific volume and enthalpy of R-134a at the initial state of 90.4 kPa and -10°C and at the final state of 90.4 kPa and 15°C are (from EES)

$$\nu_1 = 0.2302 \text{ m}^3/\text{kg} \quad h_1 = 247.76 \text{ kJ/kg}$$

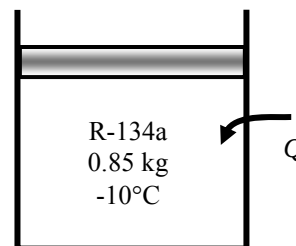
$$\nu_2 = 0.2544 \text{ m}^3/\text{kg} \quad h_2 = 268.16 \text{ kJ/kg}$$

The initial and the final volumes and the volume change are

$$\nu_1 = m \nu_1 = (0.85 \text{ kg})(0.2302 \text{ m}^3/\text{kg}) = 0.1957 \text{ m}^3$$

$$\nu_2 = m \nu_2 = (0.85 \text{ kg})(0.2544 \text{ m}^3/\text{kg}) = 0.2162 \text{ m}^3$$

$$\Delta \nu = 0.2162 - 0.1957 = \mathbf{0.0205 \text{ m}^3}$$



(c) The total enthalpy change is determined from

$$\Delta H = m(h_2 - h_1) = (0.85 \text{ kg})(268.16 - 247.76) \text{ kJ/kg} = \mathbf{17.4 \text{ kJ/kg}}$$

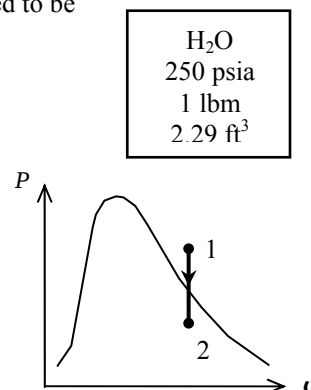
4-32E A rigid container that is filled with water is cooled. The initial temperature and final pressure are to be determined.

Analysis The initial state is superheated vapor. The temperature is determined to be

$$\left. \begin{array}{l} P_1 = 250 \text{ psia} \\ \nu_1 = 2.29 \text{ ft}^3/\text{lbm} \end{array} \right\} T_1 = \mathbf{550^\circ\text{F}} \quad (\text{Table A - 6E})$$

This is a constant volume cooling process ($\nu = \nu/m = \text{constant}$). The final state is saturated mixture and thus the pressure is the saturation pressure at the final temperature:

$$\left. \begin{array}{l} T_2 = 100^\circ\text{F} \\ \nu_2 = \nu_1 = 2.29 \text{ ft}^3/\text{lbm} \end{array} \right\} P_2 = P_{\text{sat @ } 100^\circ\text{F}} = \mathbf{0.9505 \text{ psia}} \quad (\text{Table A - 4E})$$



4-33 A piston-cylinder device that is filled with R-134a is heated. The final volume is to be determined.

Analysis The initial specific volume is

$$\nu_1 = \frac{V_1}{m} = \frac{0.14 \text{ m}^3}{1 \text{ kg}} = 0.14 \text{ m}^3/\text{kg}$$

This is a constant-pressure process. The initial state is determined to be a mixture, and thus the pressure is the saturation pressure at the given temperature

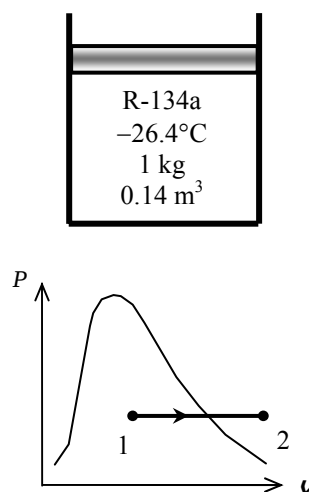
$$P_1 = P_2 = P_{\text{sat @ } -26.4^\circ\text{C}} = 100 \text{ kPa} \quad (\text{Table A - 12})$$

The final state is superheated vapor and the specific volume is

$$\left. \begin{array}{l} P_2 = 100 \text{ kPa} \\ T_2 = 100^\circ\text{C} \end{array} \right\} \nu_2 = 0.30138 \text{ m}^3/\text{kg} \quad (\text{Table A - 13})$$

The final volume is then

$$\nu_2 = m\nu_2 = (1 \text{ kg})(0.30138 \text{ m}^3/\text{kg}) = \mathbf{0.30138 \text{ m}^3}$$



4-34 Left chamber of a partitioned system contains water at a specified state while the right chamber is evacuated. The partition is now ruptured and heat is transferred from the water. The pressure at the final state is to be determined.

Analysis The initial specific volume is

$$\nu_1 = \frac{V_1}{m} = \frac{1.1989 \text{ m}^3}{1 \text{ kg}} = 1.1989 \text{ m}^3/\text{kg}$$

At the final state, the water occupies three times the initial volume. Then,

$$\nu_2 = 3\nu_1 = 3(1.1989 \text{ m}^3/\text{kg}) = 3.5967 \text{ m}^3/\text{kg}$$

Based on this specific volume and the final temperature, the final state is a saturated mixture and the pressure is

$$P_2 = P_{\text{sat}@3^\circ\text{C}} = \mathbf{0.768 \text{ kPa}} \quad (\text{Table A - 4})$$

Water 200 kPa 1 kg 1.1989 m ³	Evacuated
---	-----------

4-35E A piston-cylinder device that is filled with water is cooled. The final pressure and volume of the water are to be determined.

Analysis The initial specific volume is

$$\nu_1 = \frac{V_1}{m} = \frac{2.3615 \text{ ft}^3}{1 \text{ lbm}} = 2.3615 \text{ ft}^3/\text{lbm}$$

This is a constant-pressure process. The initial state is determined to be superheated vapor and thus the pressure is determined to be

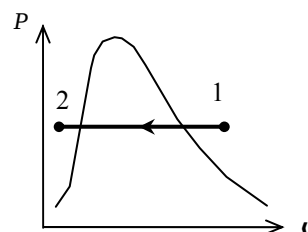
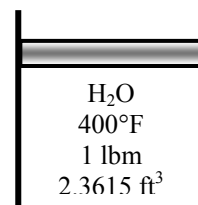
$$\left. \begin{array}{l} T_1 = 400^\circ\text{F} \\ \nu_1 = 2.3615 \text{ ft}^3/\text{lbm} \end{array} \right\} P_1 = P_2 = \mathbf{200 \text{ psia}} \quad (\text{Table A - 6E})$$

The saturation temperature at 200 psia is 381.8°F. Since the final temperature is less than this temperature, the final state is compressed liquid. Using the incompressible liquid approximation,

$$\nu_2 = \nu_f @ 100^\circ\text{F} = 0.01613 \text{ ft}^3/\text{lbm} \quad (\text{Table A - 4E})$$

The final volume is then

$$\nu_2 = m\nu_2 = (1 \text{ lbm})(0.01613 \text{ ft}^3/\text{lbm}) = \mathbf{0.01613 \text{ ft}^3}$$



4-36 A piston-cylinder device that is filled with R-134a is heated. The final volume is to be determined.

Analysis This is a constant pressure process. The initial specific volume is

$$\nu_1 = \frac{V}{m} = \frac{1.595 \text{ m}^3}{10 \text{ kg}} = 0.1595 \text{ m}^3/\text{kg}$$

The initial state is determined to be a mixture, and thus the pressure is the saturation pressure at the given temperature

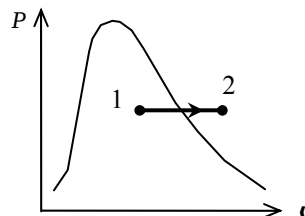
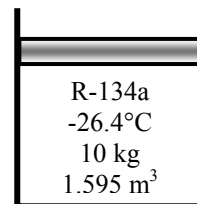
$$P_1 = P_{\text{sat @ } -26.4^\circ\text{C}} = 100 \text{ kPa} \quad (\text{Table A - 12})$$

The final state is superheated vapor and the specific volume is

$$\left. \begin{array}{l} P_2 = 100 \text{ kPa} \\ T_2 = 100^\circ\text{C} \end{array} \right\} \nu_2 = 0.30138 \text{ m}^3/\text{kg} \quad (\text{Table A - 13})$$

The final volume is then

$$V_2 = m\nu_2 = (10 \text{ kg})(0.30138 \text{ m}^3/\text{kg}) = \mathbf{3.0138 \text{ m}^3}$$



4-37 The internal energy of water at a specified state is to be determined.

Analysis The state of water is superheated vapor. From the steam tables,

$$\left. \begin{array}{l} P = 50 \text{ kPa} \\ T = 200^\circ\text{C} \end{array} \right\} u = \mathbf{2660.0 \text{ kJ/kg}} \quad (\text{Table A - 6})$$

4-38 The specific volume of water at a specified state is to be determined using the incompressible liquid approximation and it is to be compared to the more accurate value.

Analysis The state of water is compressed liquid. From the steam tables,

$$\left. \begin{array}{l} P = 5 \text{ MPa} \\ T = 100^\circ\text{C} \end{array} \right\} \nu = \mathbf{0.001041 \text{ m}^3/\text{kg}} \quad (\text{Table A - 7})$$

Based upon the incompressible liquid approximation,

$$\left. \begin{array}{l} P = 2 \text{ MPa} \\ T = 100^\circ\text{C} \end{array} \right\} \nu \cong \nu_f @ 100^\circ\text{C} = \mathbf{0.001043 \text{ m}^3/\text{kg}} \quad (\text{Table A - 4})$$

The error involved is

$$\text{Percent Error} = \frac{0.001043 - 0.001041}{0.001041} \times 100 = \mathbf{0.19\%}$$

which is quite acceptable in most engineering calculations.

4-39E The total internal energy and enthalpy of water in a container are to be determined.

Analysis The specific volume is

$$\nu = \frac{V}{m} = \frac{2 \text{ ft}^3}{1 \text{ lbm}} = 2 \text{ ft}^3/\text{lbm}$$

At this specific volume and the given pressure, the state is a saturated mixture. The quality, internal energy, and enthalpy at this state are (Table A-5E)

Water
100 psia
2 ft ³

$$x = \frac{\nu - \nu_f}{\nu_{fg}} = \frac{(2 - 0.01774) \text{ ft}^3/\text{lbm}}{(4.4327 - 0.01774) \text{ ft}^3/\text{lbm}} = 0.4490$$

$$u = u_f + xu_{fg} = 298.19 + (0.4490)(807.29) = 660.7 \text{ Btu/lbm}$$

$$h = h_f + xh_{fg} = 298.51 + (0.4490)(888.99) = 697.7 \text{ Btu/lbm}$$

The total internal energy and enthalpy are then

$$U = mu = (1 \text{ lbm})(660.7 \text{ Btu/lbm}) = \mathbf{660.7 \text{ Btu}}$$

$$H = mh = (1 \text{ lbm})(697.7 \text{ Btu/lbm}) = \mathbf{697.7 \text{ Btu}}$$

4-40 The volume of a container that contains water at a specified state is to be determined.

Analysis The specific volume is determined from steam tables by interpolation to be

$$\left. \begin{array}{l} P = 100 \text{ kPa} \\ T = 360^\circ\text{C} \end{array} \right\} \nu = 2.9172 \text{ m}^3/\text{kg} \text{ (Table A - 6)}$$

The volume of the container is then

$$V = m\nu = (3 \text{ kg})(2.9172 \text{ m}^3/\text{kg}) = \mathbf{8.752 \text{ m}^3}$$

Water
3 kg
100 kPa
360°C

4-41 A rigid container that is filled with R-134a is heated. The temperature and total enthalpy are to be determined at the initial and final states.

Analysis This is a constant volume process. The specific volume is

$$\nu_1 = \nu_2 = \frac{\nu}{m} = \frac{0.014 \text{ m}^3}{10 \text{ kg}} = 0.0014 \text{ m}^3/\text{kg}$$

The initial state is determined to be a mixture, and thus the temperature is the saturation temperature at the given pressure. From Table A-12 by interpolation

$$T_1 = T_{\text{sat @ 300 kPa}} = \mathbf{0.61^\circ\text{C}}$$

and

$$x_1 = \frac{\nu_1 - \nu_f}{\nu_{fg}} = \frac{(0.0014 - 0.0007736) \text{ m}^3/\text{kg}}{(0.067978 - 0.0007736) \text{ m}^3/\text{kg}} = 0.009321$$

$$h_1 = h_f + x_1 h_{fg} = 52.67 + (0.009321)(198.13) = 54.52 \text{ kJ/kg}$$

The total enthalpy is then

$$H_1 = m h_1 = (10 \text{ kg})(54.52 \text{ kJ/kg}) = \mathbf{545.2 \text{ kJ}}$$

The final state is also saturated mixture. Repeating the calculations at this state,

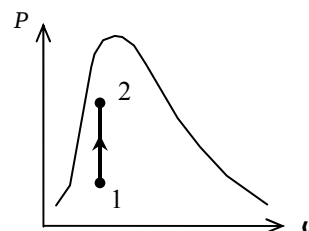
$$T_2 = T_{\text{sat @ 600 kPa}} = \mathbf{21.55^\circ\text{C}}$$

$$x_2 = \frac{\nu_2 - \nu_f}{\nu_{fg}} = \frac{(0.0014 - 0.0008199) \text{ m}^3/\text{kg}}{(0.034295 - 0.0008199) \text{ m}^3/\text{kg}} = 0.01733$$

$$h_2 = h_f + x_2 h_{fg} = 81.51 + (0.01733)(180.90) = 84.64 \text{ kJ/kg}$$

$$H_2 = m h_2 = (10 \text{ kg})(84.64 \text{ kJ/kg}) = \mathbf{846.4 \text{ kJ}}$$

R-134a
300 kPa
10 kg
14 L



4-42 A piston-cylinder device that is filled with R-134a is cooled at constant pressure. The final temperature and the change of total internal energy are to be determined.

Analysis The initial specific volume is

$$\nu_1 = \frac{\nu}{m} = \frac{12.322 \text{ m}^3}{100 \text{ kg}} = 0.12322 \text{ m}^3/\text{kg}$$

The initial state is superheated and the internal energy at this state is

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ \nu_1 = 0.12322 \text{ m}^3/\text{kg} \end{array} \right\} u_1 = 263.08 \text{ kJ/kg (Table A -13)}$$

The final specific volume is

$$\nu_2 = \frac{\nu_1}{2} = \frac{0.12322 \text{ m}^3 / \text{kg}}{2} = 0.06161 \text{ m}^3/\text{kg}$$

This is a constant pressure process. The final state is determined to be saturated mixture whose temperature is

$$T_2 = T_{\text{sat @ 200 kPa}} = -10.09^\circ\text{C (Table A -12)}$$

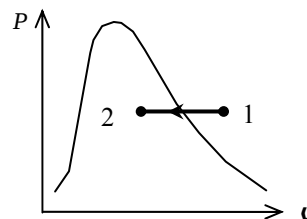
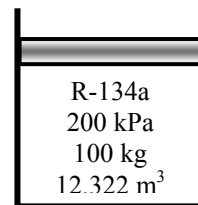
The internal energy at the final state is (Table A-12)

$$x_2 = \frac{\nu_2 - \nu_f}{\nu_{fg}} = \frac{(0.06161 - 0.0007533) \text{ m}^3/\text{kg}}{(0.099867 - 0.0007533) \text{ m}^3/\text{kg}} = 0.6140$$

$$u_2 = u_f + x_2 u_{fg} = 38.28 + (0.6140)(186.21) = 152.61 \text{ kJ/kg}$$

Hence, the change in the internal energy is

$$\Delta u = u_2 - u_1 = 152.61 - 263.08 = -110.47 \text{ kJ/kg}$$



4-43 A spring-loaded piston-cylinder device is filled with water. The water now undergoes a process until its volume is one-half of the original volume. The final temperature and the entropy are to be determined.

Analysis From the steam tables,

$$\left. \begin{array}{l} P_1 = 4 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \nu_1 = 0.07343 \text{ m}^3/\text{kg} \text{ (Table A - 6)}$$

The process experienced by this system is a linear P - ν process. The equation for this line is

$$P - P_1 = c(\nu - \nu_1)$$

where P_1 is the system pressure when its specific volume is ν_1 . The spring equation may be written as

$$P - P_1 = \frac{F_s - F_{s,1}}{A} = k \frac{x - x_1}{A} = \frac{kA}{A^2} (x - x_1) = \frac{k}{A^2} (\nu - \nu_1) = \frac{km}{A^2} (\nu - \nu_1)$$

Constant c is hence

$$c = \frac{km}{A^2} = \frac{4^2 km}{\pi^2 D^4} = \frac{(16)(90 \text{ kN/m})(0.5 \text{ kg})}{\pi^2 (0.2 \text{ m})^4} = 45,595 \text{ kN} \cdot \text{kg}/\text{m}^5$$

The final pressure is then

$$\begin{aligned} P_2 &= P_1 + c(\nu_2 - \nu_1) = P_1 + c\left(\frac{\nu_1}{2} - \nu_1\right) = P_1 - \frac{c}{2}\nu_1 \\ &= 4000 \text{ kPa} - \frac{45,595 \text{ kN} \cdot \text{kg}/\text{m}^5}{2} (0.07343 \text{ m}^3/\text{kg}) = 2326 \text{ kPa} \end{aligned}$$

and

$$\nu_2 = \frac{\nu_1}{2} = \frac{0.07343 \text{ m}^3/\text{kg}}{2} = 0.03672 \text{ m}^3/\text{kg}$$

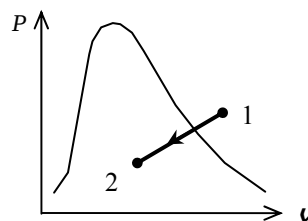
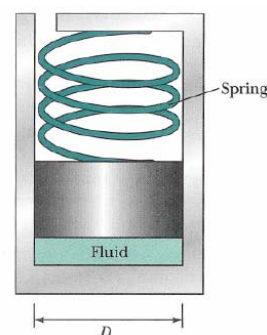
The final state is a mixture and the temperature is

$$T_2 = T_{\text{sat @ } 2326 \text{ kPa}} \cong \mathbf{220^\circ\text{C}} \text{ (Table A - 5)}$$

The quality and the entropy at the final state are

$$x_2 = \frac{\nu_2 - \nu_f}{\nu_{fg}} = \frac{(0.03672 - 0.001190) \text{ m}^3/\text{kg}}{(0.086094 - 0.001190) \text{ m}^3/\text{kg}} = 0.4185$$

$$h_2 = h_f + x_2 h_{fg} = 943.55 + (0.4185)(1857.4) = \mathbf{1720.9 \text{ kJ/kg}}$$



4-44E The local atmospheric pressure, and thus the boiling temperature, changes with the weather conditions. The change in the boiling temperature corresponding to a change of 0.3 in of mercury in atmospheric pressure is to be determined.

Properties The saturation pressures of water at 200 and 212°F are 11.538 and 14.709 psia, respectively (Table A-4E). One in. of mercury is equivalent to 1 inHg = 3.387 kPa = 0.491 psia (inner cover page).

Analysis A change of 0.3 in of mercury in atmospheric pressure corresponds to

$$\Delta P = (0.3 \text{ inHg}) \left(\frac{0.491 \text{ psia}}{1 \text{ inHg}} \right) = 0.147 \text{ psia}$$

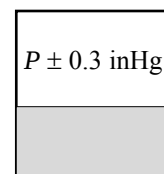
At about boiling temperature, the change in boiling temperature per 1 psia change in pressure is determined using data at 200 and 212°F to be

$$\frac{\Delta T}{\Delta P} = \frac{(212 - 200)^\circ\text{F}}{(14.709 - 11.538) \text{ psia}} = 3.783^\circ\text{F/psia}$$

Then the change in saturation (boiling) temperature corresponding to a change of 0.147 psia becomes

$$\Delta T_{\text{boiling}} = (3.783^\circ\text{F/psia})\Delta P = (3.783^\circ\text{F/psia})(0.147 \text{ psia}) = \mathbf{0.56^\circ\text{F}}$$

which is very small. Therefore, the effect of variation of atmospheric pressure on the boiling temperature is negligible.



4-45 A person cooks a meal in a pot that is covered with a well-fitting lid, and leaves the food to cool to the room temperature. It is to be determined if the lid will open or the pan will move up together with the lid when the person attempts to open the pan by lifting the lid up.

Assumptions 1 The local atmospheric pressure is 1 atm = 101.325 kPa. 2 The weight of the lid is small and thus its effect on the boiling pressure and temperature is negligible. 3 No air has leaked into the pan during cooling.

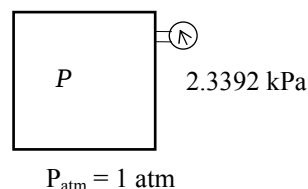
Properties The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4).

Analysis Noting that the weight of the lid is negligible, the reaction force F on the lid after cooling at the pan-lid interface can be determined from a force balance on the lid in the vertical direction to be

$$PA + F = P_{\text{atm}}A$$

or,

$$\begin{aligned} F &= A(P_{\text{atm}} - P) = (\pi D^2 / 4)(P_{\text{atm}} - P) \\ &= \frac{\pi(0.3 \text{ m})^2}{4}(101,325 - 2339.2) \text{ Pa} \\ &= 6997 \text{ m}^2\text{Pa} = \mathbf{6997 \text{ N}} \quad (\text{since } 1 \text{ Pa} = 1 \text{ N/m}^2) \end{aligned}$$



The weight of the pan and its contents is

$$W = mg = (8 \text{ kg})(9.81 \text{ m/s}^2) = \mathbf{78.5 \text{ N}}$$

which is much less than the reaction force of 6997 N at the pan-lid interface. Therefore, the pan will **move up** together with the lid when the person attempts to open the pan by lifting the lid up. In fact, it looks like the lid will not open even if the mass of the pan and its contents is several hundred kg.

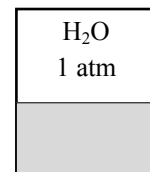
4-46 Water is boiled at 1 atm pressure in a pan placed on an electric burner. The water level drops by 10 cm in 45 min during boiling. The rate of heat transfer to the water is to be determined.

Properties The properties of water at 1 atm and thus at a saturation temperature of $T_{\text{sat}} = 100^\circ\text{C}$ are $h_{\text{fg}} = 2256.5 \text{ kJ/kg}$ and $\nu_f = 0.001043 \text{ m}^3/\text{kg}$ (Table A-4).

Analysis The rate of evaporation of water is

$$m_{\text{evap}} = \frac{V_{\text{evap}}}{\nu_f} = \frac{(\pi D^2 / 4)L}{\nu_f} = \frac{[\pi(0.25 \text{ m})^2 / 4](0.10 \text{ m})}{0.001043} = 4.704 \text{ kg}$$

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{4.704 \text{ kg}}{45 \times 60 \text{ s}} = 0.001742 \text{ kg/s}$$



Then the rate of heat transfer to water becomes

$$\dot{Q} = \dot{m}_{\text{evap}} h_{\text{fg}} = (0.001742 \text{ kg/s})(2256.5 \text{ kJ/kg}) = \mathbf{3.93 \text{ kW}}$$

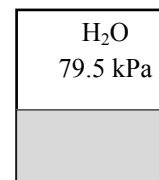
4-47 Water is boiled at a location where the atmospheric pressure is 79.5 kPa in a pan placed on an electric burner. The water level drops by 10 cm in 45 min during boiling. The rate of heat transfer to the water is to be determined.

Properties The properties of water at 79.5 kPa are $T_{\text{sat}} = 93.3^\circ\text{C}$, $h_{\text{fg}} = 2273.9 \text{ kJ/kg}$ and $\nu_f = 0.001038 \text{ m}^3/\text{kg}$ (Table A-5).

Analysis The rate of evaporation of water is

$$m_{\text{evap}} = \frac{V_{\text{evap}}}{\nu_f} = \frac{(\pi D^2 / 4)L}{\nu_f} = \frac{[\pi(0.25 \text{ m})^2 / 4](0.10 \text{ m})}{0.001038} = 4.727 \text{ kg}$$

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{4.727 \text{ kg}}{45 \times 60 \text{ s}} = 0.001751 \text{ kg/s}$$



Then the rate of heat transfer to water becomes

$$\dot{Q} = \dot{m}_{\text{evap}} h_{\text{fg}} = (0.001751 \text{ kg/s})(2273.9 \text{ kJ/kg}) = \mathbf{3.98 \text{ kW}}$$

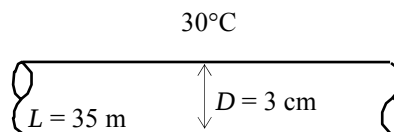
4-48 Saturated steam at $T_{\text{sat}} = 30^\circ\text{C}$ condenses on the outer surface of a cooling tube at a rate of 45 kg/h. The rate of heat transfer from the steam to the cooling water is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The condensate leaves the condenser as a saturated liquid at 30°C .

Properties The properties of water at the saturation temperature of 30°C are $h_{fg} = 2429.8 \text{ kJ/kg}$ (Table A-4).

Analysis Noting that 2429.8 kJ of heat is released as 1 kg of saturated vapor at 30°C condenses, the rate of heat transfer from the steam to the cooling water in the tube is determined directly from

$$\begin{aligned}\dot{Q} &= \dot{m}_{\text{evap}} h_{fg} \\ &= (45 \text{ kg/h})(2429.8 \text{ kJ/kg}) = 109,341 \text{ kJ/h} \\ &= \mathbf{30.4 \text{ kW}}\end{aligned}$$



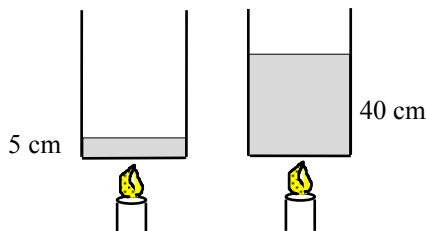
4-49 The boiling temperature of water in a 5-cm deep pan is given. The boiling temperature in a 40-cm deep pan is to be determined.

Assumptions Both pans are full of water.

Properties The density of liquid water is approximately $\rho = 1000 \text{ kg/m}^3$.

Analysis The pressure at the bottom of the 5-cm pan is the saturation pressure corresponding to the boiling temperature of 98°C :

$$P = P_{\text{sat}@98^\circ\text{C}} = 94.39 \text{ kPa} \quad (\text{Table A-4})$$



The pressure difference between the bottoms of two pans is

$$\Delta P = \rho g h = (1000 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.35 \text{ m}) \left(\frac{1 \text{ kPa}}{1000 \text{ kg/m} \cdot \text{s}^2} \right) = 3.43 \text{ kPa}$$

Then the pressure at the bottom of the 40-cm deep pan is

$$P = 94.39 + 3.43 = 97.82 \text{ kPa}$$

Then the boiling temperature becomes

$$T_{\text{boiling}} = T_{\text{sat}@97.82 \text{ kPa}} = \mathbf{99.0^\circ\text{C}} \quad (\text{Table A-5})$$

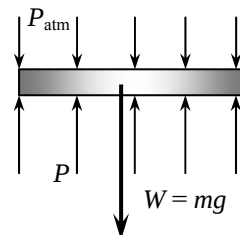
4-50 A vertical piston-cylinder device is filled with water and covered with a 20-kg piston that serves as the lid. The boiling temperature of water is to be determined.

Analysis The pressure in the cylinder is determined from a force balance on the piston,

$$PA = P_{\text{atm}}A + W$$

or,

$$\begin{aligned} P &= P_{\text{atm}} + \frac{mg}{A} \\ &= (100 \text{ kPa}) + \frac{(20 \text{ kg})(9.81 \text{ m/s}^2)}{0.01 \text{ m}^2} \left(\frac{1 \text{ kPa}}{1000 \text{ kg/m} \cdot \text{s}^2} \right) \\ &= 119.61 \text{ kPa} \end{aligned}$$



The boiling temperature is the saturation temperature corresponding to this pressure,

$$T = T_{\text{sat}@119.61 \text{ kPa}} = \mathbf{104.7^\circ\text{C}} \quad (\text{Table A-5})$$

4-51 A rigid tank that is filled with saturated liquid-vapor mixture is heated. The temperature at which the liquid in the tank is completely vaporized is to be determined, and the T - ν diagram is to be drawn.

Analysis This is a constant volume process ($\nu = V/m = \text{constant}$),

and the specific volume is determined to be

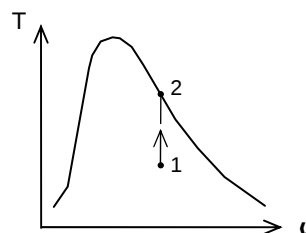
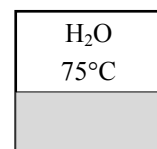
$$\nu = \frac{V}{m} = \frac{2.5 \text{ m}^3}{15 \text{ kg}} = 0.1667 \text{ m}^3/\text{kg}$$

When the liquid is completely vaporized the tank will contain saturated vapor only. Thus,

$$\nu_2 = \nu_g = 0.1667 \text{ m}^3/\text{kg}$$

The temperature at this point is the temperature that corresponds to this ν_g value,

$$T = T_{\text{sat}@}\nu_g=0.1667 \text{ m}^3/\text{kg}} = \mathbf{187.0^\circ\text{C}} \quad (\text{Table A-4})$$



4-52 A rigid vessel is filled with refrigerant-134a. The total volume and the total internal energy are to be determined.

Properties The properties of R-134a at the given state are (Table A-13).

$$\left. \begin{array}{l} P = 800 \text{ kPa} \\ T = 120^\circ \text{C} \end{array} \right\} \begin{array}{l} u = 327.87 \text{ kJ/kg} \\ \nu = 0.037625 \text{ m}^3/\text{kg} \end{array}$$

R-134a
2 kg
800 kPa
120°C

Analysis The total volume and internal energy are determined from

$$\nu = m\nu = (2 \text{ kg})(0.037625 \text{ m}^3/\text{kg}) = \mathbf{0.0753 \text{ m}^3}$$

$$U = mu = (2 \text{ kg})(327.87 \text{ kJ/kg}) = \mathbf{655.7 \text{ kJ}}$$

4-53 A rigid vessel contains R-134a at specified temperature. The pressure, total internal energy, and the volume of the liquid phase are to be determined.

Analysis (a) The specific volume of the refrigerant is

$$\nu = \frac{V}{m} = \frac{0.5 \text{ m}^3}{10 \text{ kg}} = 0.05 \text{ m}^3/\text{kg}$$

At -20°C , $\nu_f = 0.0007362 \text{ m}^3/\text{kg}$ and $\nu_g = 0.14729 \text{ m}^3/\text{kg}$ (Table A-11). Thus the tank contains saturated liquid-vapor mixture since $\nu_f < \nu < \nu_g$, and the pressure must be the saturation pressure at the specified temperature,

$$P = P_{\text{sat}@-20^\circ\text{C}} = \mathbf{132.82 \text{ kPa}}$$

R-134a
10 kg
-20°C

(b) The quality of the refrigerant-134a and its total internal energy are determined from

$$x = \frac{\nu - \nu_f}{\nu_{fg}} = \frac{0.05 - 0.0007362}{0.14729 - 0.0007362} = 0.3361$$

$$u = u_f + xu_{fg} = 25.39 + 0.3361 \times 193.45 = 90.42 \text{ kJ/kg}$$

$$U = mu = (10 \text{ kg})(90.42 \text{ kJ/kg}) = \mathbf{904.2 \text{ kJ}}$$

(c) The mass of the liquid phase and its volume are determined from

$$m_f = (1 - x)m_t = (1 - 0.3361) \times 10 = 6.639 \text{ kg}$$

$$\nu_f = m_f \nu_f = (6.639 \text{ kg})(0.0007362 \text{ m}^3/\text{kg}) = \mathbf{0.00489 \text{ m}^3}$$

4-54 CD EES A piston-cylinder device contains a saturated liquid-vapor mixture of water at 800 kPa pressure. The mixture is heated at constant pressure until the temperature rises to 350°C. The initial temperature, the total mass of water, the final volume are to be determined, and the P - v diagram is to be drawn.

Analysis (a) Initially two phases coexist in equilibrium, thus we have a saturated liquid-vapor mixture. Then the temperature in the tank must be the saturation temperature at the specified pressure,

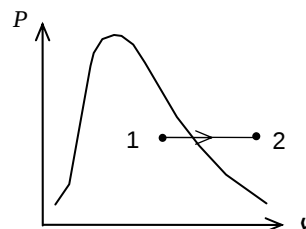
$$T = T_{\text{sat}@800 \text{ kPa}} = \mathbf{170.41^\circ\text{C}}$$

(b) The total mass in this case can easily be determined by adding the mass of each phase,

$$m_f = \frac{V_f}{v_f} = \frac{0.1 \text{ m}^3}{0.001115 \text{ m}^3/\text{kg}} = 89.704 \text{ kg}$$

$$m_g = \frac{V_g}{v_g} = \frac{0.9 \text{ m}^3}{0.24035 \text{ m}^3/\text{kg}} = 3.745 \text{ kg}$$

$$m_t = m_f + m_g = 89.704 + 3.745 = \mathbf{93.45 \text{ kg}}$$



(c) At the final state water is superheated vapor, and its specific volume is

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 350^\circ\text{C} \end{array} \right\} v_2 = 0.35442 \text{ m}^3/\text{kg} \quad (\text{Table A-6})$$

Then,

$$V_2 = m_t v_2 = (93.45 \text{ kg})(0.35442 \text{ m}^3/\text{kg}) = \mathbf{33.12 \text{ m}^3}$$

4-55 EES Problem 4-54 is reconsidered. The effect of pressure on the total mass of water in the tank as the pressure varies from 0.1 MPa to 1 MPa is to be investigated. The total mass of water is to be plotted against pressure, and results are to be discussed.

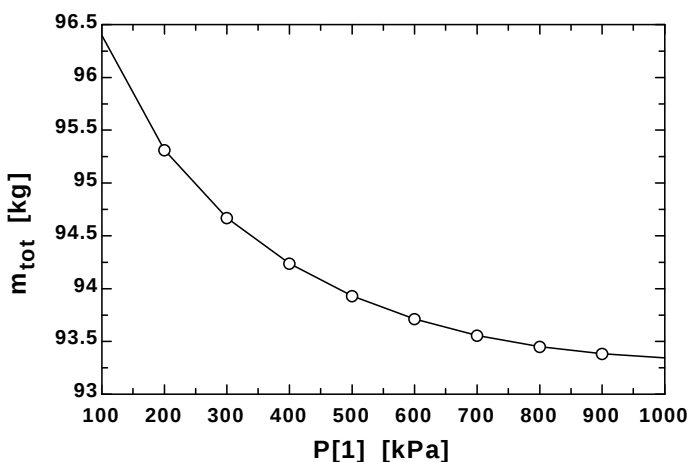
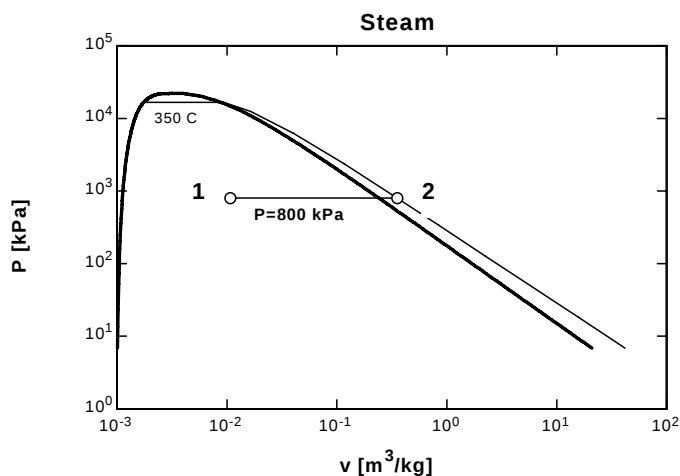
Analysis The problem is solved using EES, and the solution is given below.

```

P[1]=800 [kPa]
P[2]=P[1]
T[2]=350 [C]
V_f1 = 0.1 [m^3]
V_g1=0.9 [m^3]
spvsat_f1=volume(Steam_iapws, P=P[1],x=0) "sat. liq. specific volume, m^3/kg"
spvsat_g1=volume(Steam_iapws,P=P[1],x=1) "sat. vap. specific volume, m^3/kg"
m_f1=V_f1/spvsat_f1 "sat. liq. mass, kg"
m_g1=V_g1/spvsat_g1 "sat. vap. mass, kg"
m_tot=m_f1+m_g1
V[1]=V_f1+V_g1
spvol[1]=V[1]/m_tot "specific volume1, m^3"
T[1]=temperature(Steam_iapws, P=P[1],v=spvol[1])"C"
"The final volume is calculated from the specific volume at the final T and P"
spvol[2]=volume(Steam_iapws, P=P[2], T=T[2]) "specific volume2, m^3/kg"
V[2]=m_tot*spvol[2]

```

m_{tot} [kg]	P_1 [kPa]
96.39	100
95.31	200
94.67	300
94.24	400
93.93	500
93.71	600
93.56	700
93.45	800
93.38	900
93.34	1000



4-56E Superheated water vapor cools at constant volume until the temperature drops to 250°F. At the final state, the pressure, the quality, and the enthalpy are to be determined.

Analysis This is a constant volume process ($\nu = V/m = \text{constant}$), and the initial specific volume is determined to be

$$\left. \begin{array}{l} P_1 = 180 \text{ psia} \\ T_1 = 500^\circ \text{F} \end{array} \right\} \nu_1 = 3.0433 \text{ ft}^3/\text{lbm} \quad (\text{Table A-6E})$$

At 250°F, $\nu_f = 0.01700 \text{ ft}^3/\text{lbm}$ and $\nu_g = 13.816 \text{ ft}^3/\text{lbm}$. Thus at the final state, the tank will contain saturated liquid-vapor mixture since $\nu_f < \nu < \nu_g$, and the final pressure must be the saturation pressure at the final temperature,

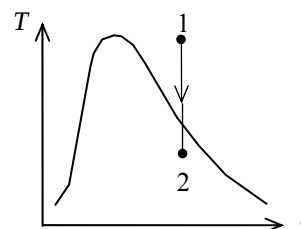
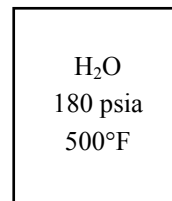
$$P = P_{\text{sat}@250^\circ \text{F}} = \mathbf{29.84 \text{ psia}}$$

(b) The quality at the final state is determined from

$$x_2 = \frac{\nu_2 - \nu_f}{\nu_{fg}} = \frac{3.0433 - 0.01700}{13.816 - 0.01700} = \mathbf{0.219}$$

(c) The enthalpy at the final state is determined from

$$h = h_f + xh_{fg} = 218.63 + 0.219 \times 945.41 = \mathbf{426.0 \text{ Btu/lbm}}$$



4-57E EES Problem 4-56E is reconsidered. The effect of initial pressure on the quality of water at the final state as the pressure varies from 100 psi to 300 psi is to be investigated. The quality is to be plotted against initial pressure, and the results are to be discussed.

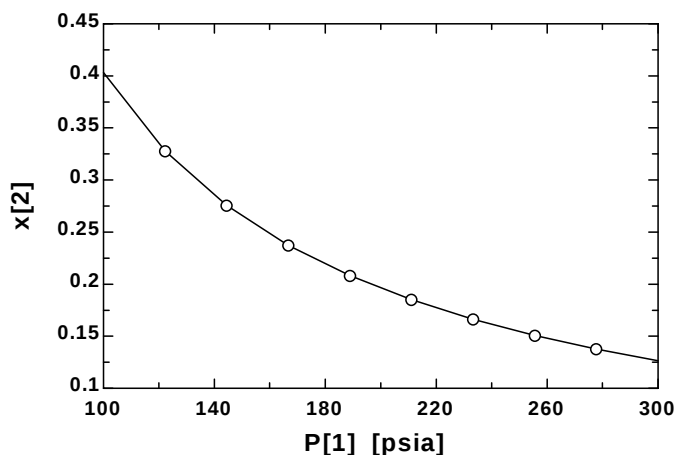
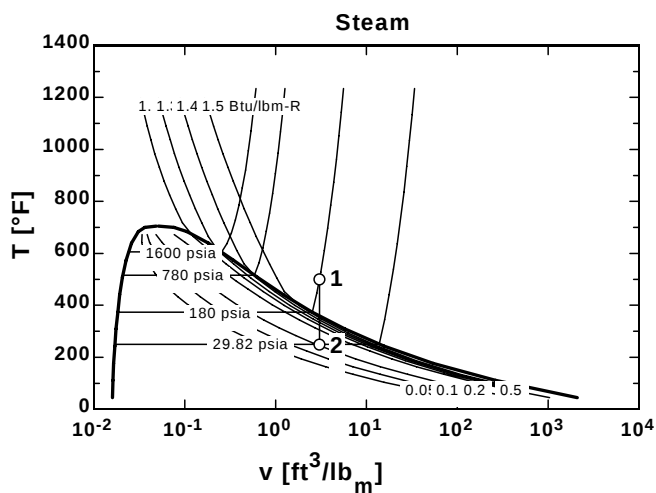
Analysis The problem is solved using EES, and the solution is given below.

```

T[1]=500 [F]
P[1]=180 [psia]
T[2]=250 [F]
v[1]=volume(steam_iapws,T=T[1],P=P[1])
v[2]=v[1]
P[2]=pressure(steam_iapws,T=T[2],v=v[2])
h[2]=enthalpy(steam_iapws,T=T[2],v=v[2])
x[2]=quality(steam_iapws,T=T[2],v=v[2])

```

P_1 [psia]	x_2
100	0.4037
122.2	0.3283
144.4	0.2761
166.7	0.2378
188.9	0.2084
211.1	0.1853
233.3	0.1665
255.6	0.1510
277.8	0.1379
300	0.1268



4-58 A rigid vessel that contains a saturated liquid-vapor mixture is heated until it reaches the critical state. The mass of the liquid water and the volume occupied by the liquid at the initial state are to be determined.

Analysis This is a constant volume process ($\nu = V/m = \text{constant}$) to the critical state, and thus the initial specific volume will be equal to the final specific volume, which is equal to the critical specific volume of water,

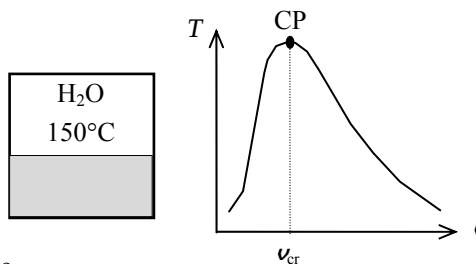
$$\nu_1 = \nu_2 = \nu_{cr} = 0.003106 \text{ m}^3/\text{kg} \quad (\text{last row of Table A-4})$$

The total mass is

$$m = \frac{V}{\nu} = \frac{0.3 \text{ m}^3}{0.003106 \text{ m}^3/\text{kg}} = 96.60 \text{ kg}$$

At 150°C , $\nu_f = 0.001091 \text{ m}^3/\text{kg}$ and $\nu_g = 0.39248 \text{ m}^3/\text{kg}$ (Table A-4). Then the quality of water at the initial state is

$$x_1 = \frac{\nu_1 - \nu_f}{\nu_{fg}} = \frac{0.003106 - 0.001091}{0.39248 - 0.001091} = 0.005149$$



Then the mass of the liquid phase and its volume at the initial state are determined from

$$m_f = (1 - x_1)m_t = (1 - 0.005149)(96.60) = \mathbf{96.10 \text{ kg}}$$

$$V_f = m_f \nu_f = (96.10 \text{ kg})(0.001091 \text{ m}^3/\text{kg}) = \mathbf{0.105 \text{ m}^3}$$

4-59 The properties of compressed liquid water at a specified state are to be determined using the compressed liquid tables, and also by using the saturated liquid approximation, and the results are to be compared.

Analysis Compressed liquid can be approximated as saturated liquid at the given temperature. Then from Table A-4,

$$T = 100^\circ\text{C} \Rightarrow \nu \cong \nu_f @ 100^\circ\text{C} = 0.001043 \text{ m}^3/\text{kg} \quad (0.72\% \text{ error})$$

$$u \cong u_f @ 100^\circ\text{C} = 419.06 \text{ kJ/kg} \quad (1.02\% \text{ error})$$

$$h \cong h_f @ 100^\circ\text{C} = 419.17 \text{ kJ/kg} \quad (2.61\% \text{ error})$$

From compressed liquid table (Table A-7),

$$\left. \begin{array}{l} P = 15 \text{ MPa} \\ T = 100^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu = 0.001036 \text{ m}^3/\text{kg} \\ u = 414.85 \text{ kJ/kg} \\ h = 430.39 \text{ kJ/kg} \end{array}$$

The percent errors involved in the saturated liquid approximation are listed above in parentheses.

4-60 EES Problem 4-59 is reconsidered. Using EES, the indicated properties of compressed liquid are to be determined, and they are to be compared to those obtained using the saturated liquid approximation.

Analysis The problem is solved using EES, and the solution is given below.

```
Fluid$='Steam_IAPWS'
T = 100 [C]
P = 15000 [kPa]
v = VOLUME(Fluid$,T=T,P=P)
u = INTENERGY(Fluid$,T=T,P=P)
h = ENTHALPY(Fluid$,T=T,P=P)
v_app = VOLUME(Fluid$,T=T,x=0)
u_app = INTENERGY(Fluid$,T=T,x=0)
h_app_1 = ENTHALPY(Fluid$,T=T,x=0)
h_app_2 = ENTHALPY(Fluid$,T=T,x=0)+v_app*(P-pressure(Fluid$,T=T,x=0))
```

SOLUTION

```
Fluid$='Steam_IAPWS'
h=430.4 [kJ/kg]
h_app_1=419.2 [kJ/kg]
h_app_2=434.7 [kJ/kg]
P=15000 [kPa]
T=100 [C]
u=414.9 [kJ/kg]
u_app=419.1 [kJ/kg]
v=0.001036 [m^3/kg]
v_app=0.001043 [m^3/kg]
```

4-61 Superheated steam in a piston-cylinder device is cooled at constant pressure until half of the mass condenses. The final temperature and the volume change are to be determined, and the process should be shown on a T - v diagram.

Analysis (b) At the final state the cylinder contains saturated liquid-vapor mixture, and thus the final temperature must be the saturation temperature at the final pressure,

$$T = T_{\text{sat}@1 \text{ MPa}} = \mathbf{179.88^\circ\text{C}} \quad (\text{Table A-5})$$

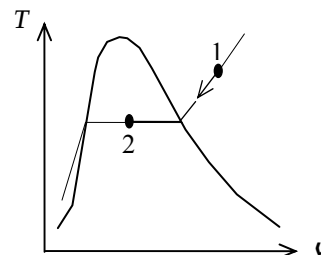
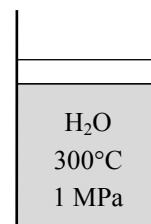
(c) The quality at the final state is specified to be $x_2 = 0.5$. The specific volumes at the initial and the final states are

$$\left. \begin{array}{l} P_1 = 1.0 \text{ MPa} \\ T_1 = 300^\circ\text{C} \end{array} \right\} v_1 = 0.25799 \text{ m}^3/\text{kg} \quad (\text{Table A-6})$$

$$\left. \begin{array}{l} P_2 = 1.0 \text{ MPa} \\ x_2 = 0.5 \end{array} \right\} \begin{aligned} v_2 &= v_f + x_2 v_{fg} \\ &= 0.001127 + 0.5 \times (0.19436 - 0.001127) \\ &= 0.09775 \text{ m}^3/\text{kg} \end{aligned}$$

Thus,

$$\Delta V = m(v_2 - v_1) = (0.8 \text{ kg})(0.09775 - 0.25799) \text{ m}^3/\text{kg} = \mathbf{-0.1282 \text{ m}^3}$$



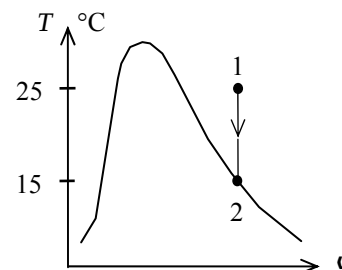
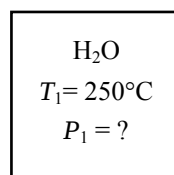
4-62 The water in a rigid tank is cooled until the vapor starts condensing. The initial pressure in the tank is to be determined.

Analysis This is a constant volume process ($\nu = V/m = \text{constant}$), and the initial specific volume is equal to the final specific volume that is

$$\nu_1 = \nu_2 = \nu_{g@150^\circ\text{C}} = 0.39248 \text{ m}^3/\text{kg} \quad (\text{Table A-4})$$

since the vapor starts condensing at 150°C . Then from Table A-6,

$$\left. \begin{array}{l} T_1 = 250^\circ\text{C} \\ \nu_1 = 0.39248 \text{ m}^3/\text{kg} \end{array} \right\} P_1 = \mathbf{0.60 \text{ MPa}}$$



4-63 Heat is supplied to a piston-cylinder device that contains water at a specified state. The volume of the tank, the final temperature and pressure, and the internal energy change of water are to be determined.

Properties The saturated liquid properties of water at 200°C are: $\nu_f = 0.001157 \text{ m}^3/\text{kg}$ and $u_f = 850.46 \text{ kJ/kg}$ (Table A-4).

Analysis (a) The cylinder initially contains saturated liquid water. The volume of the cylinder at the initial state is

$$\nu_1 = m\nu_1 = (1.4 \text{ kg})(0.001157 \text{ m}^3/\text{kg}) = 0.001619 \text{ m}^3$$

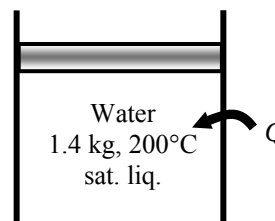
The volume at the final state is

$$\nu = 4(0.001619) = \mathbf{0.006476 \text{ m}^3}$$

(b) The final state properties are

$$\nu_2 = \frac{\nu}{m} = \frac{0.006476 \text{ m}^3}{1.4 \text{ kg}} = 0.004626 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} \nu_2 = 0.004626 \text{ m}^3/\text{kg} \\ x_2 = 1 \end{array} \right\} \begin{array}{l} T_2 = \mathbf{371.3^\circ\text{C}} \\ P_2 = \mathbf{21,367 \text{ kPa}} \\ u_2 = 2201.5 \text{ kJ/kg} \end{array} \quad (\text{Table A-4 or A-5 or EES})$$



(c) The total internal energy change is determined from

$$\Delta U = m(u_2 - u_1) = (1.4 \text{ kg})(2201.5 - 850.46) \text{ kJ/kg} = \mathbf{1892 \text{ kJ}}$$

4-64 Heat is lost from a piston-cylinder device that contains steam at a specified state. The initial temperature, the enthalpy change, and the final pressure and quality are to be determined.

Analysis (a) The saturation temperature of steam at 3.5 MPa is

$$T_{\text{sat}@3.5 \text{ MPa}} = 242.6^\circ\text{C} \quad (\text{Table A-5})$$

Then, the initial temperature becomes

$$T_1 = 242.6 + 5 = \mathbf{247.6^\circ\text{C}}$$

Also,
$$\left. \begin{array}{l} P_1 = 3.5 \text{ MPa} \\ T_1 = 247.6^\circ\text{C} \end{array} \right\} h_1 = 2821.1 \text{ kJ/kg} \quad (\text{Table A-6})$$

(b) The properties of steam when the piston first hits the stops are

$$\left. \begin{array}{l} P_2 = P_1 = 3.5 \text{ MPa} \\ x_2 = 0 \end{array} \right\} \begin{array}{l} h_2 = 1049.7 \text{ kJ/kg} \\ v_2 = 0.001235 \text{ m}^3/\text{kg} \end{array} \quad (\text{Table A-5})$$

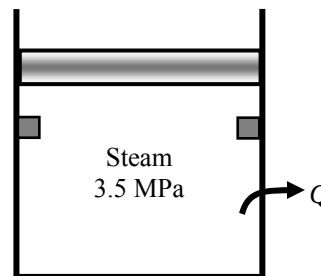
Then, the enthalpy change of steam becomes

$$\Delta h = h_2 - h_1 = 1049.7 - 2821.1 = \mathbf{-1771 \text{ kJ/kg}}$$

(c) At the final state

$$\left. \begin{array}{l} v_3 = v_2 = 0.001235 \text{ m}^3/\text{kg} \\ T_3 = 200^\circ\text{C} \end{array} \right\} \begin{array}{l} P_3 = \mathbf{1555 \text{ kPa}} \\ x_3 = \mathbf{0.0006} \end{array} \quad (\text{Table A-4 or EES})$$

The cylinder contains saturated liquid-vapor mixture with a small mass of vapor at the final state.



4-65E The error involved in using the enthalpy of water by the incompressible liquid approximation is to be determined.

Analysis The state of water is compressed liquid. From the steam tables,

$$\left. \begin{array}{l} P = 1500 \text{ psia} \\ T = 400^\circ\text{F} \end{array} \right\} h = 376.51 \text{ Btu/lbm} \quad (\text{Table A - 7E})$$

Based upon the incompressible liquid approximation,

$$\left. \begin{array}{l} P = 1500 \text{ psia} \\ T = 400^\circ\text{F} \end{array} \right\} h \cong h_f @ 400^\circ\text{F} = 375.04 \text{ Btu/lbm} \quad (\text{Table A - 4E})$$

The error involved is

$$\text{Percent Error} = \frac{376.51 - 375.04}{376.51} \times 100 = \mathbf{0.39\%}$$

which is quite acceptable in most engineering calculations.

4-66 The errors involved in using the specific volume and enthalpy of water by the incompressible liquid approximation are to be determined.

Analysis The state of water is compressed liquid. From the steam tables,

$$\left. \begin{array}{l} P = 10 \text{ MPa} \\ T = 100^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu = 0.0010385 \text{ m}^3/\text{kg} \\ h = 426.62 \text{ kJ/kg} \end{array} \quad (\text{Table A - 7})$$

Based upon the incompressible liquid approximation,

$$\left. \begin{array}{l} P = 10 \text{ MPa} \\ T = 100^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu \cong \nu_f @ 100^\circ\text{C} = 0.001043 \text{ m}^3/\text{kg} \\ h \cong h_f @ 100^\circ\text{C} = 419.17 \text{ kJ/kg} \end{array} \quad (\text{Table A - 4})$$

The errors involved are

$$\text{Percent Error (specific volume)} = \frac{0.001043 - 0.0010385}{0.0010385} \times 100 = \mathbf{0.43\%}$$

$$\text{Percent Error (enthalpy)} = \frac{426.62 - 419.17}{426.62} \times 100 = \mathbf{1.75\%}$$

which are quite acceptable in most engineering calculations.

4-67 The specific volume and internal energy of R-134a at a specified state are to be determined.

Analysis The state of R-134a is compressed liquid. Based upon the incompressible liquid approximation,

$$\left. \begin{array}{l} P = 700 \text{ kPa} \\ T = 20^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu \cong \nu_f @ 20^\circ\text{C} = 0.0008161 \text{ m}^3/\text{kg} \\ u \cong u_f @ 20^\circ\text{C} = 78.86 \text{ kJ/kg} \end{array} \quad (\text{Table A - 11})$$

4-68 A piston-cylinder device that is filled with R-134a is heated. The volume change is to be determined.

Analysis The initial specific volume is

$$\left. \begin{array}{l} P_1 = 60 \text{ kPa} \\ T_1 = -20^\circ\text{C} \end{array} \right\} \nu_1 = 0.33608 \text{ m}^3/\text{kg} \text{ (Table A-13)}$$

and the initial volume is

$$V_1 = m\nu_1 = (0.100 \text{ kg})(0.33608 \text{ m}^3/\text{kg}) = 0.033608 \text{ m}^3$$

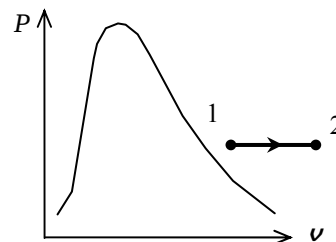
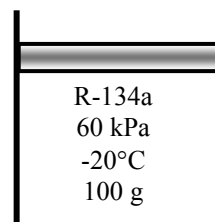
At the final state, we have

$$\left. \begin{array}{l} P_2 = 60 \text{ kPa} \\ T_2 = 100^\circ\text{C} \end{array} \right\} \nu_2 = 0.50410 \text{ m}^3/\text{kg} \text{ (Table A-13)}$$

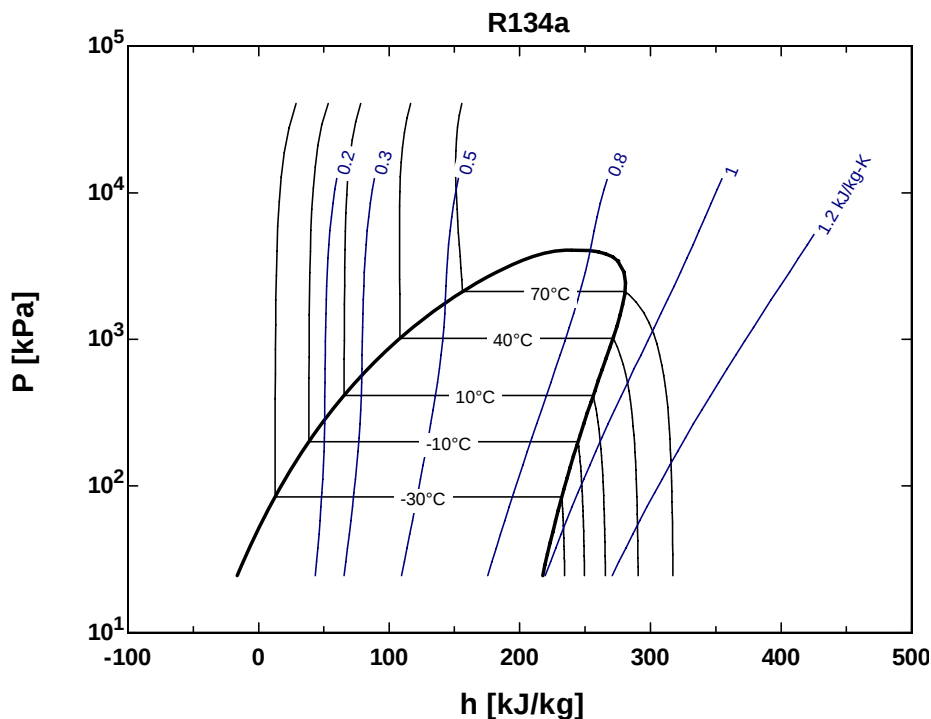
$$V_2 = m\nu_2 = (0.100 \text{ kg})(0.50410 \text{ m}^3/\text{kg}) = 0.050410 \text{ m}^3$$

The volume change is then

$$\Delta V = V_2 - V_1 = 0.050410 - 0.033608 = \mathbf{0.0168 \text{ m}^3}$$



4-69 EES The Pressure-Enthalpy diagram of R-134a showing some constant-temperature and constant-entropy lines are obtained using Property Plot feature of EES.



Ideal Gas

4-70C Propane (molar mass = 44.1 kg/kmol) poses a greater fire danger than methane (molar mass = 16 kg/kmol) since propane is heavier than air (molar mass = 29 kg/kmol), and it will settle near the floor. Methane, on the other hand, is lighter than air and thus it will rise and leak out.

4-71C A gas can be treated as an ideal gas when it is at a high temperature or low pressure relative to its critical temperature and pressure.

4-72C R_u is the universal gas constant that is the same for all gases whereas R is the specific gas constant that is different for different gases. These two are related to each other by $R = R_u / M$, where M is the molar mass of the gas.

4-73C Mass m is simply the amount of matter; molar mass M is the mass of one mole in grams or the mass of one kmol in kilograms. These two are related to each other by $m = NM$, where N is the number of moles.

4-74E The specific volume of oxygen at a specified state is to be determined.

Assumptions At specified conditions, oxygen behaves as an ideal gas.

Properties The gas constant of oxygen is $R = 0.3353 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R}$ (Table A-1E).

Analysis According to the ideal gas equation of state,

$$\nu = \frac{RT}{P} = \frac{(0.3353 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R})(80 + 460 \text{ R})}{25 \text{ psia}} = \mathbf{7.242 \text{ ft}^3 / \text{lbm}}$$

4-75 The pressure in a container that is filled with air is to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg} \cdot \text{K}$ (Table A-1).

Analysis The definition of the specific volume gives

$$\nu = \frac{V}{m} = \frac{0.100 \text{ m}^3}{1 \text{ kg}} = 0.100 \text{ m}^3 / \text{kg}$$

Using the ideal gas equation of state, the pressure is

$$P = \frac{RT}{\nu} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3 / \text{kg} \cdot \text{K})(27 + 273 \text{ K})}{0.100 \text{ m}^3 / \text{kg}} = \mathbf{861 \text{ kPa}}$$

4-76E The volume of a tank that is filled with argon at a specified state is to be determined.

Assumptions At specified conditions, argon behaves as an ideal gas.

Properties The gas constant of argon is $R = 0.2686 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$ (Table A-1E)

Analysis According to the ideal gas equation of state,

$$\nu = \frac{mRT}{P} = \frac{(1 \text{ lbm})(0.2686 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(100 + 460 \text{ R})}{200 \text{ psia}} = \mathbf{0.7521 \text{ ft}^3}$$

4-77 A balloon is filled with helium gas. The mole number and the mass of helium in the balloon are to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The universal gas constant is $R_u = 8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K}$. The molar mass of helium is 4.0 kg/kmol (Table A-1).

Analysis The volume of the sphere is

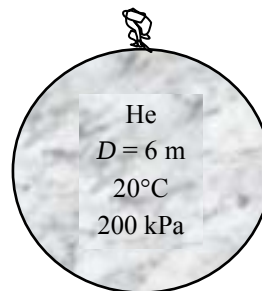
$$\nu = \frac{4}{3} \pi r^3 = \frac{4}{3} \pi (3 \text{ m})^3 = 113.1 \text{ m}^3$$

Assuming ideal gas behavior, the mole numbers of He is determined from

$$N = \frac{P\nu}{R_u T} = \frac{(200 \text{ kPa})(113.1 \text{ m}^3)}{(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(293 \text{ K})} = \mathbf{9.28 \text{ kmol}}$$

Then the mass of He can be determined from

$$m = NM = (9.28 \text{ kmol})(4.0 \text{ kg/kmol}) = \mathbf{37.15 \text{ kg}}$$



4-78 EES Problem 4-77 is to be reconsidered. The effect of the balloon diameter on the mass of helium contained in the balloon is to be determined for the pressures of (a) 100 kPa and (b) 200 kPa as the diameter varies from 5 m to 15 m. The mass of helium is to be plotted against the diameter for both cases.

Analysis The problem is solved using EES, and the solution is given below.

"Given Data"

{D=6 [m]}

{P=200 [kPa]}

T=20 [C]

P=200 [kPa]

R_u=8.314 [kJ/kmol-K]

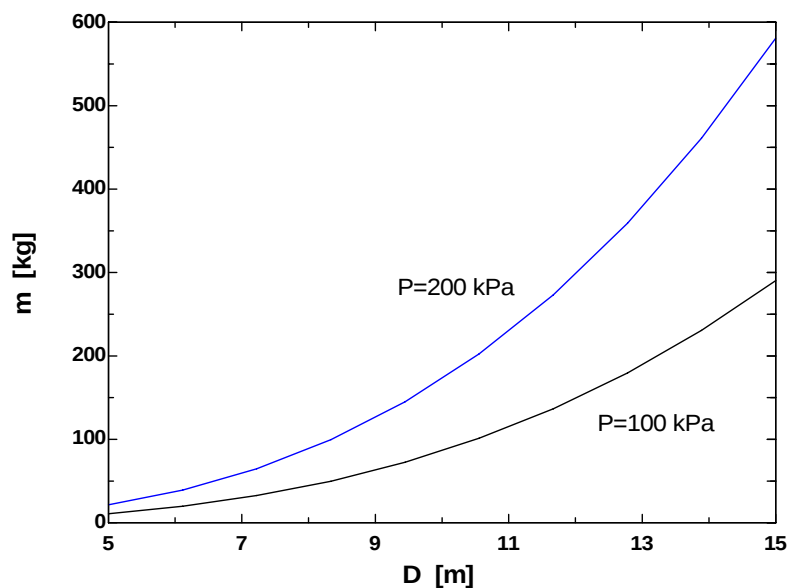
"Solution"

$P \cdot V = N \cdot R_u \cdot (T + 273)$

$V = 4 \cdot \pi \cdot (D/2)^3 / 3$

$m = N \cdot \text{MOLARMASS}(\text{Helium})$

D [m]	m [kg]
5	21.51
6.111	39.27
7.222	64.82
8.333	99.57
9.444	145
10.56	202.4
11.67	273.2
12.78	359
13.89	461
15	580.7



4-79 An automobile tire is inflated with air. The pressure rise of air in the tire when the tire is heated and the amount of air that must be bled off to reduce the temperature to the original value are to be determined.

Assumptions 1 At specified conditions, air behaves as an ideal gas. 2 The volume of the tire remains constant.

Properties The gas constant of air is $R = 0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).

Analysis Initially, the absolute pressure in the tire is

$$P_1 = P_g + P_{\text{atm}} = 210 + 100 = 310 \text{ kPa}$$

Treating air as an ideal gas and assuming the volume of the tire to remain constant, the final pressure in the tire can be determined from

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{323 \text{ K}}{298 \text{ K}} (310 \text{ kPa}) = 336 \text{ kPa}$$

Thus the pressure rise is

$$\Delta P = P_2 - P_1 = 336 - 310 = \mathbf{26 \text{ kPa}}$$

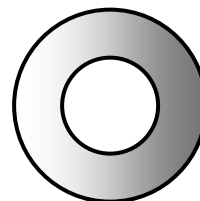
The amount of air that needs to be bled off to restore pressure to its original value is

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})} = 0.0906 \text{ kg}$$

$$m_2 = \frac{P_1 V}{RT_2} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323 \text{ K})} = 0.0836 \text{ kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070 \text{ kg}}$$

Tire
25°C



4-80 Two rigid tanks connected by a valve to each other contain air at specified conditions. The volume of the second tank and the final equilibrium pressure when the valve is opened are to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis Let's call the first and the second tanks A and B. Treating air as an ideal gas, the volume of the second tank and the mass of air in the first tank are determined to be

$$V_B = \left(\frac{m_1 R T_1}{P_1} \right)_B = \frac{(5 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(308 \text{ K})}{200 \text{ kPa}} = 2.21 \text{ m}^3$$

$$m_A = \left(\frac{P_1 V}{R T_1} \right)_A = \frac{(500 \text{ kPa})(1.0 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 5.846 \text{ kg}$$

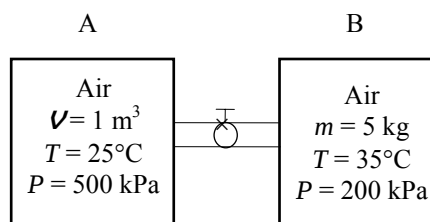
Thus,

$$V = V_A + V_B = 1.0 + 2.21 = 3.21 \text{ m}^3$$

$$m = m_A + m_B = 5.846 + 5.0 = 10.846 \text{ kg}$$

Then the final equilibrium pressure becomes

$$P_2 = \frac{m R T_2}{V} = \frac{(10.846 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})}{3.21 \text{ m}^3} = 284.1 \text{ kPa}$$



4-81E The validity of a statement that tires lose roughly 1 psi of pressure for every 10°F drop in outside temperature is to be investigated.

Assumptions 1 The air in the tire is an ideal gas. 2 The volume of air in the tire is constant. 3 The tire is in thermal equilibrium with the outside air. 4 The atmospheric conditions are 70°F and 1 atm = 14.7 psia.

Analysis The pressure in a tire should be checked at least once a month when a vehicle has sat for at least one hour to ensure that the tires are cool. The recommended gage pressure in cool tires is typically above 30 psi. Taking the initial gage pressure to be 32 psi, the gage pressure after the outside temperature drops by 10°F is determined from the ideal gas relation to be

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \rightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{(60 + 460) \text{ R}}{(70 + 460) \text{ R}} (32 + 14.7 \text{ psia}) = 45.8 \text{ psia} = 31.1 \text{ psig (gage)}$$

Then the drop in pressure corresponding to a drop of 10°F in temperature becomes

$$\Delta P = P_1 - P_2 = 32.0 - 31.1 = 0.9 \text{ psi}$$

which is sufficiently close to 1 psi. Therefore, the statement is **valid**.

Discussion Note that we used *absolute* temperatures and pressures in ideal gas calculations. Using gage pressures would result in pressure drop of 0.6 psi, which is considerably lower than 1 psi. Therefore, it is important to use absolute temperatures and pressures in the ideal gas relation.

4-82 A piston-cylinder device containing oxygen is cooled. The change of the volume is to be determined.

Assumptions At specified conditions, oxygen behaves as an ideal gas.

Properties The gas constant of oxygen is $R = 0.2598 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

Analysis According to the ideal gas equation of state, the initial volume of the oxygen is

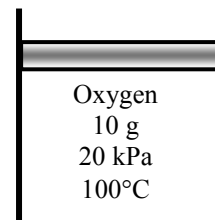
$$v_1 = \frac{mRT_1}{P_1} = \frac{(0.010 \text{ kg})(0.2598 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(100 + 273 \text{ K})}{20 \text{ kPa}} = 0.04845 \text{ m}^3$$

Similarly, the final volume is

$$v_2 = \frac{mRT_2}{P_2} = \frac{(0.010 \text{ kg})(0.2598 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(0 + 273 \text{ K})}{20 \text{ kPa}} = 0.03546 \text{ m}^3$$

The change of volume is then

$$\Delta V = v_2 - v_1 = 0.03546 - 0.04845 = -0.013 \text{ m}^3$$



4-83 A rigid vessel containing helium is heated. The temperature change is to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The gas constant of helium is $R = 2.0769 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

Analysis According to the ideal gas equation of state, the initial temperature is

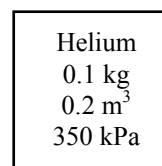
$$T_1 = \frac{P_1 v}{mR} = \frac{(350 \text{ kPa})(0.2 \text{ m}^3)}{(0.1 \text{ kg})(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 337 \text{ K}$$

Since the specific volume remains constant, the ideal gas equation gives

$$v_1 = \frac{RT_1}{P_1} = v_2 = \frac{RT_2}{P_2} \longrightarrow T_2 = T_1 \frac{P_2}{P_1} = (337 \text{ K}) \frac{700 \text{ kPa}}{350 \text{ kPa}} = 674 \text{ K}$$

The temperature change is then

$$\Delta T = T_2 - T_1 = 674 - 337 = 337 \text{ K}$$



4-84 A piston-cylinder device containing argon undergoes an isothermal process. The final pressure is to be determined.

Assumptions At specified conditions, argon behaves as an ideal gas.

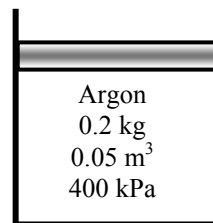
Properties The gas constant of argon is $R = 0.2081 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

Analysis Since the temperature remains constant, the ideal gas equation gives

$$m = \frac{P_1 V_1}{RT} = \frac{P_2 V_2}{RT} \longrightarrow P_1 V_1 = P_2 V_2$$

which when solved for final pressure becomes

$$P_2 = P_1 \frac{V_1}{V_2} = P_1 \frac{V_1}{2V_1} = 0.5P_1 = 0.5(400 \text{ kPa}) = \mathbf{200 \text{ kPa}}$$



Compressibility Factor

4-85C It represents the deviation from ideal gas behavior. The further away it is from 1, the more the gas deviates from ideal gas behavior.

4-86C All gases have the same compressibility factor Z at the same reduced temperature and pressure.

4-87C Reduced pressure is the pressure normalized with respect to the critical pressure; and reduced temperature is the temperature normalized with respect to the critical temperature.

4-88 The specific volume of steam is to be determined using the ideal gas relation, the compressibility chart, and the steam tables. The errors involved in the first two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 647.1 \text{ K}, \quad P_{\text{cr}} = 22.06 \text{ MPa}$$

Analysis (a) From the ideal gas equation of state,

$$\nu = \frac{RT}{P} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(673 \text{ K})}{(10,000 \text{ kPa})} = \mathbf{0.03106 \text{ m}^3/\text{kg} \text{ (17.6\% error)}}$$

(b) From the compressibility chart (Fig. A-15),

$$\left. \begin{aligned} P_R &= \frac{P}{P_{\text{cr}}} = \frac{10 \text{ MPa}}{22.06 \text{ MPa}} = 0.453 \\ T_R &= \frac{T}{T_{\text{cr}}} = \frac{673 \text{ K}}{647.1 \text{ K}} = 1.04 \end{aligned} \right\} Z = 0.84$$

H ₂ O
10 MPa
400°C

Thus,

$$\nu = Z\nu_{\text{ideal}} = (0.84)(0.03106 \text{ m}^3/\text{kg}) = \mathbf{0.02609 \text{ m}^3/\text{kg} \text{ (1.2\% error)}}$$

(c) From the superheated steam table (Table A-6),

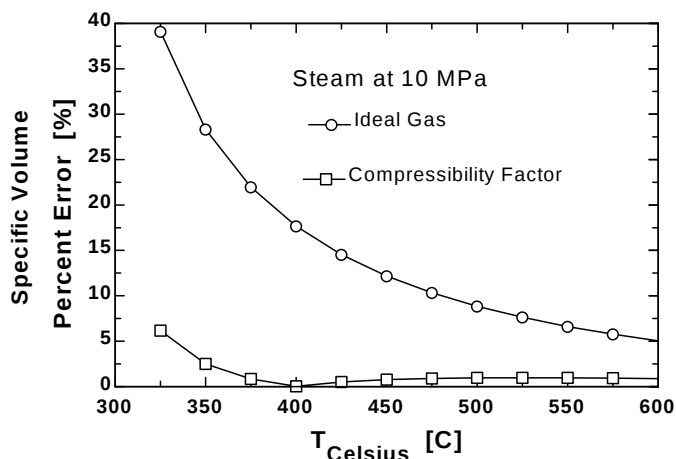
$$\left. \begin{aligned} P &= 10 \text{ MPa} \\ T &= 400^\circ\text{C} \end{aligned} \right\} \nu = \mathbf{0.02644 \text{ m}^3/\text{kg}}$$

4-89 EES Problem 4-88 is reconsidered. The problem is to be solved using the general compressibility factor feature of EES (or other) software. The specific volume of water for the three cases at 10 MPa over the temperature range of 325°C to 600°C in 25°C intervals is to be compared, and the %error involved in the ideal gas approximation is to be plotted against temperature.

Analysis The problem is solved using EES, and the solution is given below.

```
P=10 [MPa]*Convert(MPa,kPa)
{T_Celsius= 400 [C]}
T=T_Celsius+273 "[K]"
T_critical=T_CRIT(Steam_iapws)
P_critical=P_CRIT(Steam_iapws)
{v=Vol/m}
P_table=P; P_comp=P;P_idealgas=P
T_table=T; T_comp=T;T_idealgas=T
v_table=volume(Steam_iapws,P=P_table,T=T_table) "EES data for steam as a real gas"
{P_table=pressure(Steam_iapws, T=T_table,v=v)}
{T_sat=temperature(Steam_iapws,P=P_table,v=v)}
MM=MOLARMASS(water)
R_u=8.314 [kJ/kmol-K] "Universal gas constant"
R=R_u/MM "[kJ/kg-K], Particular gas constant"
P_idealgas*v_idealgas=R*T_idealgas "Ideal gas equation"
z = COMPRESS(T_comp/T_critical,P_comp/P_critical)
P_comp*v_comp=z*R*T_comp "generalized Compressibility factor"
Error_idealgas=Abs(v_table-v_idealgas)/v_table*Convert(, %)
Error_comp=Abs(v_table-v_comp)/v_table*Convert(, %)
```

Error _{comp} [%]	Error _{ideal gas} [%]	T _{Celsius} [C]
6.088	38.96	325
2.422	28.2	350
0.7425	21.83	375
0.129	17.53	400
0.6015	14.42	425
0.8559	12.07	450
0.9832	10.23	475
1.034	8.755	500
1.037	7.55	525
1.01	6.55	550
0.9652	5.712	575
0.9093	5	600



4-90 The specific volume of R-134a is to be determined using the ideal gas relation, the compressibility chart, and the R-134a tables. The errors involved in the first two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of refrigerant-134a are, from Table A-1,

$$R = 0.08149 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 374.2 \text{ K}, \quad P_{\text{cr}} = 4.059 \text{ MPa}$$

Analysis (a) From the ideal gas equation of state,

$$\nu = \frac{RT}{P} = \frac{(0.08149 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(343 \text{ K})}{900 \text{ kPa}} = \mathbf{0.03105 \text{ m}^3/\text{kg}} \quad (13.3\% \text{ error})$$

(b) From the compressibility chart (Fig. A-15),

$$\left. \begin{aligned} P_R &= \frac{P}{P_{\text{cr}}} = \frac{0.9 \text{ MPa}}{4.059 \text{ MPa}} = 0.222 \\ T_R &= \frac{T}{T_{\text{cr}}} = \frac{343 \text{ K}}{374.2 \text{ K}} = 0.917 \end{aligned} \right\} Z = 0.894$$

R-134a 0.9 MPa 70°C

Thus,

$$\nu = Z\nu_{\text{ideal}} = (0.894)(0.03105 \text{ m}^3/\text{kg}) = \mathbf{0.02776 \text{ m}^3/\text{kg}} \quad (1.3\% \text{ error})$$

(c) From the superheated refrigerant table (Table A-13),

$$\left. \begin{aligned} P &= 0.9 \text{ MPa} \\ T &= 70^\circ\text{C} \end{aligned} \right\} \nu = \mathbf{0.027413 \text{ m}^3/\text{kg}}$$

4-91 The specific volume of nitrogen gas is to be determined using the ideal gas relation and the compressibility chart. The errors involved in these two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of nitrogen are, from Table A-1,

$$R = 0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 126.2 \text{ K}, \quad P_{\text{cr}} = 3.39 \text{ MPa}$$

Analysis (a) From the ideal gas equation of state,

$$\nu = \frac{RT}{P} = \frac{(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(150 \text{ K})}{10,000 \text{ kPa}} = \mathbf{0.004452 \text{ m}^3/\text{kg}} \quad (86.4\% \text{ error})$$

(b) From the compressibility chart (Fig. A-15),

$$\left. \begin{aligned} P_R &= \frac{P}{P_{\text{cr}}} = \frac{10 \text{ MPa}}{3.39 \text{ MPa}} = 2.95 \\ T_R &= \frac{T}{T_{\text{cr}}} = \frac{150 \text{ K}}{126.2 \text{ K}} = 1.19 \end{aligned} \right\} Z = 0.54$$

N ₂ 10 MPa 150 K

Thus,

$$\nu = Z\nu_{\text{ideal}} = (0.54)(0.004452 \text{ m}^3/\text{kg}) = \mathbf{0.002404 \text{ m}^3/\text{kg}} \quad (0.7\% \text{ error})$$

4-92 The specific volume of steam is to be determined using the ideal gas relation, the compressibility chart, and the steam tables. The errors involved in the first two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 647.1 \text{ K}, \quad P_{\text{cr}} = 22.06 \text{ MPa}$$

Analysis (a) From the ideal gas equation of state,

$$\nu = \frac{RT}{P} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(723 \text{ K})}{3500 \text{ kPa}} = \mathbf{0.09533 \text{ m}^3/\text{kg}} \quad (3.7\% \text{ error})$$

(b) From the compressibility chart (Fig. A-15),

$$\left. \begin{aligned} P_R &= \frac{P}{P_{\text{cr}}} = \frac{3.5 \text{ MPa}}{22.06 \text{ MPa}} = 0.159 \\ T_R &= \frac{T}{T_{\text{cr}}} = \frac{723 \text{ K}}{647.1 \text{ K}} = 1.12 \end{aligned} \right\} Z = 0.961$$

H ₂ O
3.5 MPa
450°C

Thus,

$$\nu = Z\nu_{\text{ideal}} = (0.961)(0.09533 \text{ m}^3/\text{kg}) = \mathbf{0.09161 \text{ m}^3/\text{kg}} \quad (0.4\% \text{ error})$$

(c) From the superheated steam table (Table A-6),

$$\left. \begin{aligned} P &= 3.5 \text{ MPa} \\ T &= 450^\circ\text{C} \end{aligned} \right\} \nu = \mathbf{0.09196 \text{ m}^3/\text{kg}}$$

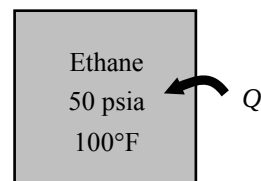
4-93E Ethane in a rigid vessel is heated. The final pressure is to be determined using the compressibility chart.

Properties The gas constant, the critical pressure, and the critical temperature of ethane are, from Table A-1E,

$$R = 0.3574 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}, \quad T_{\text{cr}} = 549.8 \text{ R}, \quad P_{\text{cr}} = 708 \text{ psia}$$

Analysis From the compressibility chart at the initial state (Fig. A-15),

$$\left. \begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{560 \text{ R}}{549.8 \text{ R}} = 1.019 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{50 \text{ psia}}{708 \text{ psia}} = 0.0706 \end{aligned} \right\} Z_1 = 0.977$$



The specific volume does not change during the process. Then,

$$\nu_1 = \nu_2 = \frac{Z_1 R T_1}{P_1} = \frac{(0.977)(0.3574 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(560 \text{ R})}{50 \text{ psia}} = 3.911 \text{ ft}^3/\text{lbm}$$

At the final state,

$$\left. \begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{1060 \text{ R}}{549.8 \text{ R}} = 1.928 \\ \nu_{R2} &= \frac{\nu_{2,\text{actual}}}{R T_{\text{cr}}/P_{\text{cr}}} = \frac{3.911 \text{ ft}^3/\text{lbm}}{(0.3574 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(549.8 \text{ R})/(708 \text{ psia})} = 14.09 \end{aligned} \right\} Z_2 = 1.0$$

Thus,

$$P_2 = \frac{Z_2 R T_2}{\nu_2} = \frac{(1.0)(0.3574 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(1060 \text{ R})}{3.911 \text{ ft}^3/\text{lbm}} = \mathbf{96.9 \text{ psia}}$$

4-94 Ethylene is heated at constant pressure. The specific volume change of ethylene is to be determined using the compressibility chart.

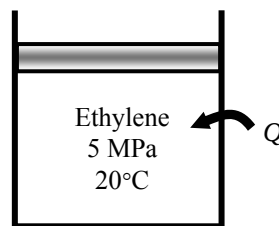
Properties The gas constant, the critical pressure, and the critical temperature of ethane are, from Table A-1,

$$R = 0.2964 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 282.4 \text{ K}, \quad P_{\text{cr}} = 5.12 \text{ MPa}$$

Analysis From the compressibility chart at the initial and final states (Fig. A-15),

$$\left. \begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{293 \text{ K}}{282.4 \text{ K}} = 1.038 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{5 \text{ MPa}}{5.12 \text{ MPa}} = 0.977 \end{aligned} \right\} Z_1 = 0.56$$

$$\left. \begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{473 \text{ K}}{282.4 \text{ K}} = 1.675 \\ P_{R2} &= P_{R1} = 0.977 \end{aligned} \right\} Z_2 = 0.961$$



The specific volume change is

$$\begin{aligned} \Delta v &= \frac{R}{P} (Z_2 T_2 - Z_1 T_1) \\ &= \frac{0.2964 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}}{5000 \text{ kPa}} [(0.961)(473 \text{ K}) - (0.56)(293 \text{ K})] \\ &= \mathbf{0.0172 \text{ m}^3/\text{kg}} \end{aligned}$$

4-95 Water vapor is heated at constant pressure. The final temperature is to be determined using ideal gas equation, the compressibility charts, and the steam tables.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}, \quad T_{\text{cr}} = 647.1 \text{ K}, \quad P_{\text{cr}} = 22.06 \text{ MPa}$$

Analysis (a) From the ideal gas equation,

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (350 + 273 \text{ K})(2) = \mathbf{1246 \text{ K}}$$

(b) The pressure of the steam is

$$P_1 = P_2 = P_{\text{sat}@350^\circ\text{C}} = 16,529 \text{ kPa}$$

From the compressibility chart at the initial state (Fig. A-15),

$$\left. \begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{623 \text{ K}}{647.1 \text{ K}} = 0.963 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{16.529 \text{ MPa}}{22.06 \text{ MPa}} = 0.749 \end{aligned} \right\} Z_1 = 0.593, \nu_{R1} = 0.75$$

At the final state,

$$\left. \begin{aligned} P_{R2} &= P_{R1} = 0.749 \\ \nu_{R2} &= 2\nu_{R1} = 2(0.75) = 1.50 \end{aligned} \right\} Z_2 = 0.88$$

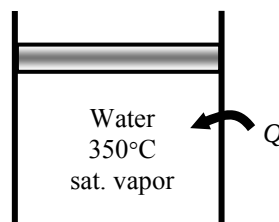
Thus,

$$T_2 = \frac{P_2 \nu_2}{Z_2 R} = \frac{P_2}{Z_2} \frac{\nu_{R2} T_{\text{cr}}}{P_{\text{cr}}} = \frac{16,529 \text{ kPa}}{0.88} \frac{(1.50)(647.1 \text{ K})}{22,060 \text{ kPa}} = \mathbf{826 \text{ K}}$$

(c) From the superheated steam table,

$$\left. \begin{aligned} T_1 &= 350^\circ\text{C} \\ x_1 &= 1 \end{aligned} \right\} \nu_1 = 0.008806 \text{ m}^3/\text{kg} \quad (\text{Table A-4})$$

$$\left. \begin{aligned} P_2 &= 16,529 \text{ kPa} \\ \nu_2 &= 2\nu_1 = 0.01761 \text{ m}^3/\text{kg} \end{aligned} \right\} T_2 = 477^\circ\text{C} = \mathbf{750 \text{ K}} \quad (\text{from Table A-6 or EES})$$



4-96E Water vapor is heated at constant pressure. The final temperature is to be determined using ideal gas equation, the compressibility charts, and the steam tables.

Properties The critical pressure and the critical temperature of water are, from Table A-1E,

$$R = 0.5956 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R}, \quad T_{\text{cr}} = 1164.8 \text{ R}, \quad P_{\text{cr}} = 3200 \text{ psia}$$

Analysis (a) From the ideal gas equation,

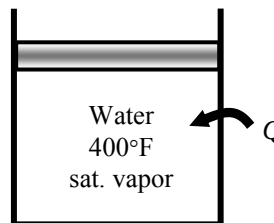
$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (400 + 460 \text{ R})(2) = \mathbf{1720 \text{ R}}$$

(b) The properties of steam are (Table A-4E)

$$P_1 = P_2 = P_{\text{sat}@400^\circ\text{F}} = 247.26 \text{ psia}$$

$$\nu_1 = \nu_{\text{g}@400^\circ\text{F}} = 1.8639 \text{ ft}^3 / \text{lbm}$$

$$\nu_2 = 2\nu_1 = 3.7278 \text{ ft}^3 / \text{lbm}$$



At the final state, from the compressibility chart (Fig. A-15),

$$\left. \begin{aligned} P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{247.26 \text{ psia}}{3200 \text{ psia}} = 0.0773 \\ \nu_{R2} &= \frac{\nu_{2,\text{actual}}}{RT_{\text{cr}}/P_{\text{cr}}} = \frac{3.7278 \text{ ft}^3 / \text{lbm}}{(0.5956 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R})(1164.8 \text{ R}) / (3200 \text{ psia})} = 17.19 \end{aligned} \right\} Z_2 = 0.985$$

Thus,

$$T_2 = \frac{P_2 \nu_2}{Z_2 R} = \frac{(247.26 \text{ psia})(3.7278 \text{ ft}^3 / \text{lbm})}{(0.985)(0.5956 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R})} = \mathbf{1571 \text{ R}}$$

(c) From the superheated steam table,

$$\left. \begin{aligned} P_2 &= 247.26 \text{ psia} \\ \nu_2 &= 3.7278 \text{ ft}^3 / \text{lbm} \end{aligned} \right\} T_2 = 1100^\circ\text{F} = \mathbf{1560 \text{ R}} \quad (\text{from Table A-6E or EES})$$

4-97 Methane is heated at constant pressure. The final temperature is to be determined using ideal gas equation and the compressibility charts.

Properties The gas constant, the critical pressure, and the critical temperature of methane are, from Table A-1,

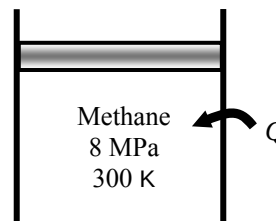
$$R = 0.5182 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}, \quad T_{\text{cr}} = 191.1 \text{ K}, \quad P_{\text{cr}} = 4.64 \text{ MPa}$$

Analysis From the ideal gas equation,

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (300 \text{ K})(1.5) = \mathbf{450 \text{ K}}$$

From the compressibility chart at the initial state (Fig. A-15),

$$\left. \begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{300 \text{ K}}{191.1 \text{ K}} = 1.570 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{8 \text{ MPa}}{4.64 \text{ MPa}} = 1.724 \end{aligned} \right\} Z_1 = 0.88, \nu_{R1} = 0.80$$



At the final state,

$$\left. \begin{aligned} P_{R2} &= P_{R1} = 1.724 \\ \nu_{R2} &= 1.5\nu_{R1} = 1.5(0.80) = 1.2 \end{aligned} \right\} Z_2 = 0.975$$

Thus,

$$T_2 = \frac{P_2 \nu_2}{Z_2 R} = \frac{P_2}{Z_2} \frac{\nu_{R2} T_{\text{cr}}}{P_{\text{cr}}} = \frac{8000 \text{ kPa}}{0.975} \frac{(1.2)(191.1 \text{ K})}{4640 \text{ kPa}} = \mathbf{406 \text{ K}}$$

Of these two results, the accuracy of the second result is limited by the accuracy with which the charts may be read. Accepting the error associated with reading charts, the second temperature is the more accurate.

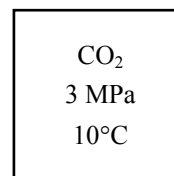
4-98 The percent error involved in treating CO_2 at a specified state as an ideal gas is to be determined.

Properties The critical pressure, and the critical temperature of CO_2 are, from Table A-1,

$$T_{\text{cr}} = 304.2 \text{ K} \text{ and } P_{\text{cr}} = 7.39 \text{ MPa}$$

Analysis From the compressibility chart (Fig. A-15),

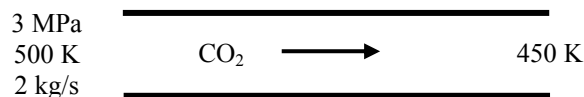
$$\left. \begin{aligned} P_R &= \frac{P}{P_{\text{cr}}} = \frac{3 \text{ MPa}}{7.39 \text{ MPa}} = 0.406 \\ T_R &= \frac{T}{T_{\text{cr}}} = \frac{283 \text{ K}}{304.2 \text{ K}} = 0.93 \end{aligned} \right\} Z = 0.80$$



Then the error involved in treating CO_2 as an ideal gas is

$$\text{Error} = \frac{\nu - \nu_{\text{ideal}}}{\nu} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.80} = -0.25 \text{ or } \mathbf{25.0\%}$$

4-99 CO₂ gas flows through a pipe. The volume flow rate and the density at the inlet and the volume flow rate at the exit of the pipe are to be determined.



Properties The gas constant, the critical pressure, and the critical temperature of CO₂ are (Table A-1)

$$R = 0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 304.2 \text{ K}, \quad P_{\text{cr}} = 7.39 \text{ MPa}$$

Analysis (a) From the ideal gas equation of state,

$$\dot{V}_1 = \frac{\dot{m}RT_1}{P_1} = \frac{(2 \text{ kg/s})(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(500 \text{ K})}{(3000 \text{ kPa})} = \mathbf{0.06297 \text{ m}^3/\text{kg} \text{ (2.1\% error)}}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{(3000 \text{ kPa})}{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(500 \text{ K})} = \mathbf{31.76 \text{ kg/m}^3 \text{ (2.1\% error)}}$$

$$\dot{V}_2 = \frac{\dot{m}RT_2}{P_2} = \frac{(2 \text{ kg/s})(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(450 \text{ K})}{(3000 \text{ kPa})} = \mathbf{0.05667 \text{ m}^3/\text{kg} \text{ (3.6\% error)}}$$

(b) From the compressibility chart (EES function for compressibility factor is used)

$$\left. \begin{aligned} P_R &= \frac{P_1}{P_{\text{cr}}} = \frac{3 \text{ MPa}}{7.39 \text{ MPa}} = 0.407 \\ T_{R,1} &= \frac{T_1}{T_{\text{cr}}} = \frac{500 \text{ K}}{304.2 \text{ K}} = 1.64 \end{aligned} \right\} Z_1 = 0.9791$$

$$\left. \begin{aligned} P_R &= \frac{P_2}{P_{\text{cr}}} = \frac{3 \text{ MPa}}{7.39 \text{ MPa}} = 0.407 \\ T_{R,2} &= \frac{T_2}{T_{\text{cr}}} = \frac{450 \text{ K}}{304.2 \text{ K}} = 1.48 \end{aligned} \right\} Z_2 = 0.9656$$

Thus, $\dot{V}_1 = \frac{Z_1 \dot{m}RT_1}{P_1} = \frac{(0.9791)(2 \text{ kg/s})(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(500 \text{ K})}{(3000 \text{ kPa})} = \mathbf{0.06165 \text{ m}^3/\text{kg}}$

$$\rho_1 = \frac{P_1}{Z_1 RT_1} = \frac{(3000 \text{ kPa})}{(0.9791)(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(500 \text{ K})} = \mathbf{32.44 \text{ kg/m}^3}$$

$$\dot{V}_2 = \frac{Z_2 \dot{m}RT_2}{P_2} = \frac{(0.9656)(2 \text{ kg/s})(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(450 \text{ K})}{(3000 \text{ kPa})} = \mathbf{0.05472 \text{ m}^3/\text{kg}}$$

Other Equations of State

4-100C The constant a represents the increase in pressure as a result of intermolecular forces; the constant b represents the volume occupied by the molecules. They are determined from the requirement that the critical isotherm has an inflection point at the critical point.

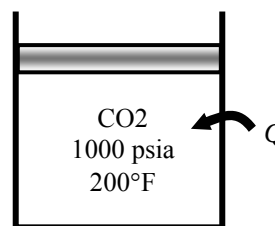
4-101E Carbon dioxide is heated in a constant pressure apparatus. The final volume of the carbon dioxide is to be determined using the ideal gas equation and the Benedict-Webb-Rubin equation of state.

Properties The gas constant and molar mass of CO₂ are (Table A-1E)

$$R = 0.2438 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R}, \quad M = 44.01 \text{ lbm/lbmol}$$

Analysis (a) From the ideal gas equation of state,

$$\nu_2 = \frac{mRT_2}{P} = \frac{(1 \text{ lbm})(0.2438 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R})(1260 \text{ R})}{1000 \text{ psia}} = \mathbf{0.3072 \text{ ft}^3}$$



(b) Using the coefficients of Table 4-4 for carbon dioxide and the given data in SI units, the Benedict-Webb-Rubin equation of state for state 2 is

$$P_2 = \frac{R_u T_2}{\bar{\nu}_2} + \left(B_0 R_u T_2 - A_0 - \frac{C_0}{T_2^2} \right) \frac{1}{\bar{\nu}_2^2} + \frac{b R_u T_2 - a}{\bar{\nu}_2^3} + \frac{a \alpha}{\bar{\nu}_2^6} + \frac{c}{\bar{\nu}_2^3 T_2^2} \left(1 + \frac{\gamma}{\bar{\nu}_2^2} \right) \exp(-\gamma / \bar{\nu}_2^2)$$

$$6895 = \frac{(8.314)(700)}{\bar{\nu}_2} + \left(0.04991 \times 8.314 \times 700 - 277.30 - \frac{1.404 \times 10^7}{700^2} \right) \frac{1}{\bar{\nu}_2^2} + \frac{0.007210 \times 8.314 \times 700 - 13.86}{\bar{\nu}_2^3}$$

$$+ \frac{13.86 \times 8.470 \times 10^{-5}}{\bar{\nu}_2^6} + \frac{1.511 \times 10^6}{\bar{\nu}_2^3 (700)^2} \left(1 + \frac{0.00539}{\bar{\nu}_2^2} \right) \exp(-0.00539 / \bar{\nu}_2^2)$$

The solution of this equation by an equation solver such as EES gives

$$\bar{\nu}_2 = 0.8477 \text{ m}^3 / \text{kmol}$$

Then,

$$\nu_2 = \frac{\bar{\nu}_2}{M} = \frac{0.8477 \text{ m}^3 / \text{kmol}}{44.01 \text{ kg/kmol}} = 0.01926 \text{ m}^3 / \text{kg}$$

$$\nu_2 = m \nu_2 = (1 / 2.2046 \text{ kg})(0.01926 \text{ m}^3 / \text{kg}) = 0.008741 \text{ m}^3$$

$$= (0.008741 \text{ m}^3) \left(\frac{35.315 \text{ ft}^3}{1 \text{ m}^3} \right) = \mathbf{0.3087 \text{ ft}^3}$$

4-102 Methane is heated in a rigid container. The final pressure of the methane is to be determined using the ideal gas equation and the Benedict-Webb-Rubin equation of state.

Analysis (a) From the ideal gas equation of state,

$$P_2 = P_1 \frac{T_2}{T_1} = (100 \text{ kPa}) \frac{673 \text{ K}}{293 \text{ K}} = \mathbf{229.7 \text{ kPa}}$$

The specific molar volume of the methane is

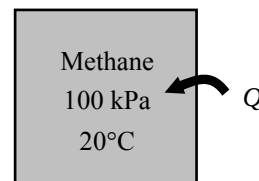
$$\bar{v}_1 = \bar{v}_2 = \frac{R_u T_1}{P_1} = \frac{(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(293 \text{ K})}{100 \text{ kPa}} = 24.36 \text{ m}^3/\text{kmol}$$

(b) The specific molar volume of the methane is

$$\bar{v}_1 = \bar{v}_2 = \frac{R_u T_1}{P_1} = \frac{(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(293 \text{ K})}{100 \text{ kPa}} = 24.36 \text{ m}^3/\text{kmol}$$

Using the coefficients of Table 4-4 for methane and the given data, the Benedict-Webb-Rubin equation of state for state 2 gives

$$\begin{aligned} P_2 &= \frac{R_u T_2}{\bar{v}_2} + \left(B_0 R_u T_2 - A_0 - \frac{C_0}{T_2^2} \right) \frac{1}{\bar{v}_2^2} + \frac{b R_u T_2 - a}{\bar{v}_2^3} + \frac{a \alpha}{\bar{v}_2^6} + \frac{c}{\bar{v}_2^3 T_2^2} \left(1 + \frac{\gamma}{\bar{v}_2^2} \right) \exp(-\gamma / \bar{v}_2^2) \\ &= \frac{(8.314)(673)}{24.36} + \left(0.04260 \times 8.314 \times 673 - 187.91 - \frac{2.286 \times 10^6}{673^2} \right) \frac{1}{24.36^2} + \frac{0.003380 \times 8.314 \times 673 - 5.00}{24.36^3} \\ &\quad + \frac{5.00 \times 1.244 \times 10^{-4}}{24.36^6} + \frac{2.578 \times 10^5}{24.36^3 (673)^2} \left(1 + \frac{0.0060}{24.36^2} \right) \exp(-0.0060 / 24.36^2) \\ &= \mathbf{229.8 \text{ kPa}} \end{aligned}$$



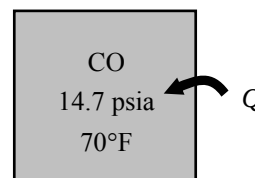
4-103E Carbon monoxide is heated in a rigid container. The final pressure of the CO is to be determined using the ideal gas equation and the Benedict-Webb-Rubin equation of state.

Properties The gas constant and molar mass of CO are (Table A-1)

$$R = 0.2968 \text{ kPa} \cdot \text{m}^3 / \text{kg} \cdot \text{K}, \quad M = 28.011 \text{ kg/kmol}$$

Analysis (a) From the ideal gas equation of state,

$$P_2 = P_1 \frac{T_2}{T_1} = (14.7 \text{ psia}) \frac{1260 \text{ R}}{530 \text{ R}} = \mathbf{34.95 \text{ psia}}$$



The specific molar volume of the CO in SI units is

$$\bar{v}_1 = \bar{v}_2 = \frac{R_u T_1}{P_1} = \frac{(8.314 \text{ kPa} \cdot \text{m}^3 / \text{kmol} \cdot \text{K})(294 \text{ K})}{101 \text{ kPa}} = 24.20 \text{ m}^3 / \text{kmol}$$

(b) The specific molar volume of the CO in SI units is

$$\bar{v}_1 = \bar{v}_2 = \frac{R_u T_1}{P_1} = \frac{(8.314 \text{ kPa} \cdot \text{m}^3 / \text{kmol} \cdot \text{K})(294 \text{ K})}{101 \text{ kPa}} = 24.20 \text{ m}^3 / \text{kmol}$$

Using the coefficients of Table 4-4 for CO and the given data, the Benedict-Webb-Rubin equation of state for state 2 gives

$$\begin{aligned} P_2 &= \frac{R_u T_2}{\bar{v}_2} + \left(B_0 R_u T_2 - A_0 - \frac{C_0}{T_2^2} \right) \frac{1}{\bar{v}_2^2} + \frac{b R_u T_2 - a}{\bar{v}_2^3} + \frac{a \alpha}{\bar{v}_2^6} + \frac{c}{\bar{v}_2^3 T_2^2} \left(1 + \frac{\gamma}{\bar{v}_2^2} \right) \exp(-\gamma / \bar{v}_2^2) \\ &= \frac{(8.314)(700)}{24.20} + \left(0.05454 \times 8.314 \times 700 - 135.87 - \frac{8.673 \times 10^5}{700^2} \right) \frac{1}{24.20^2} + \frac{0.002632 \times 8.314 \times 700 - 3.71}{24.20^3} \\ &\quad + \frac{3.71 \times 1.350 \times 10^{-4}}{24.20^6} + \frac{1.054 \times 10^5}{24.20^3 (700)^2} \left(1 + \frac{0.0060}{24.20^2} \right) \exp(-0.0060 / 24.20^2) \\ &= 240.8 \text{ kPa} \end{aligned}$$

The pressure in English unit is

$$P_2 = (240.8 \text{ kPa}) \left(\frac{1 \text{ psia}}{6.8948 \text{ kPa}} \right) = \mathbf{34.92 \text{ psia}}$$

4-104 Carbon dioxide is compressed in a piston-cylinder device in a polytropic process. The final temperature is to be determined using the ideal gas and van der Waals equations.

Properties The gas constant, molar mass, critical pressure, and critical temperature of carbon dioxide are (Table A-1)

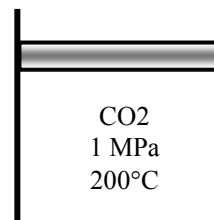
$$R = 0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad M = 44.01 \text{ kg/kmol}, \quad T_{\text{cr}} = 304.2 \text{ K}, \quad P_{\text{cr}} = 7.39 \text{ MPa}$$

Analysis (a) The specific volume at the initial state is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(473 \text{ K})}{1000 \text{ kPa}} = 0.08935 \text{ m}^3/\text{kg}$$

According to process specification,

$$\nu_2 = \nu_1 \left(\frac{P_1}{P_2} \right)^{1/n} = (0.08935 \text{ m}^3/\text{kg}) \left(\frac{1000 \text{ kPa}}{3000 \text{ kPa}} \right)^{1/1.2} = 0.03577 \text{ m}^3/\text{kg}$$



The final temperature is then

$$T_2 = \frac{P_2 \nu_2}{R} = \frac{(3000 \text{ kPa})(0.03577 \text{ m}^3/\text{kg})}{0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}} = \mathbf{568 \text{ K}}$$

(b) The van der Waals constants for carbon dioxide are determined from

$$a = \frac{27R^2 T_{\text{cr}}^2}{64P_{\text{cr}}} = \frac{(27)(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})^2 (304.2 \text{ K})^2}{(64)(7390 \text{ kPa})} = 0.1885 \text{ m}^6 \cdot \text{kPa}/\text{kg}^2$$

$$b = \frac{RT_{\text{cr}}}{8P_{\text{cr}}} = \frac{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(304.2 \text{ K})}{8 \times 7390 \text{ kPa}} = 0.0009720 \text{ m}^3/\text{kg}$$

Applying the van der Waals equation to the initial state,

$$\left(P + \frac{a}{\nu^2} \right) (\nu - b) = RT$$

$$\left(1000 + \frac{0.1885}{\nu^2} \right) (\nu - 0.0009720) = (0.1889)(473)$$

Solving this equation by trial-error or by EES gives

$$\nu_1 = 0.08821 \text{ m}^3/\text{kg}$$

According to process specification,

$$\nu_2 = \nu_1 \left(\frac{P_1}{P_2} \right)^{1/n} = (0.08821 \text{ m}^3/\text{kg}) \left(\frac{1000 \text{ kPa}}{3000 \text{ kPa}} \right)^{1/1.2} = 0.03531 \text{ m}^3/\text{kg}$$

Applying the van der Waals equation to the final state,

$$\left(P + \frac{a}{\nu^2} \right) (\nu - b) = RT$$

$$\left(3000 + \frac{0.1885}{0.03531^2} \right) (0.03531 - 0.0009720) = (0.1889)T$$

Solving for the final temperature gives

$$T_2 = \mathbf{573 \text{ K}}$$

4-105E The temperature of R-134a in a tank at a specified state is to be determined using the ideal gas relation, the van der Waals equation, and the refrigerant tables.

Properties The gas constant, critical pressure, and critical temperature of R-134a are (Table A-1E)

$$R = 0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}, \quad T_{\text{cr}} = 673.6 \text{ R}, \quad P_{\text{cr}} = 588.7 \text{ psia}$$

Analysis (a) From the ideal gas equation of state,

$$T = \frac{P\nu}{R} = \frac{(100 \text{ psia})(0.54022 \text{ ft}^3/\text{lbm})}{0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}} = \mathbf{513.5 \text{ R}}$$

(b) The van der Waals constants for the refrigerant are determined from

$$a = \frac{27R^2T_{\text{cr}}^2}{64P_{\text{cr}}} = \frac{(27)(0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})^2 (673.6 \text{ R})^2}{(64)(588.7 \text{ psia})} = 3.591 \text{ ft}^6 \cdot \text{psia}/\text{lbm}^2$$

$$b = \frac{RT_{\text{cr}}}{8P_{\text{cr}}} = \frac{(0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(673.6 \text{ R})}{8 \times 588.7 \text{ psia}} = 0.0150 \text{ ft}^3/\text{lbm}$$

$$\text{Then, } T = \frac{1}{R} \left(P + \frac{a}{\nu^2} \right) (\nu - b) = \frac{1}{0.1052} \left(100 + \frac{3.591}{(0.54022)^2} \right) (0.54022 - 0.0150) = \mathbf{560.7 \text{ R}}$$

(c) From the superheated refrigerant table (Table A-13E),

$$\left. \begin{array}{l} P = 100 \text{ psia} \\ \nu = 0.54022 \text{ ft}^3/\text{lbm} \end{array} \right\} T = \mathbf{120^\circ\text{F}} \quad (580\text{R})$$

4-106 CD EES The pressure of nitrogen in a tank at a specified state is to be determined using the ideal gas relation and the Beattie-Bridgeman equation. The error involved in each case is to be determined.

Properties The gas constant and molar mass of nitrogen are (Table A-1)

$$R = 0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K} \quad \text{and} \quad M = 28.013 \text{ kg/kmol}$$

Analysis (a) From the ideal gas equation of state,

$$P = \frac{RT}{\nu} = \frac{(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(150 \text{ K})}{0.041884 \text{ m}^3/\text{kg}} = \mathbf{1063 \text{ kPa}} \quad (6.3\% \text{ error})$$

$\begin{aligned} &\text{N}_2 \\ &0.041884 \text{ m}^3/\text{kg} \\ &150 \text{ K} \end{aligned}$
--

(b) The constants in the Beattie-Bridgeman equation are

$$A = A_o \left(1 - \frac{a}{\bar{\nu}} \right) = 136.2315 \left(1 - \frac{0.02617}{1.1733} \right) = 133.193$$

$$B = B_o \left(1 - \frac{b}{\bar{\nu}} \right) = 0.05046 \left(1 - \frac{-0.00691}{1.1733} \right) = 0.05076$$

$$c = 4.2 \times 10^4 \text{ m}^3 \cdot \text{K}^3/\text{kmol}$$

since $\bar{\nu} = M\nu = (28.013 \text{ kg/kmol})(0.041884 \text{ m}^3/\text{kg}) = 1.1733 \text{ m}^3/\text{kmol}$.

Substituting,

$$\begin{aligned} P &= \frac{R_u T}{\bar{\nu}^2} \left(1 - \frac{c}{\bar{\nu} T^3} \right) (\bar{\nu} + B) - \frac{A}{\bar{\nu}^2} = \frac{8.314 \times 150}{(1.1733)^2} \left(1 - \frac{4.2 \times 10^4}{1.1733 \times 150^3} \right) (1.1733 + 0.05076) - \frac{133.193}{(1.1733)^2} \\ &= \mathbf{1000.4 \text{ kPa}} \quad (\text{negligible error}) \end{aligned}$$

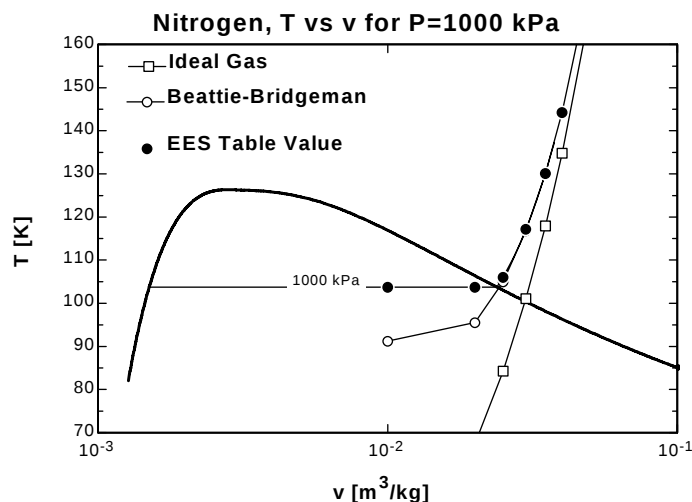
4-107 EES Problem 4-106 is reconsidered. Using EES (or other) software, the pressure results of the ideal gas and Beattie-Bridgeman equations with nitrogen data supplied by EES are to be compared. The temperature is to be plotted versus specific volume for a pressure of 1000 kPa with respect to the saturated liquid and saturated vapor lines of nitrogen over the range of $110\text{ K} < T < 150\text{ K}$.

Analysis The problem is solved using EES, and the solution is given below.

```
Function BeattBridg(T,v,M,R_u)
v_bar=v*M "Conversion from m^3/kg to m^3/kmol"
"The constants for the Beattie-Bridgeman equation of state are found in text"
Ao=136.2315; aa=0.02617; Bo=0.05046; bb=-0.00691; cc=4.20*1E4
B=Bo*(1-bb/v_bar)
A=Ao*(1-aa/v_bar)
"The Beattie-Bridgeman equation of state is"
BeattBridg:=R_u*T/(v_bar**2)*(1-cc/(v_bar*T**3))*(v_bar+B)-A/v_bar**2
End

T=150 [K]
v=0.041884 [m^3/kg]
P_exper=1000 [kPa]
T_table=T; T_BB=T; T_idealgas=T
P_table=PRESSURE(Nitrogen,T=T_table,v=v) "EES data for nitrogen as a real gas"
{T_table=temperature(Nitrogen, P=P_table,v=v)}
M=MOLARMASS(Nitrogen)
R_u=8.314 [kJ/kmol-K] "Universal gas constant"
R=R_u/M "Particular gas constant"
P_idealgas=R*T_idealgas/v "Ideal gas equation"
P_BB=BeattBridg(T_BB,v,M,R_u) "Beattie-Bridgeman equation of state Function"
```

P _{BB} [kPa]	P _{table} [kPa]	P _{idealgas} [kPa]	v [m ³ /kg]	T _{BB} [K]	T _{ideal gas} [K]	T _{table} [K]
1000	1000	1000	0.01	91.23	33.69	103.8
1000	1000	1000	0.02	95.52	67.39	103.8
1000	1000	1000	0.025	105	84.23	106.1
1000	1000	1000	0.03	116.8	101.1	117.2
1000	1000	1000	0.035	130.1	117.9	130.1
1000	1000	1000	0.04	144.4	134.8	144.3
1000	1000	1000	0.05	174.6	168.5	174.5



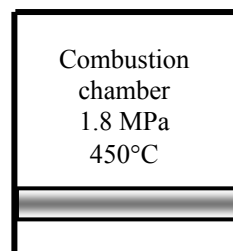
Review Problems

4-108 The cylinder conditions before the heat addition process is specified. The pressure after the heat addition process is to be determined.

Assumptions 1 The contents of cylinder are approximated by the air properties. 2 Air is an ideal gas.

Analysis The final pressure may be determined from the ideal gas relation

$$P_2 = \frac{T_2}{T_1} P_1 = \left(\frac{1300 + 273 \text{ K}}{450 + 273 \text{ K}} \right) (1800 \text{ kPa}) = \mathbf{3916 \text{ kPa}}$$



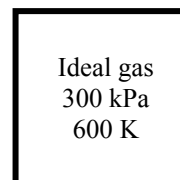
4-109 A rigid tank contains an ideal gas at a specified state. The final temperature is to be determined for two different processes.

Analysis (a) The first case is a constant volume process. When half of the gas is withdrawn from the tank, the final temperature may be determined from the ideal gas relation as

$$T_2 = \frac{m_1}{m_2} \frac{P_2}{P_1} T_1 = (2) \left(\frac{100 \text{ kPa}}{300 \text{ kPa}} \right) (600 \text{ K}) = \mathbf{400 \text{ K}}$$

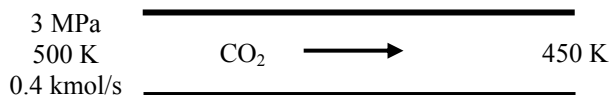
(b) The second case is a constant volume and constant mass process. The ideal gas relation for this case yields

$$P_2 = \frac{T_2}{T_1} P_1 = \left(\frac{400 \text{ K}}{600 \text{ K}} \right) (300 \text{ kPa}) = \mathbf{200 \text{ kPa}}$$



4-110 Carbon dioxide flows through a pipe at a given state. The volume and mass flow rates and the density of CO₂ at the given state and the volume flow rate at the exit of the pipe are to be determined.

Analysis (a) The volume and mass flow rates may be determined from ideal gas relation as



$$\dot{V}_1 = \frac{\dot{N} R_u T_1}{P} = \frac{(0.4 \text{ kmol/s})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(500 \text{ K})}{3000 \text{ kPa}} = \mathbf{0.5543 \text{ m}^3/\text{s}}$$

$$\dot{m}_1 = \frac{P_1 \dot{V}_1}{R T_1} = \frac{(3000 \text{ kPa})(0.5543 \text{ m}^3/\text{s})}{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500 \text{ K})} = \mathbf{17.60 \text{ kg/s}}$$

The density is

$$\rho_1 = \frac{\dot{m}_1}{\dot{V}_1} = \frac{(17.60 \text{ kg/s})}{(0.5543 \text{ m}^3/\text{s})} = \mathbf{31.76 \text{ kg/m}^3}$$

(b) The volume flow rate at the exit is

$$\dot{V}_2 = \frac{\dot{N} R_u T_2}{P} = \frac{(0.4 \text{ kmol/s})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(450 \text{ K})}{3000 \text{ kPa}} = \mathbf{0.4988 \text{ m}^3/\text{s}}$$

4-111 The cylinder conditions before the heat addition process is specified. The temperature after the heat addition process is to be determined.

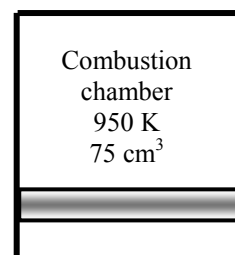
Assumptions 1 The contents of cylinder is approximated by the air properties. 2 Air is an ideal gas.

Analysis The ratio of the initial to the final mass is

$$\frac{m_1}{m_2} = \frac{AF}{AF + 1} = \frac{22}{22 + 1} = \frac{22}{23}$$

The final temperature may be determined from ideal gas relation

$$T_2 = \frac{m_1}{m_2} \frac{V_2}{V_1} T_1 = \left(\frac{22}{23} \right) \left(\frac{150 \text{ cm}^3}{75 \text{ cm}^3} \right) (950 \text{ K}) = \mathbf{1817 \text{ K}}$$



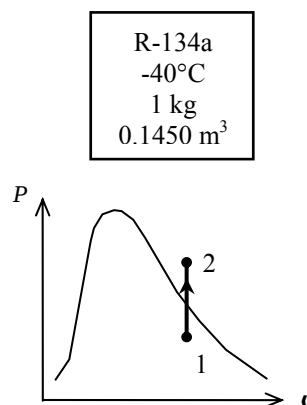
4-112 A rigid container that is filled with R-134a is heated. The initial pressure and the final temperature are to be determined.

Analysis The initial specific volume is $0.1450 \text{ m}^3/\text{kg}$. Using this with the initial temperature reveals that the initial state is a mixture. The initial pressure is then the saturation pressure,

$$\left. \begin{array}{l} T_1 = -40^\circ\text{C} \\ \nu_1 = 0.1450 \text{ m}^3/\text{kg} \end{array} \right\} P_1 = P_{\text{sat @ } -40^\circ\text{C}} = \mathbf{51.25 \text{ kPa}} \quad (\text{Table A-11})$$

This is a constant volume cooling process ($\nu = \nu/m = \text{constant}$). The final state is superheated vapor and the final temperature is then

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ \nu_2 = \nu_1 = 0.1450 \text{ m}^3/\text{kg} \end{array} \right\} T_2 = \mathbf{90^\circ\text{C}} \quad (\text{Table A-13})$$



4-113E A piston-cylinder device that is filled with water is cooled. The final pressure and volume of the water are to be determined.

Analysis The initial specific volume is

$$\nu_1 = \frac{V_1}{m} = \frac{2.649 \text{ ft}^3}{1 \text{ lbm}} = 2.649 \text{ ft}^3/\text{lbm}$$

This is a constant-pressure process. The initial state is determined to be superheated vapor and thus the pressure is determined to be

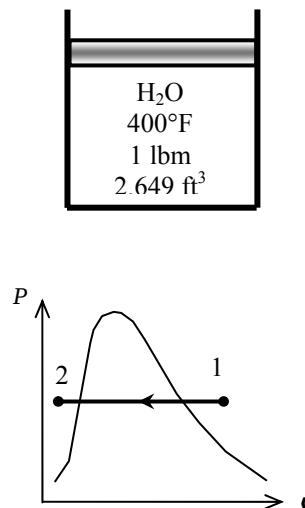
$$\left. \begin{array}{l} T_1 = 400^\circ\text{F} \\ \nu_1 = 2.649 \text{ ft}^3/\text{lbm} \end{array} \right\} P_1 = P_2 = \mathbf{180 \text{ psia}} \quad (\text{Table A-6E})$$

The saturation temperature at 180 psia is 373.1°F . Since the final temperature is less than this temperature, the final state is compressed liquid. Using the incompressible liquid approximation,

$$\nu_2 = \nu_f @ 100^\circ\text{F} = 0.01613 \text{ ft}^3/\text{lbm} \quad (\text{Table A-4E})$$

The final volume is then

$$V_2 = m\nu_2 = (1 \text{ lbm})(0.01613 \text{ ft}^3/\text{lbm}) = \mathbf{0.01613 \text{ ft}^3}$$



4-114 The volume of chamber 1 of the two-piston cylinder shown in the figure is to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The gas constant of helium is $R = 2.0769 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

Analysis Since the water vapor in chamber 2 is condensing, the pressure in this chamber is the saturation pressure,

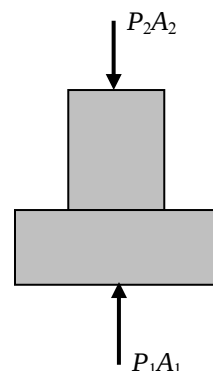
$$P_2 = P_{\text{sat @ } 200^\circ\text{C}} = 1555 \text{ kPa} \quad (\text{Table A-4})$$

Summing the forces acting on the piston in the vertical direction gives

$$P_1 = P_2 \frac{A_2}{A_1} = P_2 \left(\frac{D_2}{D_1} \right)^2 = (1555 \text{ kPa}) \left(\frac{4}{10} \right)^2 = 248.8 \text{ kPa}$$

According to the ideal gas equation of state,

$$V_1 = \frac{mRT}{P_1} = \frac{(1 \text{ kg})(2.0769 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(200 + 273 \text{ K})}{248.8 \text{ kPa}} = \mathbf{3.95 \text{ m}^3}$$



4-115 A propane tank contains 5 L of liquid propane at the ambient temperature. Now a leak develops at the top of the tank and propane starts to leak out. The temperature of propane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire propane in the tank is vaporized are to be determined.

Properties The properties of propane at 1 atm are $T_{\text{sat}} = -42.1^\circ\text{C}$, $\rho = 581 \text{ kg/m}^3$, and $h_{\text{fg}} = 427.8 \text{ kJ/kg}$ (Table A-3).

Analysis The temperature of propane when the pressure drops to 1 atm is simply the saturation pressure at that temperature,

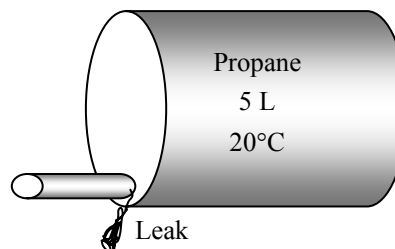
$$T = T_{\text{sat @ 1 atm}} = \mathbf{-42.1^\circ\text{C}}$$

The initial mass of liquid propane is

$$m = \rho V = (581 \text{ kg/m}^3)(0.005 \text{ m}^3) = 2.905 \text{ kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

$$Q_{\text{absorbed}} = mh_{\text{fg}} = (2.905 \text{ kg})(427.8 \text{ kJ/kg}) = \mathbf{1243 \text{ kJ}}$$



4-116 An isobutane tank contains 5 L of liquid isobutane at the ambient temperature. Now a leak develops at the top of the tank and isobutane starts to leak out. The temperature of isobutane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire isobutane in the tank is vaporized are to be determined.

Properties The properties of isobutane at 1 atm are $T_{\text{sat}} = -11.7^\circ\text{C}$, $\rho = 593.8 \text{ kg/m}^3$, and $h_{\text{fg}} = 367.1 \text{ kJ/kg}$ (Table A-3).

Analysis The temperature of isobutane when the pressure drops to 1 atm is simply the saturation pressure at that temperature,

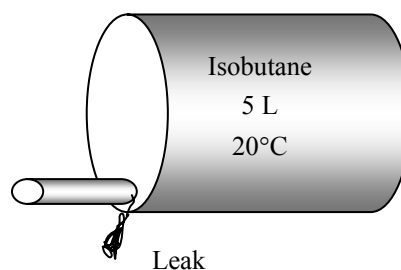
$$T = T_{\text{sat}@1 \text{ atm}} = -11.7^\circ\text{C}$$

The initial mass of liquid isobutane is

$$m = \rho V = (593.8 \text{ kg/m}^3)(0.005 \text{ m}^3) = 2.969 \text{ kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

$$Q_{\text{absorbed}} = m h_{\text{fg}} = (2.969 \text{ kg})(367.1 \text{ kJ/kg}) = \mathbf{1090 \text{ kJ}}$$



4-117 A tank contains helium at a specified state. Heat is transferred to helium until it reaches a specified temperature. The final gage pressure of the helium is to be determined.

Assumptions 1 Helium is an ideal gas.

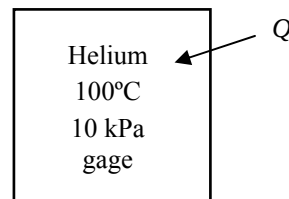
Properties The local atmospheric pressure is given to be 100 kPa.

Analysis Noting that the specific volume of helium in the tank remains constant, from ideal gas relation, we have

$$P_2 = P_1 \frac{T_2}{T_1} = (10 + 100 \text{ kPa}) \frac{(300 + 273)\text{K}}{(100 + 273)\text{K}} = 169.0 \text{ kPa}$$

Then the gage pressure becomes

$$P_{\text{gage},2} = P_2 - P_{\text{atm}} = 169.0 - 100 = \mathbf{69.0 \text{ kPa}}$$



4-118 ... 4-119 Design and Essay Problems



Solutions Manual

for

Introduction to Thermodynamics and Heat Transfer**Yunus A. Cengel****2nd Edition, 2008****Chapter 5****ENERGY ANALYSIS OF CLOSED
SYSTEMS****PROPRIETARY AND CONFIDENTIAL**

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Moving Boundary Work

5-1C It represents the boundary work for quasi-equilibrium processes.

5-2C Yes.

5-3C The area under the process curve, and thus the boundary work done, is greater in the constant pressure case.

$$\mathbf{5-4C} \quad 1 \text{ kPa} \cdot \text{m}^3 = 1 \text{ k}(\text{N}/\text{m}^2) \cdot \text{m}^3 = 1 \text{ kN} \cdot \text{m} = 1 \text{ kJ}$$

5-5 Helium is compressed in a piston-cylinder device. The initial and final temperatures of helium and the work required to compress it are to be determined.

Assumptions The process is quasi-equilibrium.

Properties The gas constant of helium is $R = 2.0769 \text{ kJ/kg} \cdot \text{K}$ (Table A-1).

Analysis The initial specific volume is

$$\nu_1 = \frac{V_1}{m} = \frac{5 \text{ m}^3}{1 \text{ kg}} = 5 \text{ m}^3/\text{kg}$$

Using the ideal gas equation,

$$T_1 = \frac{P_1 \nu_1}{R} = \frac{(200 \text{ kPa})(5 \text{ m}^3/\text{kg})}{2.0769 \text{ kJ/kg} \cdot \text{K}} = \mathbf{481.5 \text{ K}}$$

Since the pressure stays constant,

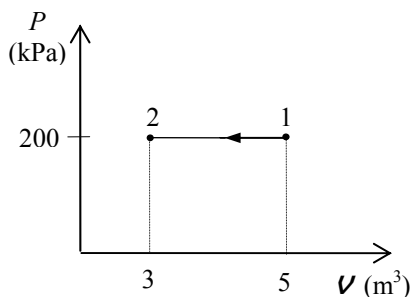
$$T_2 = \frac{\nu_2}{\nu_1} T_1 = \frac{3 \text{ m}^3}{5 \text{ m}^3} (481.5 \text{ K}) = \mathbf{288.9 \text{ K}}$$

and the work integral expression gives

$$W_{b,\text{out}} = \int_1^2 P d\nu = P(\nu_2 - \nu_1) = (200 \text{ kPa})(3 - 5) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = -400 \text{ kJ}$$

That is,

$$W_{b,\text{in}} = \mathbf{400 \text{ kJ}}$$

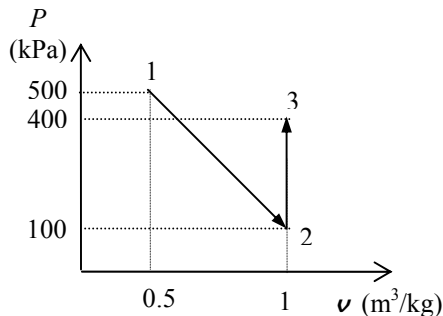


5-6 The boundary work done during the process shown in the figure is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis No work is done during the process 2-3 since the area under process line is zero. Then the work done is equal to the area under the process line 1-2:

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2} m(\nu_2 - \nu_1) \\ &= \frac{(100 + 500)\text{kPa}}{2} (2\text{ kg})(1.0 - 0.5)\text{m}^3/\text{kg} \left(\frac{1\text{ kJ}}{1\text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{300\text{ kJ}} \end{aligned}$$

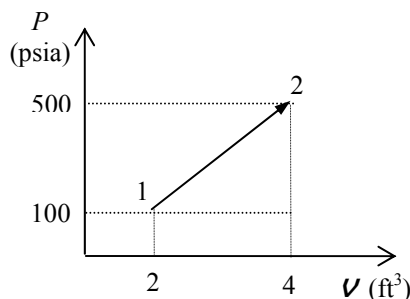


5-7E The boundary work done during the process shown in the figure is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The work done is equal to the area under the process line 1-2:

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2} (\nu_2 - \nu_1) \\ &= \frac{(100 + 500)\text{psia}}{2} (4.0 - 2.0)\text{ft}^3 \left(\frac{1\text{ Btu}}{5.404\text{ psia} \cdot \text{ft}^3} \right) \\ &= \mathbf{111\text{ Btu}} \end{aligned}$$



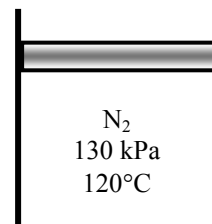
5-8 A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the polytropic expansion of nitrogen.

Properties The gas constant for nitrogen is 0.2968 kJ/kg·K (Table A-2).

Analysis The mass and volume of nitrogen at the initial state are

$$m = \frac{P_1 \nu_1}{RT_1} = \frac{(130\text{ kPa})(0.07\text{ m}^3)}{(0.2968\text{ kJ/kg} \cdot \text{K})(120 + 273\text{ K})} = 0.07802\text{ kg}$$

$$\nu_2 = \frac{mRT_2}{P_2} = \frac{(0.07802\text{ kg})(0.2968\text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(100 + 273\text{ K})}{100\text{ kPa}} = 0.08637\text{ m}^3$$



The polytropic index is determined from

$$P_1 \nu_1^n = P_2 \nu_2^n \longrightarrow (130\text{ kPa})(0.07\text{ m}^3)^n = (100\text{ kPa})(0.08637\text{ m}^3)^n \longrightarrow n = 1.249$$

The boundary work is determined from

$$W_b = \frac{P_2 \nu_2 - P_1 \nu_1}{1 - n} = \frac{(100\text{ kPa})(0.08637\text{ m}^3) - (130\text{ kPa})(0.07\text{ m}^3)}{1 - 1.249} = \mathbf{1.86\text{ kJ}}$$

5-9 A piston-cylinder device with a set of stops contains steam at a specified state. Now, the steam is cooled. The compression work for two cases and the final temperature are to be determined.

Analysis (a) The specific volumes for the initial and final states are (Table A-6)

$$\left. \begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \nu_1 = 0.30661 \text{ m}^3/\text{kg} \quad \left. \begin{array}{l} P_2 = 1 \text{ MPa} \\ T_2 = 250^\circ\text{C} \end{array} \right\} \nu_2 = 0.23275 \text{ m}^3/\text{kg}$$

Noting that pressure is constant during the process, the boundary work is determined from

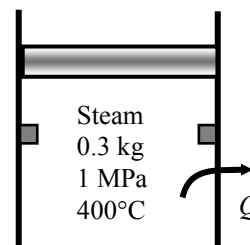
$$W_b = mP(\nu_1 - \nu_2) = (0.3 \text{ kg})(1000 \text{ kPa})(0.30661 - 0.23275) \text{ m}^3/\text{kg} = \mathbf{22.16 \text{ kJ}}$$

(b) The volume of the cylinder at the final state is 60% of initial volume. Then, the boundary work becomes

$$W_b = mP(\nu_1 - 0.60\nu_1) = (0.3 \text{ kg})(1000 \text{ kPa})(0.30661 - 0.60 \times 0.30661) \text{ m}^3/\text{kg} = \mathbf{36.79 \text{ kJ}}$$

The temperature at the final state is

$$\left. \begin{array}{l} P_2 = 0.5 \text{ MPa} \\ \nu_2 = (0.60 \times 0.30661) \text{ m}^3/\text{kg} \end{array} \right\} T_2 = \mathbf{151.8^\circ\text{C}} \quad (\text{Table A-5})$$



5-10 A piston-cylinder device contains nitrogen gas at a specified state. The final temperature and the boundary work are to be determined for the isentropic expansion of nitrogen.

Properties The properties of nitrogen are $R = 0.2968 \text{ kJ/kg}\cdot\text{K}$, $k = 1.4$ (Table A-2a)

Analysis The mass and the final volume of nitrogen are

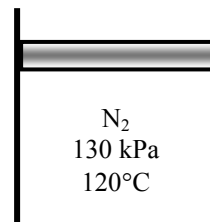
$$m = \frac{P_1 V_1}{RT_1} = \frac{(130 \text{ kPa})(0.07 \text{ m}^3)}{(0.2968 \text{ kJ/kg}\cdot\text{K})(120 + 273 \text{ K})} = 0.07802 \text{ kg}$$

$$P_1 V_1^k = P_2 V_2^k \longrightarrow (130 \text{ kPa})(0.07 \text{ m}^3)^{1.4} = (100 \text{ kPa})V_2^{1.4} \longrightarrow V_2 = 0.08443 \text{ m}^3$$

The final temperature and the boundary work are determined as

$$T_2 = \frac{P_2 V_2}{mR} = \frac{(100 \text{ kPa})(0.08443 \text{ m}^3)}{(0.07802 \text{ kg})(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{364.6 \text{ K}}$$

$$W_b = \frac{P_2 V_2 - P_1 V_1}{1 - k} = \frac{(100 \text{ kPa})(0.08443 \text{ m}^3) - (130 \text{ kPa})(0.07 \text{ m}^3)}{1 - 1.4} = \mathbf{1.64 \text{ kJ}}$$



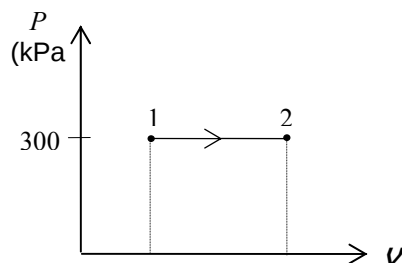
5-11 Saturated water vapor in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Properties Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4 through A-6)

$$\left. \begin{array}{l} P_1 = 300 \text{ kPa} \\ \text{Sat. vapor} \end{array} \right\} \nu_1 = \nu_{g@300 \text{ kPa}} = 0.60582 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} P_2 = 300 \text{ kPa} \\ T_2 = 200^\circ\text{C} \end{array} \right\} \nu_2 = 0.71643 \text{ m}^3/\text{kg}$$



Analysis The boundary work is determined from its definition to be

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P d\nu = P(\nu_2 - \nu_1) = mP(\nu_2 - \nu_1) \\ &= (5 \text{ kg})(300 \text{ kPa})(0.71643 - 0.60582) \text{ m}^3/\text{kg} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{165.9 \text{ kJ}} \end{aligned}$$

Discussion The positive sign indicates that work is done by the system (work output).

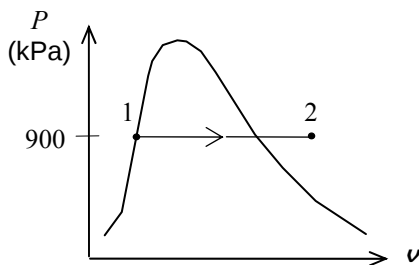
5-12 Refrigerant-134a in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Properties Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-11 through A-13)

$$\left. \begin{array}{l} P_1 = 900 \text{ kPa} \\ \text{Sat. liquid} \end{array} \right\} \nu_1 = \nu_{f@900 \text{ kPa}} = 0.0008580 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 70^\circ\text{C} \end{array} \right\} \nu_2 = 0.027413 \text{ m}^3/\text{kg}$$



Analysis The boundary work is determined from its definition to be

$$m = \frac{\nu_1}{\nu_1} = \frac{0.2 \text{ m}^3}{0.0008580 \text{ m}^3/\text{kg}} = 233.1 \text{ kg}$$

and

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P d\nu = P(\nu_2 - \nu_1) = mP(\nu_2 - \nu_1) \\ &= (233.1 \text{ kg})(900 \text{ kPa})(0.027413 - 0.0008580) \text{ m}^3/\text{kg} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{5571 \text{ kJ}} \end{aligned}$$

Discussion The positive sign indicates that work is done by the system (work output).

5-13 EES Problem 5-12 is reconsidered. The effect of pressure on the work done as the pressure varies from 400 kPa to 1200 kPa is to be investigated. The work done is to be plotted versus the pressure.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns"

Vol_1L=200 [L]

x_1=0 "saturated liquid state"

P=900 [kPa]

T_2=70 [C]

"Solution"

Vol_1=Vol_1L*convert(L,m^3)

"The work is the boundary work done by the R-134a during the constant pressure process."

W_boundary=P*(Vol_2-Vol_1)

"The mass is:"

Vol_1=m*v_1

v_1=volume(R134a,P=P,x=x_1)

Vol_2=m*v_2

v_2=volume(R134a,P=P,T=T_2)

"Plot information:"

v[1]=v_1

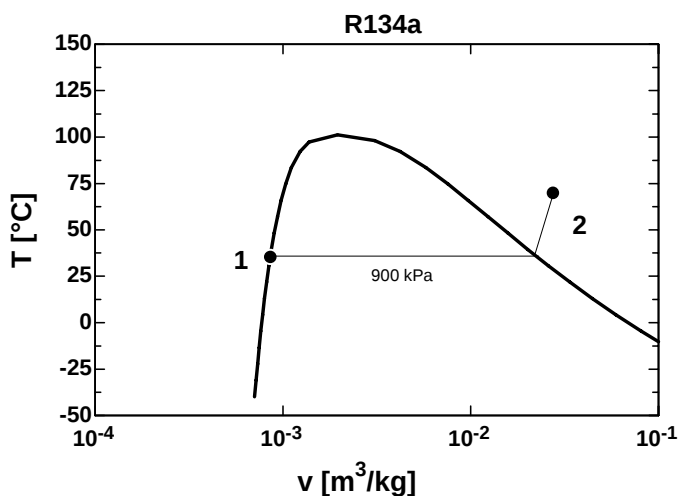
v[2]=v_2

P[1]=P

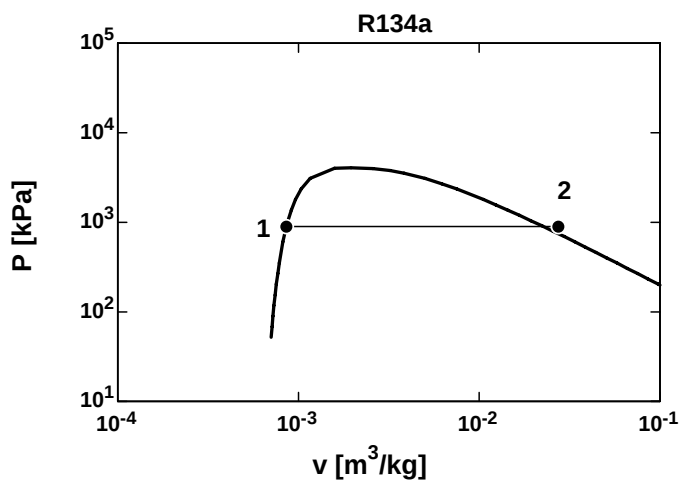
P[2]=P

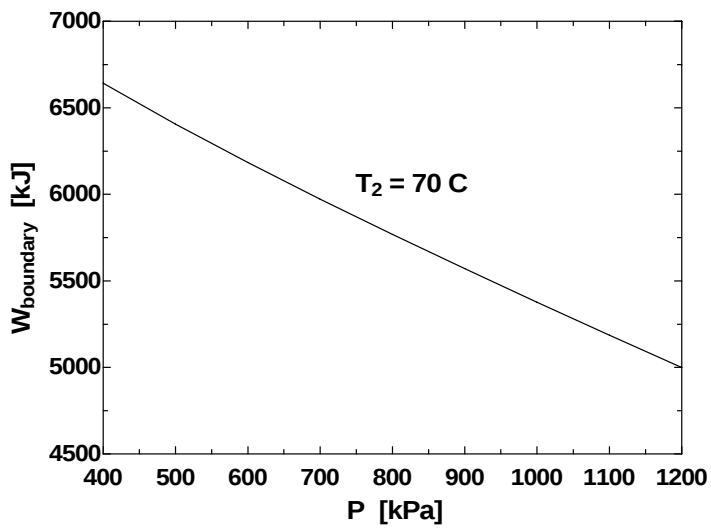
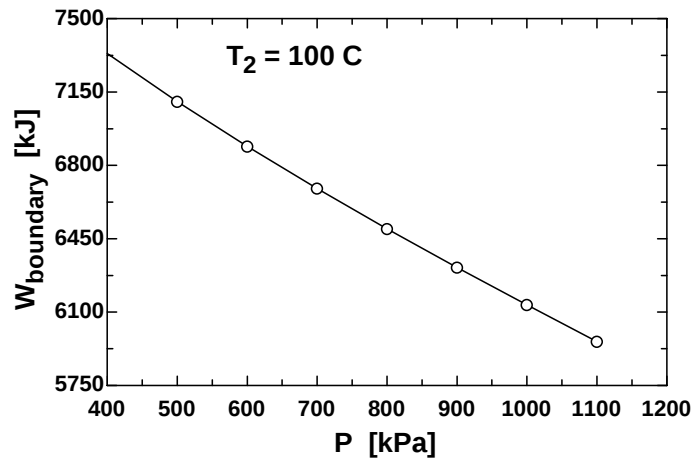
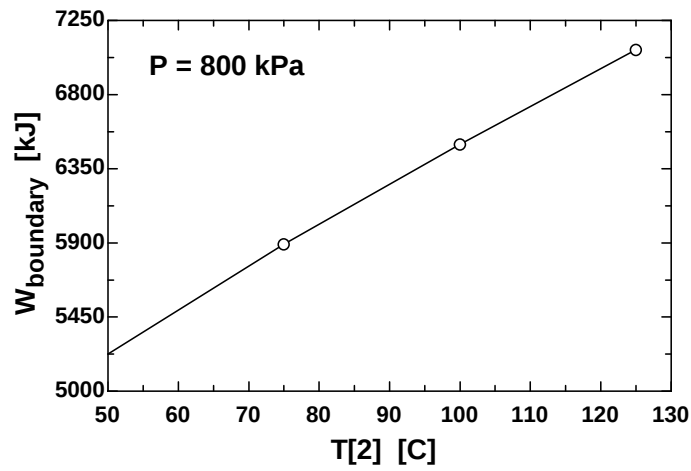
T[1]=temperature(R134a,P=P,x=x_1)

T[2]=T_2



P [kPa]	W _{boundary} [kJ]
400	6643
500	6405
600	6183
700	5972
800	5769
900	5571
1000	5377
1100	5187
1200	4999



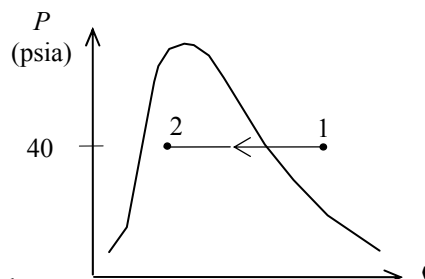


5-14E Superheated water vapor in a cylinder is cooled at constant pressure until 70% of it condenses. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Properties Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4E through A-6E)

$$\begin{aligned} P_1 = 40 \text{ psia} \\ T_1 = 600^\circ\text{F} \end{aligned} \left\{ \begin{aligned} v_1 &= 15.686 \text{ ft}^3/\text{lbm} \\ P_2 = 40 \text{ psia} \\ x_2 = 0.3 \end{aligned} \right\} \begin{aligned} v_2 &= v_f + x_2 v_{fg} \\ &= 0.01715 + 0.3(10.501 - 0.01715) \\ &= 3.1623 \text{ ft}^3/\text{lbm} \end{aligned}$$



Analysis The boundary work is determined from its definition to be

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = P(V_2 - V_1) = mP(v_2 - v_1) \\ &= (16 \text{ lbm})(40 \text{ psia})(3.1623 - 15.686) \text{ ft}^3/\text{lbm} \left(\frac{1 \text{ Btu}}{5.4039 \text{ psia} \cdot \text{ft}^3} \right) \\ &= -1483 \text{ Btu} \end{aligned}$$

Discussion The negative sign indicates that work is done on the system (work input).

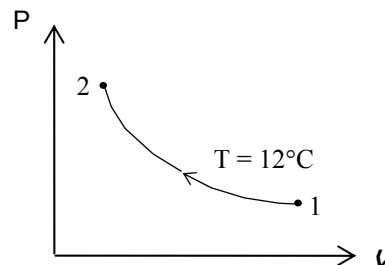
5-15 Air in a cylinder is compressed at constant temperature until its pressure rises to a specified value. The boundary work done during this process is to be determined.

Assumptions 1 The process is quasi-equilibrium. 2 Air is an ideal gas.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg} \cdot \text{K}$ (Table A-1).

Analysis The boundary work is determined from its definition to be

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = P_1 V_1 \ln \frac{V_2}{V_1} = mRT \ln \frac{P_1}{P_2} \\ &= (2.4 \text{ kg})(0.287 \text{ kJ/kg} \cdot \text{K})(285 \text{ K}) \ln \frac{150 \text{ kPa}}{600 \text{ kPa}} \\ &= -272 \text{ kJ} \end{aligned}$$



Discussion The negative sign indicates that work is done on the system (work input).

5-16E A gas in a cylinder is heated and is allowed to expand to a specified pressure in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

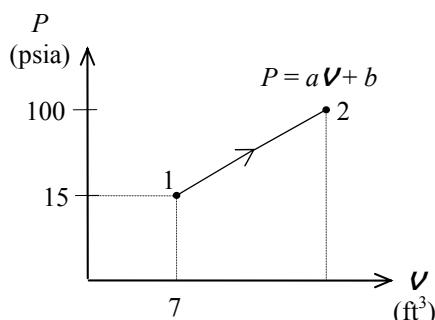
Analysis (a) The pressure of the gas changes linearly with volume, and thus the process curve on a P - V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

At state 1:

$$\begin{aligned} P_1 &= aV_1 + b \\ 15 \text{ psia} &= (5 \text{ psia/ft}^3)(7 \text{ ft}^3) + b \\ b &= -20 \text{ psia} \end{aligned}$$

At state 2:

$$\begin{aligned} P_2 &= aV_2 + b \\ 100 \text{ psia} &= (5 \text{ psia/ft}^3)V_2 + (-20 \text{ psia}) \\ V_2 &= 24 \text{ ft}^3 \end{aligned}$$



and,

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100 + 15) \text{ psia}}{2}(24 - 7) \text{ ft}^3 \left(\frac{1 \text{ Btu}}{5.4039 \text{ psia} \cdot \text{ft}^3} \right) \\ &= 181 \text{ Btu} \end{aligned}$$

Discussion The positive sign indicates that work is done by the system (work output).

5-17 CD EES A gas in a cylinder expands polytropically to a specified volume. The boundary work done during this process is to be determined.

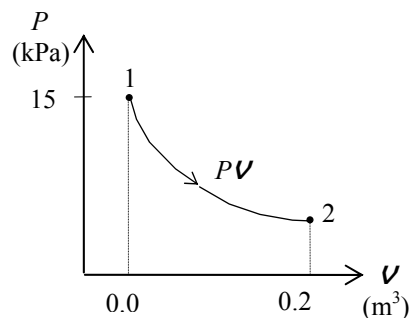
Assumptions The process is quasi-equilibrium.

Analysis The boundary work for this polytropic process can be determined directly from

$$P_2 = P_1 \left(\frac{V_1}{V_2} \right)^n = (150 \text{ kPa}) \left(\frac{0.03 \text{ m}^3}{0.2 \text{ m}^3} \right)^{1.3} = 12.74 \text{ kPa}$$

and,

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = \frac{P_2 V_2 - P_1 V_1}{1 - n} \\ &= \frac{(12.74 \times 0.2 - 150 \times 0.03) \text{ kPa} \cdot \text{m}^3}{1 - 1.3} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= 6.51 \text{ kJ} \end{aligned}$$



Discussion The positive sign indicates that work is done by the system (work output).

5-18 EES Problem 5-17 is reconsidered. The process described in the problem is to be plotted on a P - V diagram, and the effect of the polytropic exponent n on the boundary work as the polytropic exponent varies from 1.1 to 1.6 is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

Function BoundWork(P[1],V[1],P[2],V[2],n)

"This function returns the Boundary Work for the polytropic process. This function is required since the expression for boundary work depends on whether $n=1$ or $n \neq 1$ "

If $n \neq 1$ then

BoundWork:=(P[2]*V[2]-P[1]*V[1])/(1-n) "Use Equation 3-22 when $n \neq 1$ "

else

BoundWork:= P[1]*V[1]*ln(V[2]/V[1]) "Use Equation 3-20 when $n=1$ "

endif

end

"Inputs from the diagram window"

{ $n=1.3$

P[1] = 150 [kPa]

V[1] = 0.03 [m³]

V[2] = 0.2 [m³]

Gas\$='AIR'}

"System: The gas enclosed in the piston-cylinder device."

"Process: Polytropic expansion or compression, $P \cdot V^n = C$ "

$P[2] \cdot V[2]^n = P[1] \cdot V[1]^n$

" $n = 1.3$ " "Polytropic exponent"

"Input Data"

W_b = BoundWork(P[1],V[1],P[2],V[2],n) "[kJ]"

"If we modify this problem and specify the mass, then we can calculate the final temperature of the fluid for compression or expansion"

$m[1] = m[2]$ "Conservation of mass for the closed system"

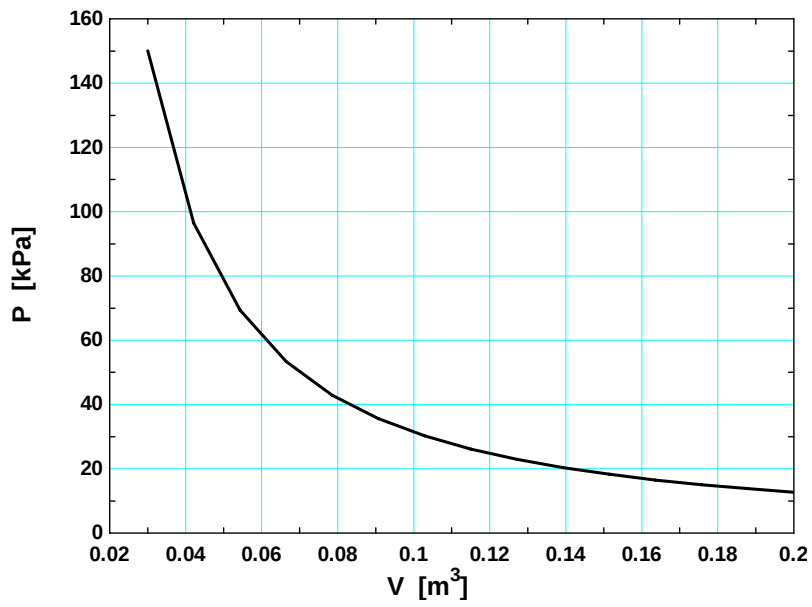
"Let's solve the problem for $m[1] = 0.05$ kg"

$m[1] = 0.05$ [kg]

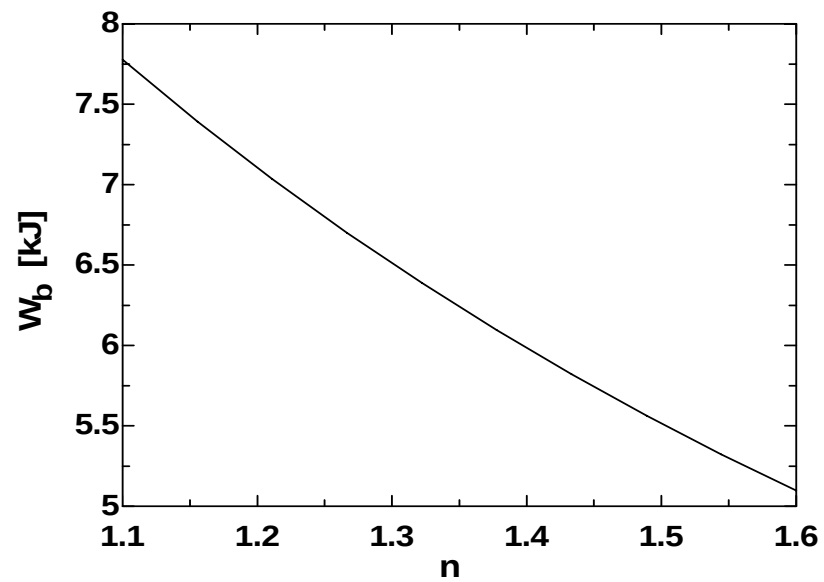
"Find the temperatures from the pressure and specific volume."

$T[1] = \text{temperature}(\text{gas}\$, P=P[1], v=V[1]/m[1])$

$T[2] = \text{temperature}(\text{gas}\$, P=P[2], v=V[2]/m[2])$



n	W_b [kJ]
1.1	7.776
1.156	7.393
1.211	7.035
1.267	6.7
1.322	6.387
1.378	6.094
1.433	5.82
1.489	5.564
1.544	5.323
1.6	5.097



5-19 Nitrogen gas in a cylinder is compressed polytropically until the temperature rises to a specified value. The boundary work done during this process is to be determined.

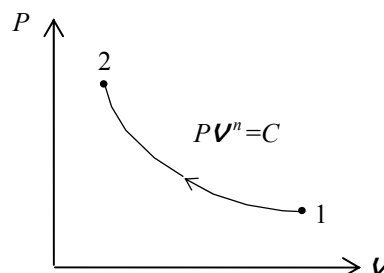
Assumptions 1 The process is quasi-equilibrium. **2** Nitrogen is an ideal gas.

Properties The gas constant for nitrogen is $R = 0.2968 \text{ kJ/kg} \cdot \text{K}$ (Table A-2a)

Analysis The boundary work for this polytropic process can be determined from

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = \frac{P_2 V_2 - P_1 V_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n} \\ &= \frac{(2 \text{ kg})(0.2968 \text{ kJ/kg} \cdot \text{K})(360 - 300) \text{ K}}{1 - 1.4} \\ &= -89.0 \text{ kJ} \end{aligned}$$

Discussion The negative sign indicates that work is done on the system (work input).



5-20 CD EES A gas whose equation of state is $\bar{v}(P + 10/\bar{v}^2) = R_u T$ expands in a cylinder isothermally to a specified volume. The unit of the quantity 10 and the boundary work done during this process are to be determined.

Assumptions The process is quasi-equilibrium.

Analysis (a) The term $10/\bar{v}^2$ must have pressure units since it is added to P .

Thus the quantity 10 must have the unit $\text{kPa} \cdot \text{m}^6/\text{kmol}^2$.

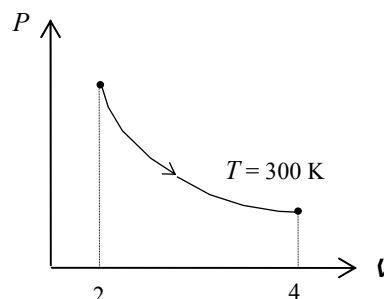
(b) The boundary work for this process can be determined from

$$P = \frac{R_u T}{\bar{v}} - \frac{10}{\bar{v}^2} = \frac{R_u T}{V/N} - \frac{10}{(V/N)^2} = \frac{NR_u T}{V} - \frac{10N^2}{V^2}$$

and

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = \int_1^2 \left(\frac{NR_u T}{V} - \frac{10N^2}{V^2} \right) dV = NR_u T \ln \frac{V_2}{V_1} + 10N^2 \left(\frac{1}{V_2} - \frac{1}{V_1} \right) \\ &= (0.5 \text{ kmol})(8.314 \text{ kJ/kmol} \cdot \text{K})(300 \text{ K}) \ln \frac{4 \text{ m}^3}{2 \text{ m}^3} \\ &\quad + (10 \text{ kPa} \cdot \text{m}^6/\text{kmol}^2)(0.5 \text{ kmol})^2 \left(\frac{1}{4 \text{ m}^3} - \frac{1}{2 \text{ m}^3} \right) \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= 864 \text{ kJ} \end{aligned}$$

Discussion The positive sign indicates that work is done by the system (work output).



5-21 EES Problem 5-20 is reconsidered. Using the integration feature, the work done is to be calculated and compared, and the process is to be plotted on a P - $\bar{v}$ diagram.

Analysis The problem is solved using EES, and the solution is given below.

"Input Data"

$N=0.5$ [kmol]

$v1_bar=2/N$ "[m^3/kmol]"

$v2_bar=4/N$ "[m^3/kmol]"

$T=300$ [K]

$R_u=8.314$ [kJ/kmol-K]

"The equation of state is:"

$v_bar*(P+10/v_bar^2)=R_u*T$ "P is in kPa"

"using the EES integral function, the boundary work, W_bEES , is"

$W_b_EES=N*integral(P,v_bar,v1_bar,v2_bar,0.01)$

"We can show that $W_bhand=$ integral of Pdv_bar is

(one should solve for $P=F(v_bar)$ and do the integral 'by hand' for practice)."

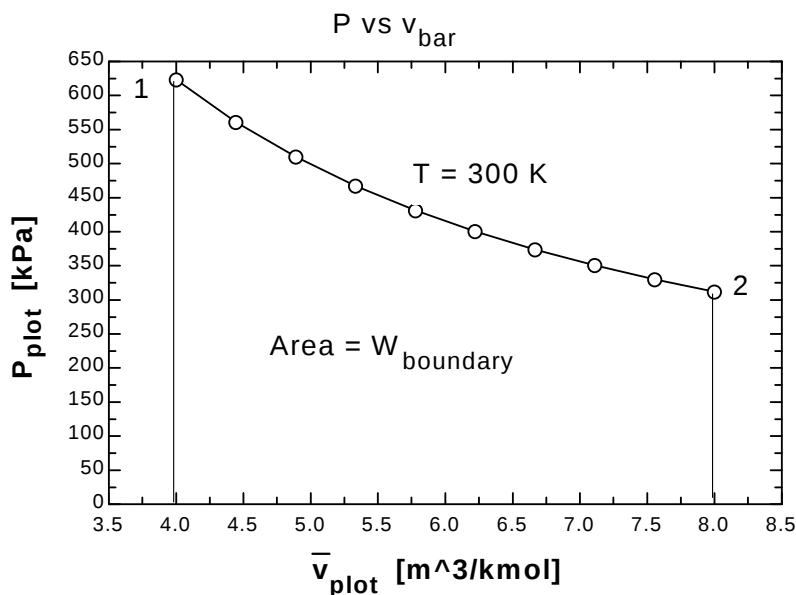
$W_b_hand = N*(R_u*T*\ln(v2_bar/v1_bar) + 10*(1/v2_bar-1/v1_bar))$

"To plot P vs v_bar , define $P_plot=f(v_bar_plot, T)$ as"

$\{v_bar_plot*(P_plot+10/v_bar_plot^2)=R_u*T\}$

" $P=P_plot$ and $v_bar=v_bar_plot$ just to generate the parametric table for plotting purposes. To plot P vs v_bar for a new temperature or v_bar_plot range, remove the '{' and '}' from the above equation, and reset the v_bar_plot values in the Parametric Table. Then press F3 or select Solve Table from the Calculate menu. Next select New Plot Window under the Plot menu to plot the new data."

P_{plot}	v_{plot}
622.9	4
560.7	4.444
509.8	4.889
467.3	5.333
431.4	5.778
400.6	6.222
373.9	6.667
350.5	7.111
329.9	7.556
311.6	8

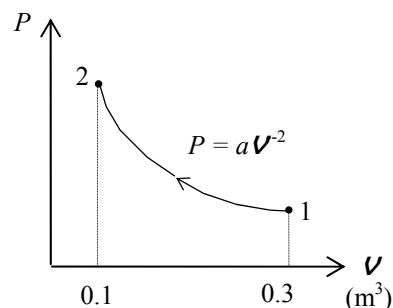


5-22 CO₂ gas in a cylinder is compressed until the volume drops to a specified value. The pressure changes during the process with volume as $P = aV^{-2}$. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The boundary work done during this process is determined from

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = \int_1^2 \left(\frac{a}{V^2} \right) dV = -a \left(\frac{1}{V_2} - \frac{1}{V_1} \right) \\ &= -(8 \text{ kPa} \cdot \text{m}^6) \left(\frac{1}{0.1 \text{ m}^3} - \frac{1}{0.3 \text{ m}^3} \right) \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= -53.3 \text{ kJ} \end{aligned}$$



Discussion The negative sign indicates that work is done on the system (work input).

5-23 Several sets of pressure and volume data are taken as a gas expands. The boundary work done during this process is to be determined using the experimental data.

Assumptions The process is quasi-equilibrium.

Analysis Plotting the given data on a P - V diagram on a graph paper and evaluating the area under the process curve, the work done is determined to be **0.25 kJ**.

5-24 A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the isothermal expansion of nitrogen.

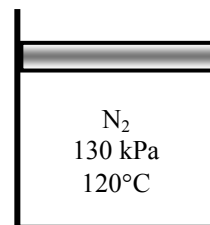
Properties The properties of nitrogen are $R = 0.2968 \text{ kJ/kg} \cdot \text{K}$, $k = 1.4$ (Table A-2a).

Analysis We first determine initial and final volumes from ideal gas relation, and find the boundary work using the relation for isothermal expansion of an ideal gas

$$V_1 = \frac{mRT}{P_1} = \frac{(0.25 \text{ kg})(0.2968 \text{ kJ/kg} \cdot \text{K})(120 + 273 \text{ K})}{(130 \text{ kPa})} = 0.2243 \text{ m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.25 \text{ kg})(0.2968 \text{ kJ/kg} \cdot \text{K})(120 + 273 \text{ K})}{(100 \text{ kPa})} = 0.2916 \text{ m}^3$$

$$W_b = P_1 V_1 \ln \left(\frac{V_2}{V_1} \right) = (130 \text{ kPa})(0.2243 \text{ m}^3) \ln \left(\frac{0.2916 \text{ m}^3}{0.2243 \text{ m}^3} \right) = \mathbf{7.65 \text{ kJ}}$$



5-25 A piston-cylinder device contains air gas at a specified state. The air undergoes a cycle with three processes. The boundary work for each process and the net work of the cycle are to be determined.

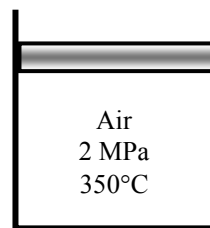
Properties The properties of air are $R = 0.287 \text{ kJ/kg}\cdot\text{K}$, $k = 1.4$ (Table A-2a).

Analysis For the isothermal expansion process:

$$V_1 = \frac{mRT}{P_1} = \frac{(0.15 \text{ kg})(0.287 \text{ kJ/kg}\cdot\text{K})(350 + 273 \text{ K})}{(2000 \text{ kPa})} = 0.01341 \text{ m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.15 \text{ kg})(0.287 \text{ kJ/kg}\cdot\text{K})(350 + 273 \text{ K})}{(500 \text{ kPa})} = 0.05364 \text{ m}^3$$

$$W_{b,1-2} = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (2000 \text{ kPa})(0.01341 \text{ m}^3) \ln\left(\frac{0.05364 \text{ m}^3}{0.01341 \text{ m}^3}\right) = \mathbf{37.18 \text{ kJ}}$$



For the polytropic compression process:

$$P_2 V_2^n = P_3 V_3^n \longrightarrow (500 \text{ kPa})(0.05364 \text{ m}^3)^{1.2} = (2000 \text{ kPa}) V_3^{1.2} \longrightarrow V_3 = 0.01690 \text{ m}^3$$

$$W_{b,2-3} = \frac{P_3 V_3 - P_2 V_2}{1 - n} = \frac{(2000 \text{ kPa})(0.01690 \text{ m}^3) - (500 \text{ kPa})(0.05364 \text{ m}^3)}{1 - 1.2} = \mathbf{-34.86 \text{ kJ}}$$

For the constant pressure compression process:

$$W_{b,3-1} = P_3 (V_1 - V_3) = (2000 \text{ kPa})(0.01341 - 0.01690) \text{ m}^3 = \mathbf{-6.97 \text{ kJ}}$$

The net work for the cycle is the sum of the works for each process

$$W_{\text{net}} = W_{b,1-2} + W_{b,2-3} + W_{b,3-1} = 37.18 + (-34.86) + (-6.97) = \mathbf{-4.65 \text{ kJ}}$$

5-26 A saturated water mixture contained in a spring-loaded piston-cylinder device is heated until the pressure and temperature rises to specified values. The work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The initial state is saturated mixture at 90°C. The pressure and the specific volume at this state are (Table A-4),

$$P_1 = 70.183 \text{ kPa}$$

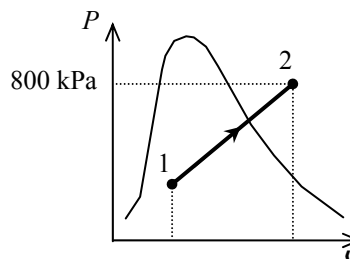
$$\begin{aligned} \nu_1 &= \nu_f + x\nu_{fg} \\ &= 0.001036 + (0.10)(2.3593 - 0.001036) \\ &= 0.23686 \text{ m}^3/\text{kg} \end{aligned}$$

The final specific volume at 800 kPa and 250°C is (Table A-6)

$$\nu_2 = 0.29321 \text{ m}^3/\text{kg}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2} m(\nu_2 - \nu_1) \\ &= \frac{(70.183 + 800) \text{ kPa}}{2} (1 \text{ kg})(0.29321 - 0.23686) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{24.52 \text{ kJ}} \end{aligned}$$



5-27 A saturated water mixture contained in a spring-loaded piston-cylinder device is cooled until it is saturated liquid at a specified temperature. The work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The initial state is saturated mixture at 1 MPa. The specific volume at this state is (Table A-5),

$$\begin{aligned} \nu_1 &= \nu_f + x\nu_{fg} \\ &= 0.001127 + (0.10)(0.19436 - 0.001127) \\ &= 0.020450 \text{ m}^3/\text{kg} \end{aligned}$$

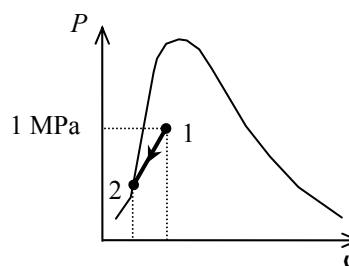
The final state is saturated liquid at 100°C (Table A-4)

$$P_2 = 101.42 \text{ kPa}$$

$$\nu_2 = \nu_f = 0.001043 \text{ m}^3/\text{kg}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2} m(\nu_2 - \nu_1) \\ &= \frac{(1000 + 101.42) \text{ kPa}}{2} (0.5 \text{ kg})(0.001043 - 0.020450) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{-5.34 \text{ kJ}} \end{aligned}$$



The negative sign shows that the work is done on the system in the amount of 5.34 kJ.

5-28 Argon is compressed in a polytropic process. The final temperature is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis For a polytropic expansion or compression process,

$$P\nu^n = \text{Constant}$$

For an ideal gas,

$$P\nu = RT$$

Combining these equations produces

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(n-1)/n} = (303 \text{ K}) \left(\frac{1200 \text{ kPa}}{120 \text{ kPa}} \right)^{0.2/1.2} = \mathbf{444.7 \text{ K}}$$

Closed System Energy Analysis

5-29 Saturated water vapor is isothermally condensed to a saturated liquid in a piston-cylinder device. The heat transfer and the work done are to be determined.

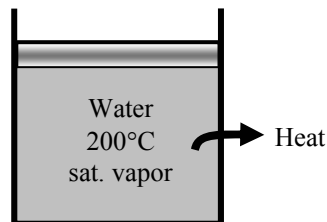
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$

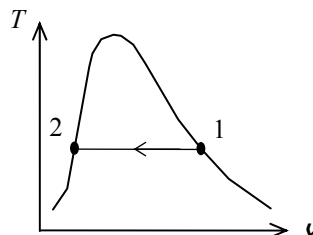


The properties at the initial and final states are (Table A-4)

$$\left. \begin{array}{l} T_1 = 200^\circ\text{C} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} v_1 = v_g = 0.12721 \text{ m}^3/\text{kg} \\ u_1 = u_g = 2594.2 \text{ kJ/kg} \end{array}$$

$$P_1 = P_2 = 1554.9 \text{ kPa}$$

$$\left. \begin{array}{l} T_2 = 200^\circ\text{C} \\ x_2 = 0 \end{array} \right\} \begin{array}{l} v_2 = v_f = 0.001157 \text{ m}^3/\text{kg} \\ u_2 = u_f = 850.46 \text{ kJ/kg} \end{array}$$



The work done during this process is

$$w_{b,\text{out}} = \int_1^2 P dV = P(v_2 - v_1) = (1554.9 \text{ kPa})(0.001157 - 0.12721) \text{ m}^3/\text{kg} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = -196.0 \text{ kJ/kg}$$

That is,

$$w_{b,\text{in}} = \mathbf{196.0 \text{ kJ/kg}}$$

Substituting the energy balance equation, we get

$$q_{\text{out}} = w_{b,\text{in}} - (u_2 - u_1) = w_{b,\text{in}} + u_{fg} = 196.0 + 1743.7 = \mathbf{1940 \text{ kJ/kg}}$$

5-30E The heat transfer during a process that a closed system undergoes without any internal energy change is to be determined.

Assumptions 1 The system is stationary and thus the kinetic and potential energy changes are zero. **2** The compression or expansion process is quasi-equilibrium.

Analysis The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{out}} = \Delta U = 0 \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{in}} = W_{\text{out}}$$

Then,

$$Q_{\text{in}} = 1.6 \times 10^6 \text{ lbf} \cdot \text{ft} \left(\frac{1 \text{ Btu}}{778.17 \text{ lbf} \cdot \text{ft}} \right) = \mathbf{2056 \text{ Btu}}$$

5-31 The table is to be completed using conservation of energy principle for a closed system.

Analysis The energy balance for a closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{out}} = E_2 - E_1 = m(e_2 - e_1)$$

Application of this equation gives the following completed table:

Q_{in} (kJ)	W_{out} (kJ)	E_1 (kJ)	E_2 (kJ)	m (kg)	$e_2 - e_1$ (kJ/kg)
280	440	1020	860	3	-53.3
-350	130	550	70	5	-96
-40	260	300	0	2	-150
300	550	750	500	1	-250
-400	-200	500	300	2	-100

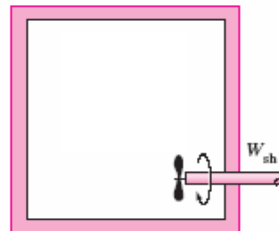
5-32 A substance is contained in a well-insulated, rigid container that is equipped with a stirring device. The change in the internal energy of this substance for a given work input is to be determined.

Assumptions 1 The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible.

Analysis This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{sh,in}} = \Delta U \quad (\text{since KE} = \text{PE} = 0)$$



Then,

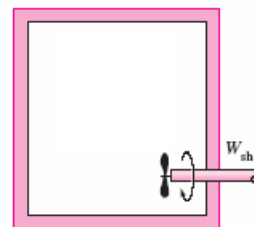
$$\Delta U = 15 \text{ kJ}$$

5-33 Motor oil is contained in a rigid container that is equipped with a stirring device. The rate of specific energy increase is to be determined.

Analysis This is a closed system since no mass enters or leaves. The energy balance for closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{sh,in}} = \Delta \dot{E}$$



Then,

$$\Delta \dot{E} = \dot{Q}_{\text{in}} + \dot{W}_{\text{sh,in}} = 1 + 1.5 = 2.5 = 2.5 \text{ W}$$

Dividing this by the mass in the system gives

$$\Delta \dot{e} = \frac{\Delta \dot{E}}{m} = \frac{2.5 \text{ J/s}}{1.5 \text{ kg}} = 1.67 \text{ J/kg} \cdot \text{s}$$

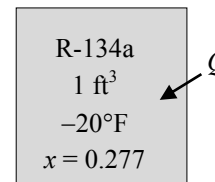
5-34E R-134a contained in a rigid vessel is heated. The heat transfer is to be determined.

Assumptions 1 The system is stationary and thus the kinetic and potential energy changes are zero. 2 There are no work interactions involved 3 The thermal energy stored in the vessel itself is negligible.

Analysis We take R-134a as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$



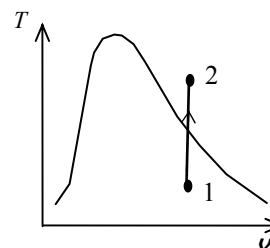
The properties at the initial and final states are (Tables A-11E, A-13E)

$$\left. \begin{array}{l} T_1 = -20^\circ\text{F} \\ x_1 = 0.277 \end{array} \right\} \begin{array}{l} v_1 = v_f + x v_{fg} = 0.01156 + (0.277)(3.4426 - 0.01156) = 0.96196 \text{ ft}^3 / \text{lbm} \\ u_1 = u_f + x u_{fg} = 6.019 + (0.277)(85.874) = 29.81 \text{ Btu/lbm} \end{array}$$

$$\left. \begin{array}{l} T_2 = 100^\circ\text{F} \\ v_2 = v_1 = 0.96196 \text{ ft}^3 / \text{lbm} \end{array} \right\} u_2 = 111.30 \text{ Btu/lbm}$$

Note that the final state is superheated vapor and the internal energy at this state should be obtained by interpolation using 50 psia and 60 psia mini tables (100°F line) in Table A-13E. The mass in the system is

$$m = \frac{v_1}{v_1} = \frac{1 \text{ ft}^3}{0.96196 \text{ ft}^3 / \text{lbm}} = 1.0395 \text{ lbm}$$



Substituting,

$$Q_{\text{in}} = m(u_2 - u_1) = (1.0395 \text{ lbm})(111.30 - 29.81) \text{ Btu/lbm} = \mathbf{84.7 \text{ Btu}}$$

5-35 An insulated rigid tank is initially filled with a saturated liquid-vapor mixture of water. An electric heater in the tank is turned on, and the entire liquid in the tank is vaporized. The length of time the heater was kept on is to be determined, and the process is to be shown on a P - ν diagram.

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The device is well-insulated and thus heat transfer is negligible. **3** The energy stored in the resistance wires, and the heat transferred to the tank itself is negligible.

Analysis We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$VI\Delta t = m(u_2 - u_1)$$

The properties of water are (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 0.25 \end{array} \right\} \begin{array}{l} \nu_f = 0.001043, \quad \nu_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, \quad u_{fg} = 2088.2 \text{ kJ/kg} \end{array}$$

$$\nu_1 = \nu_f + x_1 \nu_{fg} = 0.001043 + [0.25 \times (1.6941 - 0.001043)] = 0.42431 \text{ m}^3/\text{kg}$$

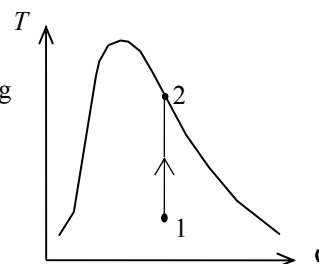
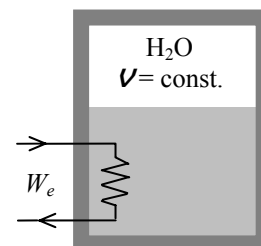
$$u_1 = u_f + x_1 u_{fg} = 417.40 + (0.25 \times 2088.2) = 939.4 \text{ kJ/kg}$$

$$\left. \begin{array}{l} \nu_2 = \nu_1 = 0.42431 \text{ m}^3/\text{kg} \\ \text{sat. vapor} \end{array} \right\} u_2 = u_{g@0.42431 \text{ m}^3/\text{kg}} = 2556.2 \text{ kJ/kg}$$

Substituting,

$$(110 \text{ V})(8 \text{ A})\Delta t = (5 \text{ kg})(2556.2 - 939.4) \text{ kJ/kg} \left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}} \right)$$

$$\Delta t = 9186 \text{ s} \cong \mathbf{153.1 \text{ min}}$$



5-36 EES Problem 5-35 is reconsidered. The effect of the initial mass of water on the length of time required to completely vaporize the liquid as the initial mass varies from 1 kg to 10 kg is to be investigated. The vaporization time is to be plotted against the initial mass.

Analysis The problem is solved using EES, and the solution is given below.

```
PROCEDURE P2X2(v[1]:P[2],x[2])
Fluid$='Steam_IAPWS'
If v[1] > V_CRIT(Fluid$) then
P[2]=pressure(Fluid$,v=v[1],x=1)
x[2]=1
else
P[2]=pressure(Fluid$,v=v[1],x=0)
x[2]=0
EndIf
End
```

"Knowns"

```
{m=5 [kg]}
P[1]=100 [kPa]
y=0.75 "moisture"
Volts=110 [V]
I=8 [amp]
```

"Solution"

"Conservation of Energy for the closed tank:"

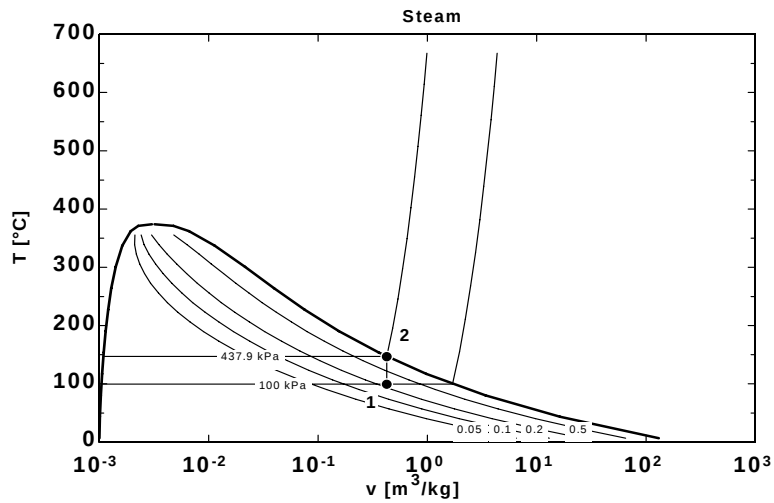
```
E_dot_in-E_dot_out=DELTA E_dot
E_dot_in=W_dot_ele "[kW]"
W_dot_ele=Volts*I*CONVERT(J/s,kW) "[kW]"
E_dot_out=0 "[kW]"
DELTA E_dot=m*(u[2]-u[1])/DELTA t_s "[kW]"
DELTA t_min=DELTA t_s*convert(s,min) "[min]"
```

"The quality at state 1 is:"

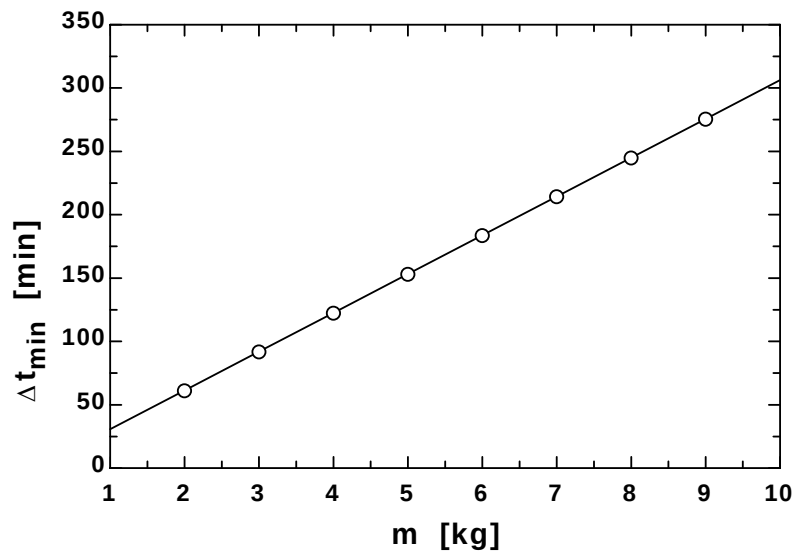
```
Fluid$='Steam_IAPWS'
x[1]=1-y
u[1]=INTENERGY(Fluid$,P=P[1], x=x[1]) "[kJ/kg]"
v[1]=volume(Fluid$,P=P[1], x=x[1]) "[m^3/kg]"
T[1]=temperature(Fluid$,P=P[1], x=x[1]) "[C]"
```

"Check to see if state 2 is on the saturated liquid line or saturated vapor line:"

```
Call P2X2(v[1]:P[2],x[2])
u[2]=INTENERGY(Fluid$,P=P[2], x=x[2]) "[kJ/kg]"
v[2]=volume(Fluid$,P=P[2], x=x[2]) "[m^3/kg]"
T[2]=temperature(Fluid$,P=P[2], x=x[2]) "[C]"
```



$\Delta t_{\min}$ [min]	m [kg]
30.63	1
61.26	2
91.89	3
122.5	4
153.2	5
183.8	6
214.4	7
245	8
275.7	9
306.3	10



5-37 A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled at constant pressure. The amount of heat loss is to be determined, and the process is to be shown on a T - ν diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$-Q_{\text{out}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are

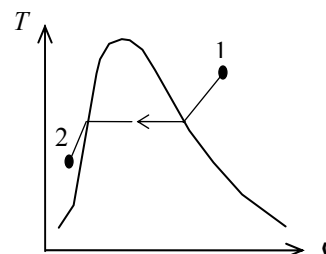
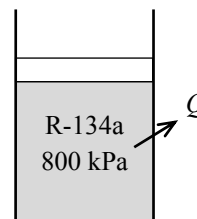
(Tables A-11 through A-13)

$$\left. \begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 70^\circ\text{C} \end{array} \right\} h_1 = 306.88 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 15^\circ\text{C} \end{array} \right\} h_2 \cong h_{f@15^\circ\text{C}} = 72.34 \text{ kJ/kg}$$

Substituting,

$$Q_{\text{out}} = - (5 \text{ kg})(72.34 - 306.88) \text{ kJ/kg} = \mathbf{1173 \text{ kJ}}$$



5-38E A cylinder contains water initially at a specified state. The water is heated at constant pressure. The final temperature of the water is to be determined, and the process is to be shown on a T - ν diagram.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-6E)

$$\nu_1 = \frac{V_1}{m} = \frac{2 \text{ ft}^3}{0.5 \text{ lbm}} = 4 \text{ ft}^3/\text{lbm}$$

$$\left. \begin{array}{l} P_1 = 120 \text{ psia} \\ \nu_1 = 4 \text{ ft}^3/\text{lbm} \end{array} \right\} h_1 = 1217.0 \text{ Btu/lbm}$$

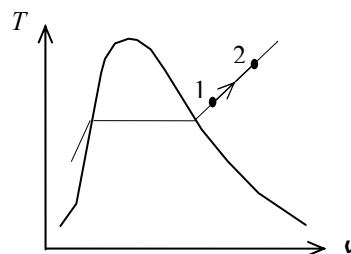
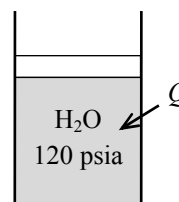
Substituting,

$$200 \text{ Btu} = (0.5 \text{ lbm})(h_2 - 1217.0) \text{ Btu/lbm}$$

$$h_2 = 1617.0 \text{ Btu/lbm}$$

Then,

$$\left. \begin{array}{l} P_2 = 120 \text{ psia} \\ h_2 = 1617.0 \text{ Btu/lbm} \end{array} \right\} T_2 = \mathbf{1161.4^\circ\text{F}}$$



5-39 A cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically as it is stirred by a paddle-wheel at constant pressure. The voltage of the current source is to be determined, and the process is to be shown on a P - ν diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} + W_{\text{pw},\text{in}} - W_{b,\text{out}} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{e,\text{in}} + W_{\text{pw},\text{in}} = m(h_2 - h_1)$$

$$(VI\Delta t) + W_{\text{pw},\text{in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_1 = 175 \text{ kPa} \\ \text{sat.liquid} \end{array} \right\} \begin{array}{l} h_1 = h_{f@175 \text{ kPa}} = 487.01 \text{ kJ/kg} \\ \nu_1 = \nu_{f@175 \text{ kPa}} = 0.001057 \text{ m}^3/\text{kg} \end{array}$$

$$\left. \begin{array}{l} P_2 = 175 \text{ kPa} \\ x_2 = 0.5 \end{array} \right\} \begin{array}{l} h_2 = h_f + x_2 h_{fg} = 487.01 + (0.5 \times 2213.1) = 1593.6 \text{ kJ/kg} \end{array}$$

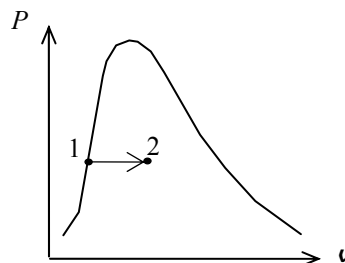
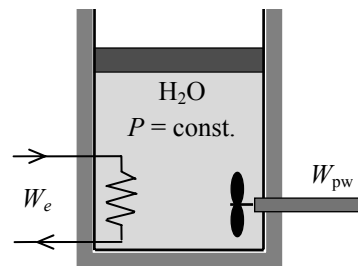
$$m = \frac{\nu_1}{\nu_1} = \frac{0.005 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} = 4.731 \text{ kg}$$

Substituting,

$$VI\Delta t + (400 \text{ kJ}) = (4.731 \text{ kg})(1593.6 - 487.01) \text{ kJ/kg}$$

$$VI\Delta t = 4835 \text{ kJ}$$

$$V = \frac{4835 \text{ kJ}}{(8 \text{ A})(45 \times 60 \text{ s})} \left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}} \right) = \mathbf{223.9 \text{ V}}$$



5-40 CD EES A cylinder equipped with an external spring is initially filled with steam at a specified state. Heat is transferred to the steam, and both the temperature and pressure rise. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a P - v diagram.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. 2 The thermal energy stored in the cylinder itself is negligible. 3 The compression or expansion process is quasi-equilibrium. 4 The spring is a linear spring.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Noting that the spring is not part of the system (it is external), the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{in}} = m(u_2 - u_1) + W_{b,\text{out}}$$

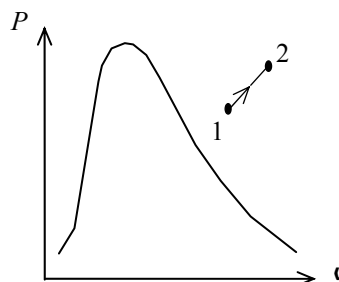
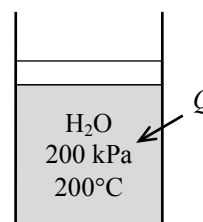
The properties of steam are (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.4628 \text{ kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6 \text{ m}^3}{0.4628 \text{ kg}} = 1.2966 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} P_2 = 500 \text{ kPa} \\ v_2 = 1.2966 \text{ m}^3/\text{kg} \end{array} \right\} \begin{array}{l} T_2 = \mathbf{1132^\circ\text{C}} \\ u_2 = 4325.2 \text{ kJ/kg} \end{array}$$



(b) The pressure of the gas changes linearly with volume, and thus the process curve on a P - V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$W_b = \text{Area} = \frac{P_1 + P_2}{2} (v_2 - v_1) = \frac{(200 + 500) \text{ kPa}}{2} (0.6 - 0.5) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{35 \text{ kJ}}$$

(c) From the energy balance we have

$$Q_{\text{in}} = (0.4628 \text{ kg})(4325.2 - 2654.6) \text{ kJ/kg} + 35 \text{ kJ} = \mathbf{808 \text{ kJ}}$$

5-41 EES Problem 5-40 is reconsidered. The effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer as the initial temperature varies from 150°C to 250°C is to be investigated. The final results are to be plotted against the initial temperature.

Analysis The problem is solved using EES, and the solution is given below.

"The process is given by:"

" $P[2]=P[1]+k \cdot x \cdot A/A$, and as the spring moves 'x' amount, the volume changes by $V[2]-V[1]$."

$P[2]=P[1]+(\text{Spring_const}) \cdot (V[2] - V[1])$ "P[2] is a linear function of V[2]"

"where $\text{Spring_const} = k/A$, the actual spring constant divided by the piston face area"

"Conservation of mass for the closed system is:"

$m[2]=m[1]$

"The conservation of energy for the closed system is"

" $E_{\text{in}} - E_{\text{out}} = \Delta E$, neglect ΔKE and ΔPE for the system"

$Q_{\text{in}} - W_{\text{out}} = m[1] \cdot (u[2] - u[1])$

$\Delta U = m[1] \cdot (u[2] - u[1])$

"Input Data"

$P[1]=200$ [kPa]

$V[1]=0.5$ [m³]

" $T[1]=200$ [C]"

$P[2]=500$ [kPa]

$V[2]=0.6$ [m³]

Fluid\$='Steam_IAPWS'

$m[1]=V[1]/\text{spvol}[1]$

$\text{spvol}[1]=\text{volume}(\text{Fluid}\$, T=T[1], P=P[1])$

$u[1]=\text{intenergy}(\text{Fluid}\$, T=T[1], P=P[1])$

$\text{spvol}[2]=V[2]/m[2]$

"The final temperature is:"

$T[2]=\text{temperature}(\text{Fluid}\$, P=P[2], v=\text{spvol}[2])$

$u[2]=\text{intenergy}(\text{Fluid}\$, P=P[2], T=T[2])$

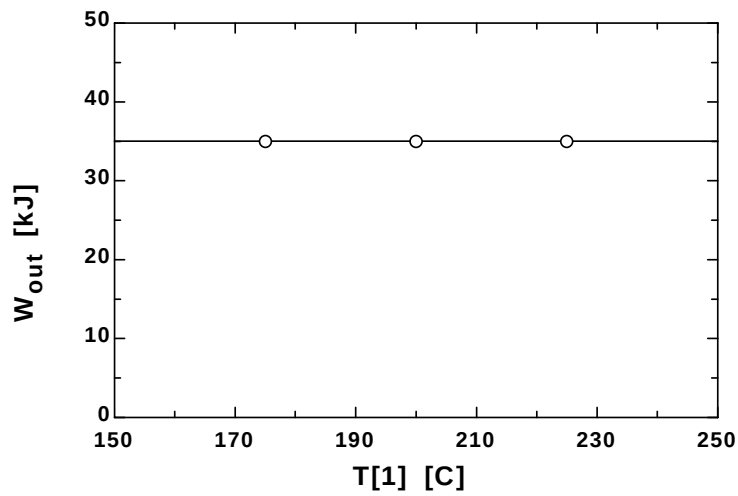
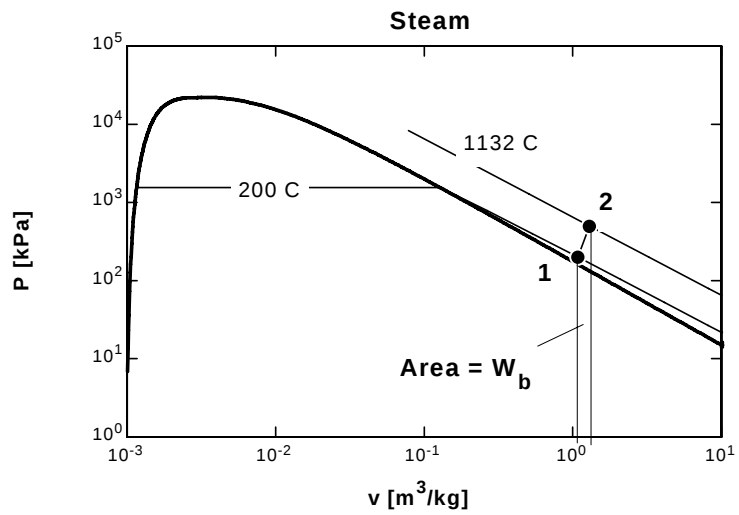
$W_{\text{net_other}} = 0$

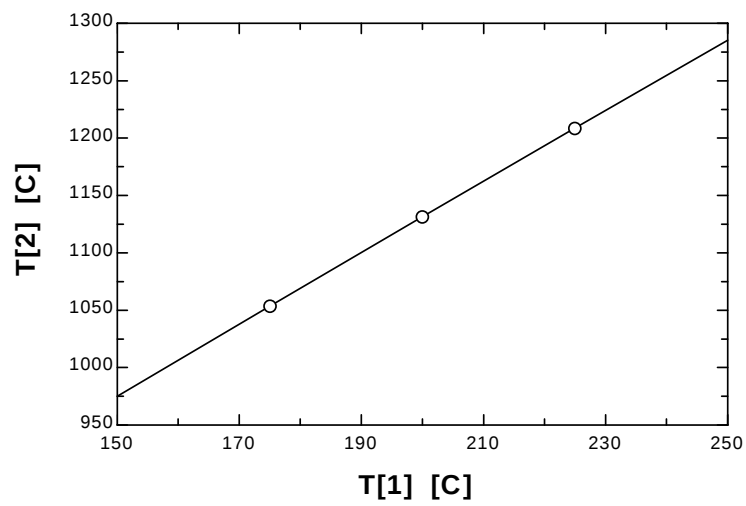
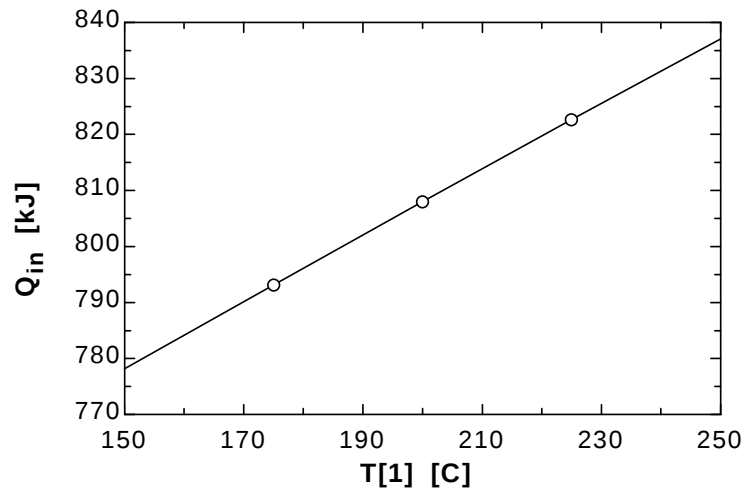
$W_{\text{out}}=W_{\text{net_other}} + W_b$

" $W_b = \text{integral of } P[2] \cdot dV[2] \text{ for } 0.5 < V[2] < 0.6 \text{ and is given by:}$ "

$W_b = P[1] \cdot (V[2] - V[1]) + \text{Spring_const}/2 \cdot (V[2] - V[1])^2$

Q_{in} [kJ]	T_1 [C]	T_2 [C]	W_{out} [kJ]
778.2	150	975	35
793.2	175	1054	35
808	200	1131	35
822.7	225	1209	35
837.1	250	1285	35

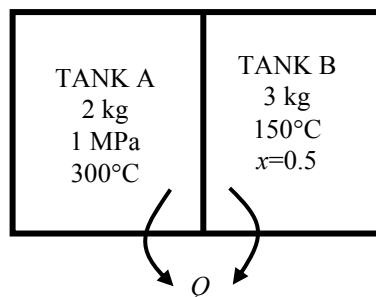




5-42 Two tanks initially separated by a partition contain steam at different states. Now the partition is removed and they are allowed to mix until equilibrium is established. The temperature and quality of the steam at the final state and the amount of heat lost from the tanks are to be determined.

Assumptions 1 The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

Analysis (a) We take the contents of both tanks as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{\text{out}} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_{1,A} = 1000 \text{ kPa} \\ T_{1,A} = 300^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_{1,A} = 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} = 2793.7 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} T_{1,B} = 150^\circ\text{C} \\ x_1 = 0.50 \end{array} \right\} \begin{array}{l} \nu_f = 0.001091, \quad \nu_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f = 631.66, \quad u_{fg} = 1927.4 \text{ kJ/kg} \end{array}$$

$$\nu_{1,B} = \nu_f + x_1 \nu_{fg} = 0.001091 + [0.50 \times (0.39248 - 0.001091)] = 0.19679 \text{ m}^3/\text{kg}$$

$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

The total volume and total mass of the system are

$$\nu = \nu_A + \nu_B = m_A \nu_{1,A} + m_B \nu_{1,B} = (2 \text{ kg})(0.25799 \text{ m}^3/\text{kg}) + (3 \text{ kg})(0.19679 \text{ m}^3/\text{kg}) = 1.106 \text{ m}^3$$

$$m = m_A + m_B = 3 + 2 = 5 \text{ kg}$$

Now, the specific volume at the final state may be determined

$$\nu_2 = \frac{\nu}{m} = \frac{1.106 \text{ m}^3}{5 \text{ kg}} = 0.22127 \text{ m}^3/\text{kg}$$

which fixes the final state and we can determine other properties

$$\left. \begin{array}{l} P_2 = 300 \text{ kPa} \\ \nu_2 = 0.22127 \text{ m}^3/\text{kg} \end{array} \right\} \begin{array}{l} T_2 = T_{\text{sat @ } 300 \text{ kPa}} = \mathbf{133.5^\circ\text{C}} \\ x_2 = \frac{\nu_2 - \nu_f}{\nu_g - \nu_f} = \frac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 = u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{array}$$

(b) Substituting,

$$\begin{aligned} -Q_{\text{out}} &= \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \\ &= (2 \text{ kg})(1282.8 - 2793.7) \text{ kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4) \text{ kJ/kg} = -3959 \text{ kJ} \end{aligned}$$

or $Q_{\text{out}} = \mathbf{3959 \text{ kJ}}$

5-43 A room is heated by an electrical radiator containing heating oil. Heat is lost from the room. The time period during which the heater is on is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa . **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa . **5** The room is air-tight so that no air leaks in and out during the process.

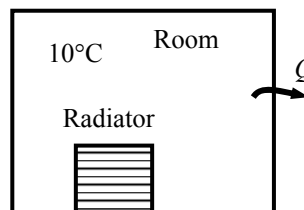
Properties The gas constant of air is $R = 0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). Also, $c_v = 0.718\text{ kJ/kg}\cdot\text{K}$ for air at room temperature (Table A-2). Oil properties are given to be $\rho = 950\text{ kg/m}^3$ and $c_p = 2.2\text{ kJ/kg}\cdot^{\circ}\text{C}$.

Analysis We take the air in the room and the oil in the radiator to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$(\dot{W}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = \Delta U_{\text{air}} + \Delta U_{\text{oil}}$$

$$\cong [mc_v(T_2 - T_1)]_{\text{air}} + [mc_p(T_2 - T_1)]_{\text{oil}} \quad (\text{since } KE = PE = 0)$$



The mass of air and oil are

$$m_{\text{air}} = \frac{P\mathcal{V}_{\text{air}}}{RT_1} = \frac{(100\text{ kPa})(50\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(10 + 273\text{ K})} = 62.32\text{ kg}$$

$$m_{\text{oil}} = \rho_{\text{oil}}\mathcal{V}_{\text{oil}} = (950\text{ kg/m}^3)(0.030\text{ m}^3) = 28.50\text{ kg}$$

Substituting,

$$(1.8 - 0.35\text{ kJ/s})\Delta t = (62.32\text{ kg})(0.718\text{ kJ/kg}\cdot^{\circ}\text{C})(20 - 10)^{\circ}\text{C} + (28.50\text{ kg})(2.2\text{ kJ/kg}\cdot^{\circ}\text{C})(50 - 10)^{\circ}\text{C}$$

$$\longrightarrow \Delta t = \mathbf{2038\text{ s} = 34.0\text{ min}}$$

Discussion In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to use ΔH instead of use ΔU in heating and air-conditioning applications.

5-44 Saturated liquid water is heated at constant pressure to a saturated vapor in a piston-cylinder device. The heat transfer is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{in}} = W_{b,\text{out}} + m(u_2 - u_1)$$

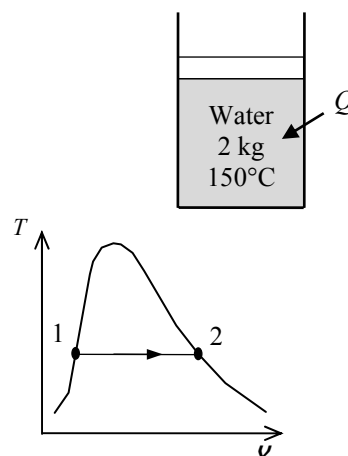
$$Q_{\text{in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Since water changes from saturated liquid to saturated vapor, we have

$$Q_{\text{in}} = mh_{fg} = (2 \text{ kg})(2113.8 \text{ kJ/kg}) = \mathbf{4228 \text{ kJ}}$$

since

$$h_{fg @ 150^\circ\text{C}} = 2113.8 \text{ kJ/kg} \quad (\text{Table A - 4})$$



5-45 A saturated water mixture contained in a spring-loaded piston-cylinder device is heated until the pressure and volume rise to specified values. The heat transfer and the work done are to be determined.

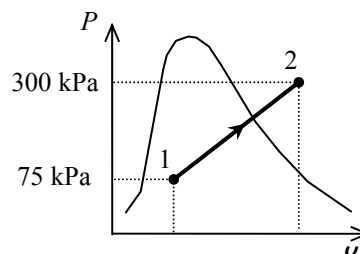
Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{in}} = W_{b,\text{out}} + m(u_2 - u_1)$$



The initial state is saturated mixture at 75 kPa. The specific volume and internal energy at this state are (Table A-5),

$$v_1 = v_f + x v_{fg} = 0.001037 + (0.13)(2.2172 - 0.001037) = 0.28914 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x u_{fg} = 384.36 + (0.13)(2111.8) = 658.89 \text{ kJ/kg}$$

The mass of water is

$$m = \frac{V_1}{v_1} = \frac{2 \text{ m}^3}{0.28914 \text{ m}^3/\text{kg}} = 6.9170 \text{ kg}$$

The final specific volume is

$$v_2 = \frac{V_2}{m} = \frac{5 \text{ m}^3}{6.9170 \text{ kg}} = 0.72285 \text{ m}^3/\text{kg}$$

The final state is now fixed. The internal energy at this specific volume and 300 kPa pressure is (Table A-6)

$$u_2 = 2657.2 \text{ kJ/kg}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2} (v_2 - v_1) = \frac{(75 + 300) \text{ kPa}}{2} (5 - 2) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{562.5 \text{ kJ}}$$

Substituting into energy balance equation gives

$$Q_{\text{in}} = W_{b,\text{out}} + m(u_2 - u_1) = 562.5 \text{ kJ} + (6.9170 \text{ kg})(2657.2 - 658.89) \text{ kJ/kg} = \mathbf{14,385 \text{ kJ}}$$

5-46 R-134a contained in a spring-loaded piston-cylinder device is cooled until the temperature and volume drop to specified values. The heat transfer and the work done are to be determined.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$

The initial state properties are (Table A-13)

$$\left. \begin{array}{l} P_1 = 600 \text{ kPa} \\ T_1 = 15^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.055522 \text{ m}^3/\text{kg} \\ u_1 = 357.96 \text{ kJ/kg} \end{array}$$

The mass of refrigerant is

$$m = \frac{v_1}{v_1} = \frac{0.3 \text{ m}^3}{0.055522 \text{ m}^3/\text{kg}} = 5.4033 \text{ kg}$$

The final specific volume is

$$v_2 = \frac{v_2}{m} = \frac{0.1 \text{ m}^3}{5.4033 \text{ kg}} = 0.018507 \text{ m}^3/\text{kg}$$

The final state at this specific volume and at -30°C is a saturated mixture. The properties at this state are (Table A-11)

$$x_2 = \frac{v_2 - v_f}{v_g - v_f} = \frac{0.018507 - 0.0007203}{0.22580 - 0.0007203} = 0.079024$$

$$u_2 = u_f + x_2 u_{fg} = 12.59 + (0.079024)(200.52) = 28.44 \text{ kJ/kg}$$

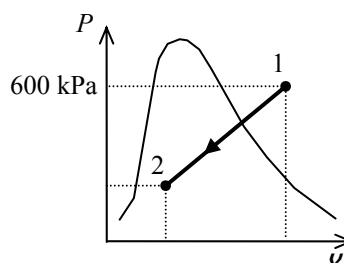
$$P_2 = 84.43 \text{ kPa}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$W_{b,\text{in}} = \text{Area} = \frac{P_1 + P_2}{2} (v_1 - v_2) = \frac{(600 + 84.43) \text{ kPa}}{2} (0.3 - 0.1) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{68.44 \text{ kJ}}$$

Substituting into energy balance equation gives

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1) = 68.44 \text{ kJ} - (5.4033 \text{ kg})(28.44 - 357.96) \text{ kJ/kg} = \mathbf{1849 \text{ kJ}}$$



5-47E Saturated R-134a vapor is condensed at constant pressure to a saturated liquid in a piston-cylinder device. The heat transfer and the work done are to be determined.

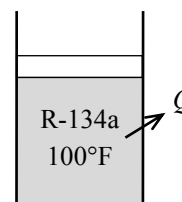
Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

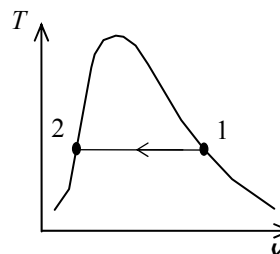
$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$



The properties at the initial and final states are (Table A-11E)

$$\left. \begin{array}{l} T_1 = 100^\circ\text{F} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} v_1 = v_g = 0.34045 \text{ ft}^3 / \text{lbm} \\ u_1 = u_g = 107.45 \text{ Btu/lbm} \end{array}$$

$$\left. \begin{array}{l} T_2 = 100^\circ\text{F} \\ x_2 = 0 \end{array} \right\} \begin{array}{l} v_2 = v_f = 0.01386 \text{ ft}^3 / \text{lbm} \\ u_2 = u_f = 44.768 \text{ Btu/lbm} \end{array}$$



Also from Table A-11E,

$$P_1 = P_2 = 138.93 \text{ psia}$$

$$u_{fg} = 62.683 \text{ Btu/lbm}$$

$$h_{fg} = 71.080 \text{ Btu/lbm}$$

The work done during this process is

$$w_{b,\text{out}} = \int_1^2 P d v = P(v_2 - v_1) = (138.93 \text{ psia})(0.01386 - 0.34045) \text{ ft}^3 / \text{lbm} \left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3} \right) = -8.396 \text{ Btu/lbm}$$

That is,

$$w_{b,\text{in}} = \mathbf{8.396 \text{ Btu/lbm}}$$

Substituting into energy balance equation gives

$$q_{\text{out}} = w_{b,\text{in}} - (u_2 - u_1) = w_{b,\text{in}} + u_{fg} = 8.396 + 62.683 = \mathbf{71.080 \text{ Btu/lbm}}$$

Discussion The heat transfer may also be determined from

$$-q_{\text{out}} = h_2 - h_1$$

$$q_{\text{out}} = h_{fg} = 71.080 \text{ Btu/lbm}$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.

5-48 Saturated R-134a liquid is contained in an insulated piston-cylinder device. Electrical work is supplied to R-134a. The time required for the refrigerant to turn into saturated vapor and the final temperature are to be determined.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

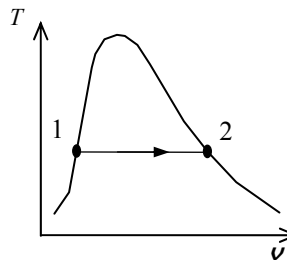
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$W_{e,\text{in}} = W_{b,\text{out}} + m(u_2 - u_1)$$

$$W_{e,\text{in}} = H_2 - H_1 = m(h_2 - h_1) = mh_{fg}$$

$$\dot{W}_{e,\text{in}} \Delta t = mh_{fg}$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The electrical power and the enthalpy of vaporization of R-134a are

$$\dot{W}_{e,\text{in}} = VI = (10 \text{ V})(2 \text{ A}) = 20 \text{ W}$$

$$h_{fg @ -10^\circ\text{C}} = 205.96 \text{ kJ/kg} \quad (\text{Table A-11})$$

Substituting,

$$(0.020 \text{ kJ/s})\Delta t = (2 \text{ kg})(205.96 \text{ kJ/kg}) \longrightarrow \Delta t = 20,596 \text{ s} = \mathbf{5.72 \text{ h}}$$

The temperature remains constant during this phase change process:

$$T_2 = T_1 = \mathbf{-10^\circ\text{C}}$$

Specific Heats, Δu , and Δh of Ideal Gases

5-49C It can be used for any kind of process of an ideal gas.

5-50C It can be used for any kind of process of an ideal gas.

5-51C The desired result is obtained by multiplying the first relation by the molar mass M ,

$$Mc_p = Mc_v + MR$$

or $\bar{c}_p = \bar{c}_v + R_u$

5-52C Very close, but no. Because the heat transfer during this process is $Q = mc_p\Delta T$, and c_p varies with temperature.

5-53C It can be either. The difference in temperature in both the K and °C scales is the same.

5-54C The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the c_p of an ideal gas does not vary with pressure.

5-55C The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the c_p of an ideal gas does not vary with volume.

5-56C Modeling both gases as ideal gases with constant specific heats, we have

$$\Delta u = c_v\Delta T$$

$$\Delta h = c_p\Delta T$$

Since both gases undergo the same temperature change, the gas with the greatest c_v will experience the largest change in internal energy. Similarly, the gas with the largest c_p will have the greatest enthalpy change. Inspection of Table A-2a indicates that air will experience the greatest change in both cases.

5-57 A spring-loaded piston-cylinder device is filled with nitrogen. Nitrogen is now heated until its volume increases by 10%. The changes in the internal energy and enthalpy of the nitrogen are to be determined.

Properties The gas constant of nitrogen is $R = 0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$. The specific heats of nitrogen at room temperature are $c_v = 0.743 \text{ kJ/kg} \cdot \text{K}$ and $c_p = 1.039 \text{ kJ/kg} \cdot \text{K}$ (Table A-2a).

Analysis The initial volume of nitrogen is

$$v_1 = \frac{mRT_1}{P_1} = \frac{(0.010 \text{ kg})(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(27 + 273 \text{ K})}{120 \text{ kPa}} = 0.00742 \text{ m}^3$$

The process experienced by this system is a linear P - v process. The equation for this line is

$$P - P_1 = c(v - v_1)$$

where P_1 is the system pressure when its specific volume is v_1 . The spring equation may be written as

$$P - P_1 = \frac{F_s - F_{s,1}}{A} = k \frac{x - x_1}{A} = \frac{kA}{A^2} (x - x_1) = \frac{k}{A^2} (v - v_1)$$

Constant c is hence

$$c = \frac{k}{A^2} = \frac{4^2 k}{\pi^2 D^4} = \frac{(16)(1 \text{ kN/m})}{\pi^2 (0.1 \text{ m})^4} = 16,211 \text{ kN/m}^5$$

The final pressure is then

$$\begin{aligned} P_2 &= P_1 + c(v_2 - v_1) = P_1 + c(1.1v_1 - v_1) = P_1 + 0.1c v_1 \\ &= 120 \text{ kPa} + 0.1(16,211 \text{ kN/m}^5)(0.00742 \text{ m}^3) \\ &= 132.0 \text{ kPa} \end{aligned}$$

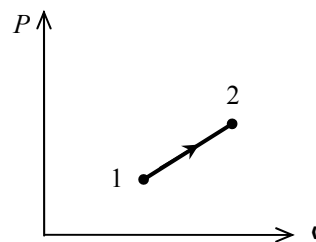
The final temperature is

$$T_2 = \frac{P_2 v_2}{mR} = \frac{(132.0 \text{ kPa})(1.1 \times 0.00742 \text{ m}^3)}{(0.010 \text{ kg})(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})} = 363 \text{ K}$$

Using the specific heats,

$$\Delta u = c_v \Delta T = (0.743 \text{ kJ/kg} \cdot \text{K})(363 - 300) \text{ K} = \mathbf{46.8 \text{ kJ/kg}}$$

$$\Delta h = c_p \Delta T = (1.039 \text{ kJ/kg} \cdot \text{K})(363 - 300) \text{ K} = \mathbf{65.5 \text{ kJ/kg}}$$



5-58E The internal energy change of air is to be determined for two cases of specified temperature changes.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The constant-volume specific heat of air at room temperature is $c_v = 0.171$ Btu/lbm·R (Table A-2Ea).

Analysis Using the specific heat at constant volume,

$$\Delta u = c_v \Delta T = (0.171 \text{ Btu/lbm} \cdot \text{R})(200 - 100)\text{R} = \mathbf{17.1 \text{ Btu/lbm}}$$

If we use the same room temperature specific heat value, the internal energy change will be the same for the second case. However, if we consider the variation of specific heat with temperature and use the specific heat values from Table A-2Eb, we have $c_v = 0.1725$ Btu/lbm·R at 150°F and $c_v = 0.1712$ Btu/lbm·R at 50°F. Then,

$$\Delta u_1 = c_v \Delta T_1 = (0.1725 \text{ Btu/lbm} \cdot \text{R})(200 - 100)\text{R} = \mathbf{17.25 \text{ Btu/lbm}}$$

$$\Delta u_2 = c_v \Delta T_2 = (0.1712 \text{ Btu/lbm} \cdot \text{R})(100 - 0)\text{R} = \mathbf{17.12 \text{ Btu/lbm}}$$

The two results differ from each other by about 0.8%.

5-59 The total internal energy changes for neon and argon are to be determined for a given temperature change.

Assumptions At specified conditions, neon and argon behave as an ideal gas.

Properties The constant-volume specific heats of neon and argon are 0.6179 kJ/kg·K and 0.3122 kJ/kg·K, respectively (Table A-2a).

Analysis The total internal energy changes are

$$\Delta U_{\text{neon}} = mc_v \Delta T = (2 \text{ kg})(0.6179 \text{ kJ/kg} \cdot \text{K})(180 - 20)\text{K} = \mathbf{129.7 \text{ kJ}}$$

$$\Delta U_{\text{argon}} = mc_v \Delta T = (2 \text{ kg})(0.3122 \text{ kJ/kg} \cdot \text{K})(180 - 20)\text{K} = \mathbf{99.9 \text{ kJ}}$$

5-60 The enthalpy changes for neon and argon are to be determined for a given temperature change.

Assumptions At specified conditions, neon and argon behave as an ideal gas.

Properties The constant-pressure specific heats of argon and neon are 0.5203 kJ/kg·K and 1.0299 kJ/kg·K, respectively (Table A-2a).

Analysis The enthalpy changes are

$$\Delta h_{\text{argon}} = c_p \Delta T = (0.5203 \text{ kJ/kg} \cdot \text{K})(400 - 100)\text{K} = \mathbf{156.1 \text{ kJ/kg}}$$

$$\Delta h_{\text{neon}} = c_p \Delta T = (1.0299 \text{ kJ/kg} \cdot \text{K})(400 - 100)\text{K} = \mathbf{309.0 \text{ kJ/kg}}$$

5-61 Neon is compressed isothermally in a compressor. The specific volume and enthalpy changes are to be determined.

Assumptions At specified conditions, neon behaves as an ideal gas.

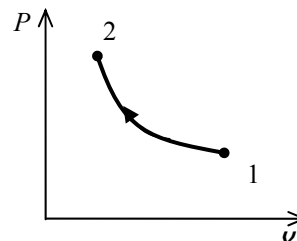
Properties The gas constant of neon is $R = 0.4119 \text{ kJ/kg}\cdot\text{K}$ and the constant-pressure specific heat of neon is $1.0299 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis At the compressor inlet, the specific volume is

$$\nu_1 = \frac{RT}{P_1} = \frac{(0.4119 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})}{100 \text{ kPa}} = 1.207 \text{ m}^3/\text{kg}$$

Similarly, at the compressor exit,

$$\nu_2 = \frac{RT}{P_2} = \frac{(0.4119 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})}{500 \text{ kPa}} = 0.2414 \text{ m}^3/\text{kg}$$



The change in the specific volume caused by the compressor is

$$\Delta\nu = \nu_2 - \nu_1 = 0.2414 - 1.207 = -0.966 \text{ m}^3/\text{kg}$$

Since the process is isothermal,

$$\Delta h = c_p \Delta T = 0 \text{ kJ/kg}$$

5-62E The enthalpy change of oxygen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

Analysis (a) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2Ec,

$$\bar{c}_p = a + bT + cT^2 + dT^3$$

where $a = 6.085$, $b = 0.2017 \times 10^{-2}$, $c = -0.05275 \times 10^{-5}$, and $d = 0.05372 \times 10^{-9}$. Then,

$$\begin{aligned}\Delta \bar{h} &= \int_1^2 \bar{c}_p(T) dT = \int_1^2 [a + bT + cT^2 + dT^3] dT \\ &= a(T_2 - T_1) + \frac{1}{2}b(T_2^2 - T_1^2) + \frac{1}{3}c(T_2^3 - T_1^3) + \frac{1}{4}d(T_2^4 - T_1^4) \\ &= 6.085(1500 - 800) + \frac{1}{2}(0.2017 \times 10^{-2})(1500^2 - 800^2) \\ &\quad - \frac{1}{3}(0.05275 \times 10^{-5})(1500^3 - 800^3) + \frac{1}{4}(0.05372 \times 10^{-9})(1500^4 - 800^4) \\ &= 5442.3 \text{ Btu/lbmol} \\ \Delta h &= \frac{\Delta \bar{h}}{M} = \frac{5442.3 \text{ Btu/lbmol}}{31.999 \text{ lbm/lbmol}} = \mathbf{170.1 \text{ Btu/lbm}}\end{aligned}$$

(b) Using the constant c_p value from Table A-2Eb at the average temperature of 1150 R,

$$\begin{aligned}c_{p,\text{avg}} &= c_{p@1150 \text{ R}} = 0.255 \text{ Btu/lbm} \cdot \text{R} \\ \Delta h &= c_{p,\text{avg}}(T_2 - T_1) = (0.255 \text{ Btu/lbm} \cdot \text{R})(1500 - 800) \text{ R} = \mathbf{178.5 \text{ Btu/lbm}}\end{aligned}$$

(c) Using the constant c_p value from Table A-2Ea at room temperature,

$$\begin{aligned}c_{p,\text{avg}} &= c_{p@537 \text{ R}} = 0.219 \text{ Btu/lbm} \cdot \text{R} \\ \Delta h &= c_{p,\text{avg}}(T_2 - T_1) = (0.219 \text{ Btu/lbm} \cdot \text{R})(1500 - 800) \text{ R} = \mathbf{153.3 \text{ Btu/lbm}}\end{aligned}$$

5-63 The internal energy change of hydrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

Analysis (a) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2c and relating it to $\bar{c}_v(T)$,

$$\bar{c}_v(T) = \bar{c}_p - R_u = (a - R_u) + bT + cT^2 + dT^3$$

where $a = 29.11$, $b = -0.1916 \times 10^{-2}$, $c = 0.4003 \times 10^{-5}$, and $d = -0.8704 \times 10^{-9}$. Then,

$$\begin{aligned} \Delta \bar{u} &= \int_1^2 \bar{c}_v(T) dT = \int_1^2 [(a - R_u) + bT + cT^2 + dT^3] dT \\ &= (a - R_u)(T_2 - T_1) + \frac{1}{2}b(T_2^2 - T_1^2) + \frac{1}{3}c(T_2^3 - T_1^3) + \frac{1}{4}d(T_2^4 - T_1^4) \\ &= (29.11 - 8.314)(800 - 200) - \frac{1}{2}(0.1961 \times 10^{-2})(800^2 - 200^2) \\ &\quad + \frac{1}{3}(0.4003 \times 10^{-5})(800^3 - 200^3) - \frac{1}{4}(0.8704 \times 10^{-9})(800^4 - 200^4) \\ &= 12,487 \text{ kJ/kmol} \\ \Delta u &= \frac{\Delta \bar{u}}{M} = \frac{12,487 \text{ kJ/kmol}}{2.016 \text{ kg/kmol}} = \mathbf{6194 \text{ kJ/kg}} \end{aligned}$$

(b) Using a constant c_p value from Table A-2b at the average temperature of 500 K,

$$\begin{aligned} c_{v,\text{avg}} &= c_{v@500 \text{ K}} = 10.389 \text{ kJ/kg} \cdot \text{K} \\ \Delta u &= c_{v,\text{avg}}(T_2 - T_1) = (10.389 \text{ kJ/kg} \cdot \text{K})(800 - 200) \text{ K} = \mathbf{6233 \text{ kJ/kg}} \end{aligned}$$

(c) Using a constant c_p value from Table A-2a at room temperature,

$$\begin{aligned} c_{v,\text{avg}} &= c_{v@300 \text{ K}} = 10.183 \text{ kJ/kg} \cdot \text{K} \\ \Delta u &= c_{v,\text{avg}}(T_2 - T_1) = (10.183 \text{ kJ/kg} \cdot \text{K})(800 - 200) \text{ K} = \mathbf{6110 \text{ kJ/kg}} \end{aligned}$$

Closed System Energy Analysis: Ideal Gases

5-64C No, it isn't. This is because the first law relation $Q - W = \Delta U$ reduces to $W = 0$ in this case since the system is adiabatic ($Q = 0$) and $\Delta U = 0$ for the isothermal processes of ideal gases. Therefore, this adiabatic system cannot receive any net work at constant temperature.

5-65E The air in a rigid tank is heated until its pressure doubles. The volume of the tank and the amount of heat transfer are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -221°F and 547 psia . **2** The kinetic and potential energy changes are negligible, $\Delta pe \cong \Delta ke \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

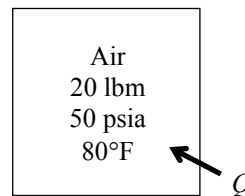
Properties The gas constant of air is $R = 0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$ (Table A-1E).

Analysis (a) The volume of the tank can be determined from the ideal gas relation,

$$\mathcal{V} = \frac{mRT_1}{P_1} = \frac{(20\text{ lbm})(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540\text{ R})}{50\text{ psia}} = \mathbf{80.0\text{ ft}^3}$$

(b) We take the air in the tank as our system. The energy balance for this stationary closed system can be expressed as

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} &= \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\ Q_{\text{in}} &= \Delta U \\ Q_{\text{in}} &= m(u_2 - u_1) \cong mc_v(T_2 - T_1) \end{aligned}$$



The final temperature of air is

$$\frac{P_1\mathcal{V}}{T_1} = \frac{P_2\mathcal{V}}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = 2 \times (540\text{ R}) = 1080\text{ R}$$

The internal energies are (Table A-21E)

$$u_1 = u_{@540\text{ R}} = 92.04\text{ Btu/lbm}$$

$$u_2 = u_{@1080\text{ R}} = 186.93\text{ Btu/lbm}$$

Substituting,

$$Q_{\text{in}} = (20\text{ lbm})(186.93 - 92.04)\text{Btu/lbm} = \mathbf{1898\text{ Btu}}$$

Alternative solutions The specific heat of air at the average temperature of $T_{\text{avg}} = (540 + 1080)/2 = 810\text{ R} = 350^\circ\text{F}$ is, from Table A-2Eb, $c_{v,\text{avg}} = 0.175\text{ Btu/lbm}\cdot\text{R}$. Substituting,

$$Q_{\text{in}} = (20\text{ lbm})(0.175\text{ Btu/lbm}\cdot\text{R})(1080 - 540)\text{ R} = \mathbf{1890\text{ Btu}}$$

Discussion Both approaches resulted in almost the same solution in this case.

5-66 A resistance heater is to raise the air temperature in the room from 7 to 23°C within 15 min. The required power rating of the resistance heater is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** Heat losses from the room are negligible. **5** The room is air-tight so that no air leaks in and out during the process.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1). Also, $c_v = 0.718 \text{ kJ/kg} \cdot \text{K}$ for air at room temperature (Table A-2).

Analysis We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

or,

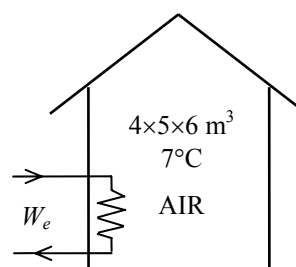
$$\dot{W}_{e,\text{in}} \Delta t = mc_{v,\text{avg}}(T_2 - T_1)$$

The mass of air is

$$\begin{aligned} V &= 4 \times 5 \times 6 = 120 \text{ m}^3 \\ m &= \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(280 \text{ K})} = 149.3 \text{ kg} \end{aligned}$$

Substituting, the power rating of the heater becomes

$$\dot{W}_{e,\text{in}} = \frac{(149.3 \text{ kg})(0.718 \text{ kJ/kg} \cdot ^\circ\text{C})(23 - 7)^\circ\text{C}}{15 \times 60 \text{ s}} = \mathbf{1.91 \text{ kW}}$$



Discussion In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to use ΔH instead of using ΔU in heating and air-conditioning applications.

5-67 A student living in a room turns her 150-W fan on in the morning. The temperature in the room when she comes back 10 h later is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa . **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded.

Properties The gas constant of air is $R = 0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). Also, $c_v = 0.718\text{ kJ/kg}\cdot\text{K}$ for air at room temperature (Table A-2).

Analysis We take the room as the system. This is a *closed system* since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

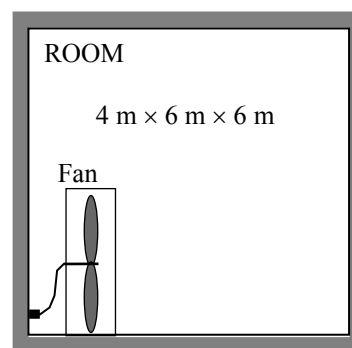
$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

The mass of air is

$$\mathcal{V} = 4 \times 6 \times 6 = 144\text{ m}^3$$

$$m = \frac{P_1 \mathcal{V}}{RT_1} = \frac{(100\text{ kPa})(144\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\text{ K})} = 174.2\text{ kg}$$



The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15\text{ kJ/s})(10 \times 3600\text{ s}) = 5400\text{ kJ}$$

Substituting and using the c_v value at room temperature,

$$5400\text{ kJ} = (174.2\text{ kg})(0.718\text{ kJ/kg}\cdot^{\circ}\text{C})(T_2 - 15)^{\circ}\text{C}$$

$$T_2 = \mathbf{58.2^{\circ}\text{C}}$$

Discussion Note that a fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room with as much energy as a 100-W resistance heater.

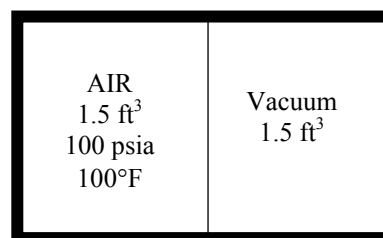
5-68E One part of an insulated rigid tank contains air while the other side is evacuated. The internal energy change of the air and the final air pressure are to be determined when the partition is removed.

Assumptions 1 Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -221.5°F and 547 psia. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. 3 The tank is insulated and thus heat transfer is negligible.

Analysis We take the entire tank as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = mc_v(T_2 - T_1)$$



Since the internal energy does not change, the temperature of the air will also not change. Applying the ideal gas equation gives

$$P_1 V_1 = P_2 V_2 \longrightarrow P_2 = P_1 \frac{V_1}{V_2} = P_1 \frac{V_2 / 2}{V_2} = \frac{P_1}{2} = \frac{100 \text{ psia}}{2} = \mathbf{50 \text{ psia}}$$

5-69 Nitrogen in a rigid vessel is cooled by rejecting heat. The internal energy change of the nitrogen is to be determined.

Assumptions 1 Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K and 3.39 MPa. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats at room temperature can be used for nitrogen.

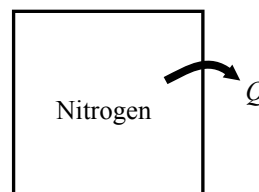
Analysis We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

Thus,

$$\Delta u = -q_{\text{out}} = \mathbf{-100 \text{ kJ/kg}}$$



5-70E Nitrogen in a rigid vessel is cooled. The work done and heat transfer are to be determined.

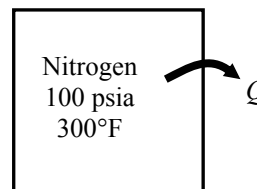
Assumptions 1 Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K (227.1 R) and 3.39 MPa (492 psia). 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats at room temperature can be used for nitrogen.

Properties For nitrogen, $c_v = 0.177$ Btu/lbm·R at room temperature (Table A-2Ea).

Analysis We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{\text{out}} = \Delta U = mc_v(T_2 - T_1)$$



There is no work done since the vessel is rigid:

$$w = \mathbf{0 \text{ Btu/lbm}}$$

Since the specific volume is constant during the process, the final temperature is determined from ideal gas equation to be

$$T_2 = T_1 \frac{P_2}{P_1} = (760 \text{ R}) \frac{50 \text{ psia}}{100 \text{ psia}} = 380 \text{ R}$$

Substituting,

$$q_{\text{out}} = c_v(T_1 - T_2) = (0.177 \text{ Btu/lbm} \cdot \text{R})(760 - 380) \text{ R} = \mathbf{67.3 \text{ Btu/lbm}}$$

5-71 Oxygen is heated to experience a specified temperature change. The heat transfer is to be determined for two cases.

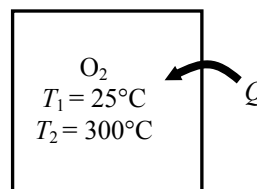
Assumptions 1 Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 154.8 K and 5.08 MPa. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats can be used for oxygen.

Properties The specific heats of oxygen at the average temperature of $(25+300)/2=162.5^\circ\text{C}=436\text{ K}$ are $c_p = 0.952\text{ kJ/kg}\cdot\text{K}$ and $c_v = 0.692\text{ kJ/kg}\cdot\text{K}$ (Table A-2b).

Analysis We take the oxygen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for a constant-volume process can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = \Delta U = mc_v(T_2 - T_1)$$



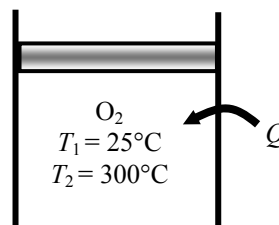
The energy balance during a constant-pressure process (such as in a piston-cylinder device) can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U$$

$$Q_{\text{in}} = W_{b,\text{out}} + \Delta U$$

$$Q_{\text{in}} = \Delta H = mc_p(T_2 - T_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting for both cases,

$$Q_{\text{in}, v=\text{const}} = mc_v(T_2 - T_1) = (1\text{ kg})(0.692\text{ kJ/kg}\cdot\text{K})(300 - 25)\text{K} = \mathbf{190.3\text{ kJ}}$$

$$Q_{\text{in}, p=\text{const}} = mc_p(T_2 - T_1) = (1\text{ kg})(0.952\text{ kJ/kg}\cdot\text{K})(300 - 25)\text{K} = \mathbf{261.8\text{ kJ}}$$

5-72 Air in a closed system undergoes an isothermal process. The initial volume, the work done, and the heat transfer are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 132.5 K and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for air.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

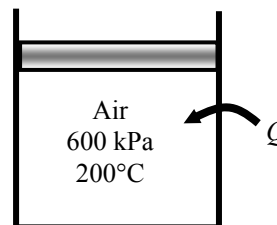
Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

$$Q_{\text{in}} - W_{b,\text{out}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{b,\text{out}}$$



The initial volume is

$$v_1 = \frac{mRT_1}{P_1} = \frac{(2 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(473 \text{ K})}{600 \text{ kPa}} = \mathbf{0.4525 \text{ m}^3}$$

Using the boundary work relation for the isothermal process of an ideal gas gives

$$W_{b,\text{out}} = m \int_1^2 P d v = mRT \int_1^2 \frac{d v}{v} = mRT \ln \frac{v_2}{v_1} = mRT \ln \frac{P_1}{P_2}$$

$$= (2 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(473 \text{ K}) \ln \frac{600 \text{ kPa}}{80 \text{ kPa}} = \mathbf{547.1 \text{ kJ}}$$

From energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out}} = \mathbf{547.1 \text{ kJ}}$$

5-73 Argon in a piston-cylinder device undergoes an isothermal process. The mass of argon and the work done are to be determined.

Assumptions 1 Argon is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 151 K and 4.86 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The gas constant of argon is $R = 0.2081 \text{ kJ/kg} \cdot \text{K}$ (Table A-1).

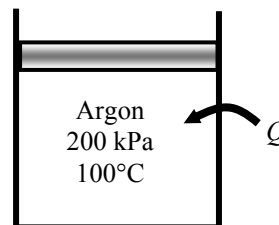
Analysis We take argon as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

$$Q_{\text{in}} - W_{b,\text{out}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{b,\text{out}}$$



Thus,

$$W_{b,\text{out}} = Q_{\text{in}} = \mathbf{1500 \text{ kJ}}$$

Using the boundary work relation for the isothermal process of an ideal gas gives

$$W_{b,\text{out}} = m \int_1^2 P d\upsilon = mRT \int_1^2 \frac{d\upsilon}{\upsilon} = mRT \ln \frac{\upsilon_2}{\upsilon_1} = mRT \ln \frac{P_1}{P_2}$$

Solving for the mass of the system,

$$m = \frac{W_{b,\text{out}}}{RT \ln \frac{P_1}{P_2}} = \frac{1500 \text{ kJ}}{(0.2081 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(373 \text{ K}) \ln \frac{200 \text{ kPa}}{50 \text{ kPa}}} = \mathbf{13.94 \text{ kg}}$$

5-74 Argon is compressed in a polytropic process. The work done and the heat transfer are to be determined.

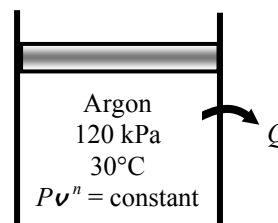
Assumptions 1 Argon is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 151 K and 4.86 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The properties of argon are $R = 0.2081 \text{ kJ/kg} \cdot \text{K}$ and $c_v = 0.3122 \text{ kJ/kg} \cdot \text{K}$ (Table A-2a).

Analysis We take argon as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$



Using the boundary work relation for the polytropic process of an ideal gas gives

$$w_{b,\text{out}} = \frac{RT_1}{1-n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] = \frac{(0.2081 \text{ kJ/kg} \cdot \text{K})(303 \text{ K})}{1-1.2} \left[\left(\frac{1200}{120} \right)^{0.2/1.2} - 1 \right] = -147.5 \text{ kJ/kg}$$

Thus,

$$w_{b,\text{in}} = \mathbf{147.5 \text{ kJ/kg}}$$

The temperature at the final state is

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(n-1)/n} = (303 \text{ K}) \left(\frac{1200 \text{ kPa}}{120 \text{ kPa}} \right)^{0.2/1.2} = 444.7 \text{ K}$$

From the energy balance equation,

$$q_{\text{in}} = w_{b,\text{out}} + c_v(T_2 - T_1) = -147.5 \text{ kJ/kg} + (0.3122 \text{ kJ/kg} \cdot \text{K})(444.7 - 303) \text{ K} = -103.3 \text{ kJ/kg}$$

Thus,

$$q_{\text{out}} = \mathbf{103.3 \text{ kJ/kg}}$$

5-75E Carbon dioxide contained in a piston-cylinder device undergoes a constant-pressure process. The work done and the heat transfer are to be determined.

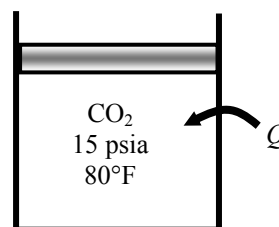
Assumptions 1 Carbon dioxide is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 547.5 R and 1071 psia. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats at room temperature can be used for CO_2 .

Properties The properties of CO_2 at room temperature are $R = 0.04513 \text{ Btu/lbm}\cdot\text{R}$ and $c_v = 0.158 \text{ Btu/lbm}\cdot\text{R}$ (Table A-2Ea).

Analysis We take CO_2 as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$



Using the boundary work relation for the isobaric process of an ideal gas gives

$$w_{b,\text{out}} = P(v_2 - v_1) = R(T_2 - T_1) = (0.04513 \text{ Btu/lbm}\cdot\text{R})(200 - 80)\text{R} = \mathbf{5.416 \text{ Btu/lbm}}$$

Substituting into energy balance equation,

$$q_{\text{in}} = w_{b,\text{out}} + c_v(T_2 - T_1) = 5.416 \text{ Btu/lbm} + (0.158 \text{ Btu/lbm}\cdot\text{R})(200 - 80)\text{R} = \mathbf{24.38 \text{ Btu/lbm}}$$

5-76 Helium contained in a spring-loaded piston-cylinder device is heated. The work done and the heat transfer are to be determined.

Assumptions 1 Helium is an ideal gas since it is at a high temperature relative to its critical temperature of 5.3 K. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The properties of helium are $R = 2.0769 \text{ kJ/kg}\cdot\text{K}$ and $c_v = 3.1156 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis We take helium as the system. This is a *closed system* since no mass crosses the boundaries of the system.

The energy balance for this system can be expressed as

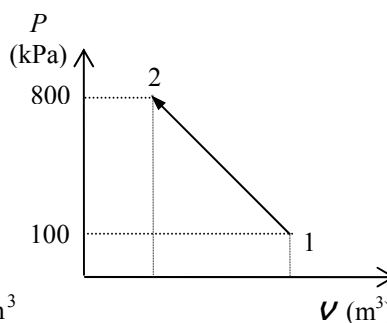
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

The initial and final specific volumes are

$$v_1 = \frac{mRT_1}{P_1} = \frac{(5 \text{ kg})(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293 \text{ K})}{100 \text{ kPa}} = 30.427 \text{ m}^3$$

$$v_2 = \frac{mRT_2}{P_2} = \frac{(5 \text{ kg})(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(160 + 273 \text{ K})}{800 \text{ kPa}} = 5.621 \text{ m}^3$$



Pressure changes linearly with volume and the work done is equal to the area under the process line 1-2:

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2} (v_2 - v_1) \\ &= \frac{(100 + 800) \text{ kPa}}{2} (5.621 - 30.427) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3} \right) \\ &= -11,163 \text{ kJ} \end{aligned}$$

Thus,

$$W_{b,\text{in}} = \mathbf{11.163 \text{ kJ}}$$

Using the energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out}} + mc_v(T_2 - T_1) = -11,163 \text{ kJ} + (5 \text{ kg})(3.1156 \text{ kJ/kg}\cdot\text{K})(160 - 20) \text{ K} = -8982 \text{ kJ}$$

Thus,

$$Q_{\text{out}} = \mathbf{8982 \text{ kJ}}$$

5-77 A piston-cylinder device contains air. A paddle wheel supplies a given amount of work to the air. The heat transfer is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 132.5 K and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for air.

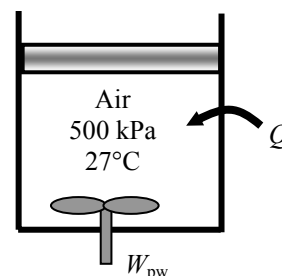
Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{pw,in}} - W_{\text{b,out}} + Q_{\text{in}} = \Delta U = mc_v(T_2 - T_1)$$

$$W_{\text{pw,in}} - W_{\text{b,out}} + Q_{\text{in}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{\text{b,out}} - W_{\text{pw,in}}$$



Using the boundary work relation on a unit mass basis for the isothermal process of an ideal gas gives

$$w_{\text{b,out}} = RT \ln \frac{v_2}{v_1} = RT \ln 3 = (0.287 \text{ kJ/kg} \cdot \text{K})(300 \text{ K}) \ln 3 = 94.6 \text{ kJ/kg}$$

Substituting into the energy balance equation (expressed on a unit mass basis) gives

$$q_{\text{in}} = w_{\text{b,out}} - w_{\text{pw,in}} = 94.6 - 50 = \mathbf{44.6 \text{ kJ/kg}}$$

Discussion Note that the energy content of the system remains constant in this case, and thus the total energy transfer output via boundary work must equal the total energy input via shaft work and heat transfer.

5-78 A cylinder equipped with a set of stops for the piston to rest on is initially filled with helium gas at a specified state. The amount of heat that must be transferred to raise the piston is to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** There are no work interactions involved. **4** The thermal energy stored in the cylinder itself is negligible.

Properties The specific heat of helium at room temperature is $c_v = 3.1156 \text{ kJ/kg}\cdot\text{K}$ (Table A-2).

Analysis We take the helium gas in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this constant volume closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{in} = \Delta U = m(u_2 - u_1)$$

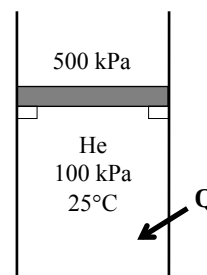
$$Q_{in} = m(u_2 - u_1) = mc_v(T_2 - T_1)$$

The final temperature of helium can be determined from the ideal gas relation to be

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{500 \text{ kPa}}{100 \text{ kPa}} (298 \text{ K}) = 1490 \text{ K}$$

Substituting into the energy balance relation gives

$$Q_{in} = (0.5 \text{ kg})(3.1156 \text{ kJ/kg}\cdot\text{K})(1490 - 298)\text{K} = \mathbf{1857 \text{ kJ}}$$



5-79 A cylinder is initially filled with air at a specified state. Air is heated electrically at constant pressure, and some heat is lost in the process. The amount of electrical energy supplied is to be determined.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** Air is an ideal gas with variable specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The initial and final enthalpies of air are (Table A-21)

$$h_1 = h_{@298\text{ K}} = 298.18 \text{ kJ/kg}$$

$$h_2 = h_{@350\text{ K}} = 350.49 \text{ kJ/kg}$$

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,in} - Q_{out} - W_{b,out} = \Delta U \longrightarrow W_{e,in} = m(h_2 - h_1) + Q_{out}$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting,

$$W_{e,in} = (15 \text{ kg})(350.49 - 298.18) \text{ kJ/kg} + (60 \text{ kJ}) = 845 \text{ kJ}$$

or,

$$W_{e,in} = (845 \text{ kJ}) \left(\frac{1 \text{ kWh}}{3600 \text{ kJ}} \right) = \mathbf{0.235 \text{ kWh}}$$

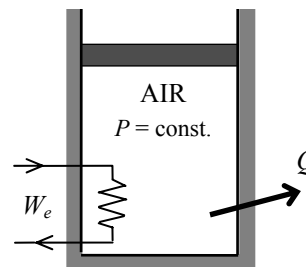
Alternative solution The specific heat of air at the average temperature of $T_{\text{avg}} = (25 + 77)/2 = 51^\circ\text{C} = 324 \text{ K}$ is, from Table A-2b, $c_{p,\text{avg}} = 1.0065 \text{ kJ/kg}\cdot^\circ\text{C}$. Substituting,

$$W_{e,in} = mc_p(T_2 - T_1) + Q_{out} = (15 \text{ kg})(1.0065 \text{ kJ/kg}\cdot^\circ\text{C})(77 - 25)^\circ\text{C} + 60 \text{ kJ} = 845 \text{ kJ}$$

or,

$$W_{e,in} = (845 \text{ kJ}) \left(\frac{1 \text{ kWh}}{3600 \text{ kJ}} \right) = \mathbf{0.235 \text{ kWh}}$$

Discussion Note that for small temperature differences, both approaches give the same result.



5-80 An insulated cylinder initially contains CO₂ at a specified state. The CO₂ is heated electrically for 10 min at constant pressure until the volume doubles. The electric current is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The CO₂ is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The gas constant and molar mass of CO₂ are $R = 0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ and $M = 44 \text{ kg/kmol}$ (Table A-1). The specific heat of CO₂ at the average temperature of $T_{\text{avg}} = (300 + 600)/2 = 450 \text{ K}$ is $c_{p,\text{avg}} = 0.978 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2b).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1) \cong mc_p(T_2 - T_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The final temperature of CO₂ is

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2 V_2}{P_1 V_1} T_1 = 1 \times 2 \times (300 \text{ K}) = 600 \text{ K}$$

The mass of CO₂ is

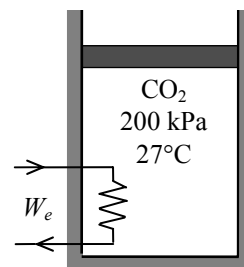
$$m = \frac{P_1 V_1}{RT_1} = \frac{(200 \text{ kPa})(0.3 \text{ m}^3)}{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 1.059 \text{ kg}$$

Substituting,

$$W_{\text{e,in}} = (1.059 \text{ kg})(0.978 \text{ kJ/kg} \cdot \text{K})(600 - 300)\text{K} = 311 \text{ kJ}$$

Then,

$$I = \frac{W_{\text{e,in}}}{V \Delta t} = \frac{311 \text{ kJ}}{(110 \text{ V})(10 \times 60 \text{ s})} \left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}} \right) = \mathbf{4.71 \text{ A}}$$



5-81 A cylinder initially contains nitrogen gas at a specified state. The gas is compressed polytropically until the volume is reduced by one-half. The work done and the heat transfer are to be determined.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are zero. 2 The N_2 is an ideal gas with constant specific heats. 3 The thermal energy stored in the cylinder itself is negligible. 4 The compression or expansion process is quasi-equilibrium.

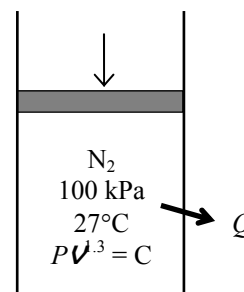
Properties The gas constant of N_2 is $R = 0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The c_v value of N_2 at the average temperature $(369+300)/2 = 335 \text{ K}$ is $0.744 \text{ kJ/kg}\cdot\text{K}$ (Table A-2b).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$W_{\text{b,in}} - Q_{\text{out}} = mc_v(T_2 - T_1)$$



The final pressure and temperature of nitrogen are

$$P_2 v_2^{1.3} = P_1 v_1^{1.3} \longrightarrow P_2 = \left(\frac{v_1}{v_2} \right)^{1.3} P_1 = 2^{1.3} (100 \text{ kPa}) = 246.2 \text{ kPa}$$

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2} \longrightarrow T_2 = \frac{P_2 v_2}{P_1 v_1} T_1 = \frac{246.2 \text{ kPa}}{100 \text{ kPa}} \times 0.5 \times (300 \text{ K}) = 369.3 \text{ K}$$

Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P dv = -\frac{P_2 v_2 - P_1 v_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.8 \text{ kg})(0.2968 \text{ kJ/kg}\cdot\text{K})(369.3 - 300)\text{K}}{1 - 1.3} = \mathbf{54.8 \text{ kJ}}$$

Substituting into the energy balance gives

$$Q_{\text{out}} = W_{\text{b,in}} - mc_v(T_2 - T_1)$$

$$= 54.8 \text{ kJ} - (0.8 \text{ kg})(0.744 \text{ kJ/kg}\cdot\text{K})(369.3 - 300)\text{K}$$

$$= \mathbf{13.6 \text{ kJ}}$$

5-82 EES Problem 5-81 is reconsidered. The process is to be plotted on a P - V diagram, and the effect of the polytropic exponent n on the boundary work and heat transfer as the polytropic exponent varies from 1.1 to 1.6 is to be investigated. The boundary work and the heat transfer are to be plotted versus the polytropic exponent.

Analysis The problem is solved using EES, and the solution is given below.

```
Procedure Work(P[2],V[2],P[1],V[1],n:W12)
```

```
If n=1 then
```

```
W12=P[1]*V[1]*ln(V[2]/V[1])
```

```
Else
```

```
W12=(P[2]*V[2]-P[1]*V[1])/(1-n)
```

```
endif
```

```
End
```

"Input Data"

Vratio=0.5 "V[2]/V[1] = Vratio"

n=1.3 "Polytropic exponent"

P[1] = 100 [kPa]

T[1] = (27+273) [K]

m=0.8 [kg]

MM=molarmass(nitrogen)

R_u=8.314 [kJ/kmol-K]

R=R_u/MM

V[1]=m*R*T[1]/P[1]

"Process equations"

V[2]=Vratio*V[1]

P[2]*V[2]/T[2]=P[1]*V[1]/T[1]"The combined ideal gas law for states 1 and 2 plus the polytropic process relation give P[2] and T[2]"

P[2]*V[2]^n=P[1]*V[1]^n

"Conservation of Energy for the closed system:"

"E_in - E_out = DeltaE, we neglect Delta KE and Delta PE for the system, the nitrogen."

Q12 - W12 = m*(u[2]-u[1])

u[1]=intenergy(N2, T=T[1]) "internal energy for nitrogen as an ideal gas, kJ/kg"

u[2]=intenergy(N2, T=T[2])

Call Work(P[2],V[2],P[1],V[1],n:W12)

"The following is required for the P-v plots"

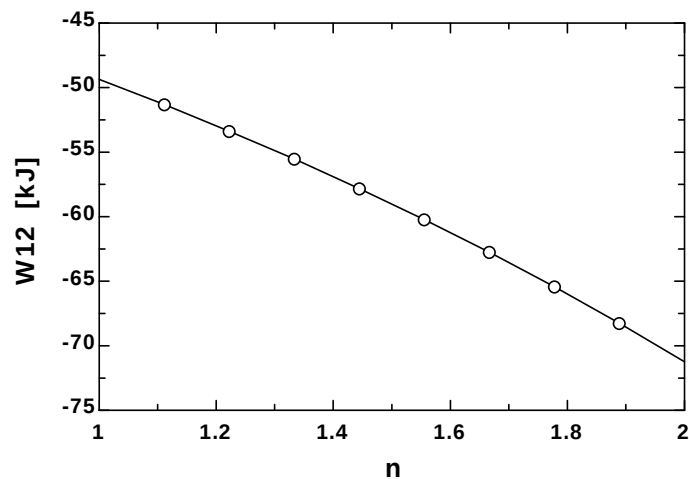
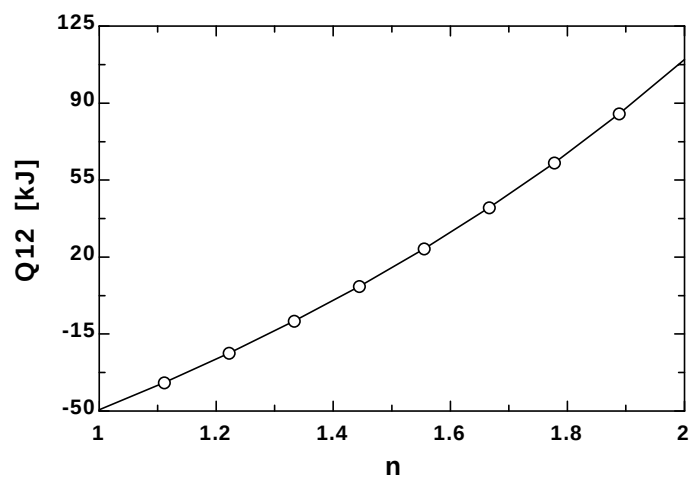
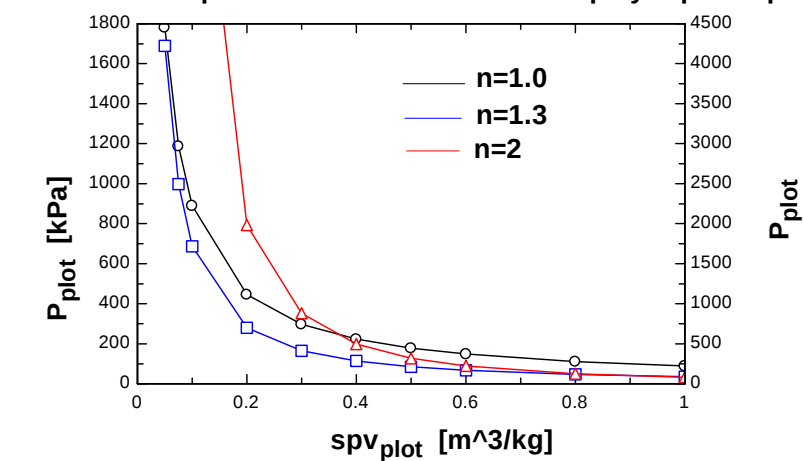
{P_plot*spv_plot/T_plot=P[1]*V[1]/m/T[1]"The combined ideal gas law for states 1 and 2 plus the polytropic process relation give P[2] and T[2]"

P_plot*spv_plot^n=P[1]*(V[1]/m)^n}

{spV_plot=R*T_plot/P_plot"[m^3]"}

n	Q12 [kJ]	W12 [kJ]
1	-49.37	-49.37
1.111	-37	-51.32
1.222	-23.59	-53.38
1.333	-9.067	-55.54
1.444	6.685	-57.82
1.556	23.81	-60.23
1.667	42.48	-62.76
1.778	62.89	-65.43
1.889	85.27	-68.25
2	109.9	-71.23

Pressure vs. specific volume as function of polytropic exponent



5-83 It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 6500 kJ/h. The power rating of the heater is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The temperature of the room is said to remain constant during this process.

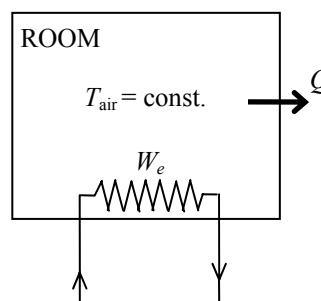
Analysis We take the room as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this system reduces to

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = 0 \longrightarrow W_{\text{e,in}} = Q_{\text{out}}$$

since $\Delta U = mc\Delta T = 0$ for isothermal processes of ideal gases. Thus,

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} = (6500 \text{ kJ/h}) \left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}} \right) = \mathbf{1.81 \text{ kW}}$$



5-84 A cylinder equipped with a set of stops for the piston is initially filled with air at a specified state. Heat is transferred to the air until the volume doubled. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a P - v diagram.

Assumptions 1 Air is an ideal gas with variable specific heats. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 The thermal energy stored in the cylinder itself is negligible. 4 The compression or expansion process is quasi-equilibrium.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis We take the air in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_3 - u_1)$$

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}}$$

The initial and the final volumes and the final temperature of air are

$$v_1 = \frac{mRT_1}{P_1} = \frac{(3 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})}{200 \text{ kPa}} = 1.29 \text{ m}^3$$

$$v_3 = 2v_1 = 2 \times 1.29 = 2.58 \text{ m}^3$$

$$\frac{P_1 v_1}{T_1} = \frac{P_3 v_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1} \frac{v_3}{v_1} T_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 2 \times (300 \text{ K}) = 1200 \text{ K}$$

No work is done during process 1-2 since $v_1 = v_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_{\text{b,out}} = \int_1^2 P dv = P_2(v_3 - v_2) = (400 \text{ kPa})(2.58 - 1.29) \text{ m}^3 = 516 \text{ kJ}$$

The initial and final internal energies of air are (Table A-21)

$$u_1 = u_{@300 \text{ K}} = 214.07 \text{ kJ/kg}$$

$$u_3 = u_{@1200 \text{ K}} = 933.33 \text{ kJ/kg}$$

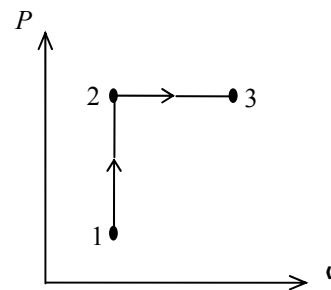
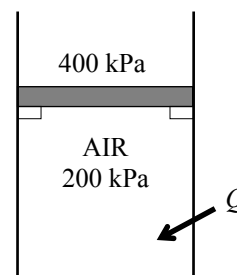
Then from the energy balance,

$$Q_{\text{in}} = (3 \text{ kg})(933.33 - 214.07) \text{ kJ/kg} + 516 \text{ kJ} = 2674 \text{ kJ}$$

Alternative solution The specific heat of air at the average temperature of $T_{\text{avg}} = (300 + 1200)/2 = 750 \text{ K}$ is, from Table A-2b, $c_{v,\text{avg}} = 0.800 \text{ kJ/kg} \cdot \text{K}$. Substituting,

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}} \cong mc_v(T_3 - T_1) + W_{\text{b,out}}$$

$$Q_{\text{in}} = (3 \text{ kg})(0.800 \text{ kJ/kg} \cdot \text{K})(1200 - 300) \text{ K} + 516 \text{ kJ} = 2676 \text{ kJ}$$



5-85 CD EES A cylinder equipped with a set of stops on the top is initially filled with air at a specified state. Heat is transferred to the air until the piston hits the stops, and then the pressure doubles. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a P - v diagram.

Assumptions 1 Air is an ideal gas with variable specific heats. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 The thermal energy stored in the cylinder itself is negligible.

Properties The gas constant of air is $R = 0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).

Analysis We take the air in the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_3 - u_1)$$

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}}$$

The initial and the final volumes and the final temperature of air are determined from

$$v_1 = \frac{mRT_1}{P_1} = \frac{(3 \text{ kg})(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{200 \text{ kPa}} = 1.29 \text{ m}^3$$

$$v_3 = 2v_1 = 2 \times 1.29 = 2.58 \text{ m}^3$$

$$\frac{P_1 v_1}{T_1} = \frac{P_3 v_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1} \frac{v_3}{v_1} T_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 2 \times (300 \text{ K}) = 1200 \text{ K}$$

No work is done during process 2-3 since $v_2 = v_3$. The pressure remains constant during process 1-2 and the work done during this process is

$$W_b = \int_1^2 P dV = P_2 (v_2 - v_1)$$

$$= (200 \text{ kPa})(2.58 - 1.29) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3} \right) = 258 \text{ kJ}$$

The initial and final internal energies of air are (Table A-21)

$$u_1 = u_{@300 \text{ K}} = 214.07 \text{ kJ/kg}$$

$$u_3 = u_{@1200 \text{ K}} = 933.33 \text{ kJ/kg}$$

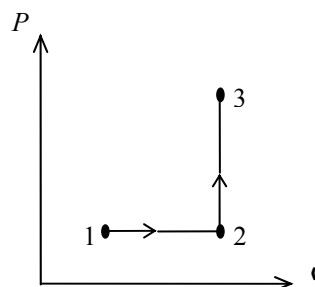
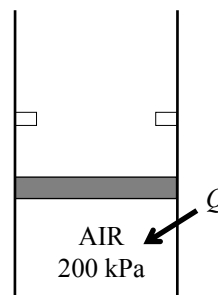
Substituting,

$$Q_{\text{in}} = (3 \text{ kg})(933.33 - 214.07) \text{ kJ/kg} + 258 \text{ kJ} = \mathbf{2416 \text{ kJ}}$$

Alternative solution The specific heat of air at the average temperature of $T_{\text{avg}} = (300 + 1200)/2 = 750 \text{ K}$ is, from Table A-2b, $c_{v,\text{avg}} = 0.800 \text{ kJ/kg}\cdot\text{K}$. Substituting

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}} \cong mc_v(T_3 - T_1) + W_{\text{b,out}}$$

$$= (3 \text{ kg})(0.800 \text{ kJ/kg}\cdot\text{K})(1200 - 300) \text{ K} + 258 \text{ kJ} = \mathbf{2418 \text{ kJ}}$$



Closed System Energy Analysis: Solids and Liquids

5-86 An iron block is heated. The internal energy and enthalpy changes are to be determined for a given temperature change.

Assumptions Iron is an incompressible substance with a constant specific heat.

Properties The specific heat of iron is 0.45 kJ/kg·K (Table A-3b).

Analysis The internal energy and enthalpy changes are equal for a solid. Then,

$$\Delta H = \Delta U = mc\Delta T = (1 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K})(80 - 20)\text{K} = \mathbf{27 \text{ kJ}}$$

5-87E Liquid water experiences a process from one state to another. The internal energy and enthalpy changes are to be determined under different assumptions.

Analysis (a) Using the properties from compressed liquid tables

$$u_1 \cong u_{f@50^\circ\text{F}} = 18.07 \text{ Btu/lbm} \quad (\text{Table A - 4E})$$

$$\begin{aligned} h_1 &= h_{f@50^\circ\text{F}} + v_f(P - P_{\text{sat}@T}) \\ &= 18.07 \text{ Btu/lbm} + (0.01602 \text{ ft}^3/\text{lbm})(50 - 0.17812) \text{ psia} = 18.87 \text{ Btu/lbm} \end{aligned}$$

$$\left. \begin{array}{l} P_2 = 2000 \text{ psia} \\ T_2 = 100^\circ\text{F} \end{array} \right\} \begin{array}{l} u_2 = 67.36 \text{ Btu/lbm} \\ h_2 = 73.30 \text{ Btu/lbm} \end{array} \quad (\text{Table A - 7E})$$

$$\Delta u = u_2 - u_1 = 67.36 - 18.07 = \mathbf{49.29 \text{ Btu/lbm}}$$

$$\Delta h = h_2 - h_1 = 73.30 - 18.87 = \mathbf{54.43 \text{ Btu/lbm}}$$

(b) Using incompressible substance approximation and property tables (Table A-4E),

$$u_1 \cong u_{f@50^\circ\text{F}} = 18.07 \text{ Btu/lbm}$$

$$h_1 \cong h_{f@50^\circ\text{F}} = 18.07 \text{ Btu/lbm}$$

$$u_2 \cong u_{f@100^\circ\text{F}} = 68.03 \text{ Btu/lbm}$$

$$h_2 \cong h_{f@100^\circ\text{F}} = 68.03 \text{ Btu/lbm}$$

$$\Delta u = u_2 - u_1 = 68.03 - 18.07 = \mathbf{49.96 \text{ Btu/lbm}}$$

$$\Delta h = h_2 - h_1 = 68.03 - 18.07 = \mathbf{49.96 \text{ Btu/lbm}}$$

(c) Using specific heats and taking the specific heat of water to be 1.00 Btu/lbm·R (Table A-3Ea),

$$\Delta h = \Delta u = c\Delta T = (1.00 \text{ Btu/lbm} \cdot \text{R})(100 - 50)\text{R} = \mathbf{50 \text{ Btu/lbm}}$$

5-88E A person shakes a canned drink in a iced water to cool it. The mass of the ice that will melt by the time the canned drink is cooled to a specified temperature is to be determined.

Assumptions **1** The thermal properties of the drink are constant, and are taken to be the same as those of water. **2** The effect of agitation on the amount of ice melting is negligible. **3** The thermal energy capacity of the can itself is negligible, and thus it does not need to be considered in the analysis.

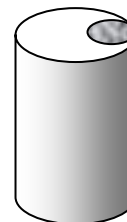
Properties The density and specific heat of water at the average temperature of $(75+45)/2 = 60^\circ\text{F}$ are $\rho = 62.3 \text{ lbm/ft}^3$, and $c_p = 1.0 \text{ Btu/lbm}\cdot^\circ\text{F}$ (Table A-3E). The heat of fusion of water is 143.5 Btu/lbm .

Analysis We take a canned drink as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{\text{out}} = \Delta U_{\text{canned drink}} = m(u_2 - u_1) \longrightarrow Q_{\text{out}} = mc(T_1 - T_2)$$

Cola
75°F



Noting that $1 \text{ gal} = 128 \text{ oz}$ and $1 \text{ ft}^3 = 7.48 \text{ gal} = 957.5 \text{ oz}$, the total amount of heat transfer from a ball is

$$m = \rho V = (62.3 \text{ lbm/ft}^3)(12 \text{ oz/can}) \left(\frac{1 \text{ ft}^3}{7.48 \text{ gal}} \right) \left(\frac{1 \text{ gal}}{128 \text{ fluid oz}} \right) = 0.781 \text{ lbm/can}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.781 \text{ lbm/can})(1.0 \text{ Btu/lbm}\cdot^\circ\text{F})(75 - 45)^\circ\text{F} = 23.4 \text{ Btu/can}$$

Noting that the heat of fusion of water is 143.5 Btu/lbm , the amount of ice that will melt to cool the drink is

$$m_{\text{ice}} = \frac{Q_{\text{out}}}{h_{if}} = \frac{23.4 \text{ Btu/can}}{143.5 \text{ Btu/lbm}} = \mathbf{0.163 \text{ lbm}} \quad (\text{per can of drink})$$

since heat transfer to the ice must be equal to heat transfer from the can.

Discussion The actual amount of ice melted will be greater since agitation will also cause some ice to melt.

5-89 An iron whose base plate is made of an aluminum alloy is turned on. The minimum time for the plate to reach a specified temperature is to be determined.

Assumptions **1** It is given that 85 percent of the heat generated in the resistance wires is transferred to the plate. **2** The thermal properties of the plate are constant. **3** Heat loss from the plate during heating is disregarded since the minimum heating time is to be determined. **4** There are no changes in kinetic and potential energies. **5** The plate is at a uniform temperature at the end of the process.

Properties The density and specific heat of the aluminum alloy plate are given to be $\rho = 2770 \text{ kg/m}^3$ and $c_p = 875 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The mass of the iron's base plate is

$$m = \rho V = \rho LA = (2770 \text{ kg/m}^3)(0.005 \text{ m})(0.03 \text{ m}^2) = 0.4155 \text{ kg}$$

Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\text{in}} = 0.85 \times 1000 \text{ W} = 850 \text{ W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as

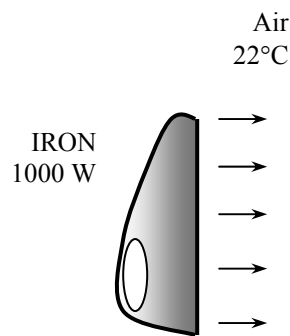
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) \longrightarrow \dot{Q}_{\text{in}} \Delta t = mc(T_2 - T_1)$$

Solving for Δt and substituting,

$$\Delta t = \frac{mc\Delta T_{\text{plate}}}{\dot{Q}_{\text{in}}} = \frac{(0.4155 \text{ kg})(875 \text{ J/kg} \cdot ^\circ\text{C})(140 - 22)^\circ\text{C}}{850 \text{ J/s}} = 50.5 \text{ s}$$

which is the time required for the plate temperature to reach the specified temperature.



5-90 Stainless steel ball bearings leaving the oven at a specified uniform temperature at a specified rate are exposed to air and are cooled before they are dropped into the water for quenching. The rate of heat transfer from the ball bearing to the air is to be determined.

Assumptions 1 The thermal properties of the bearing balls are constant. 2 The kinetic and potential energy changes of the balls are negligible. 3 The balls are at a uniform temperature at the end of the process

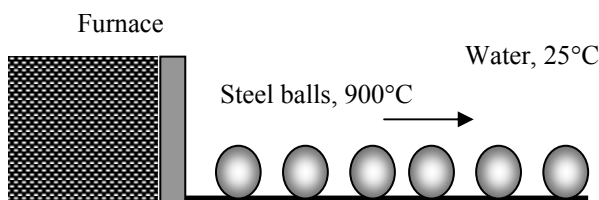
Properties The density and specific heat of the ball bearings are given to be $\rho = 8085 \text{ kg/m}^3$ and $c_p = 0.480 \text{ kJ/kg}\cdot^\circ\text{C}$.

Analysis We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8085 \text{ kg/m}^3) \frac{\pi (0.012 \text{ m})^3}{6} = 0.007315 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.007315 \text{ kg})(0.480 \text{ kJ/kg}\cdot^\circ\text{C})(900 - 850)^\circ\text{C} = 0.1756 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{out (per ball)}} = (800 \text{ balls/min}) \times (0.1756 \text{ kJ/ball}) = \mathbf{140.5 \text{ kJ/min} = 2.34 \text{ kW}}$$

Therefore, heat is lost to the air at a rate of 2.34 kW.

5-91 Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air is to be determined.

Assumptions **1** The thermal properties of the balls are constant. **2** There are no changes in kinetic and potential energies. **3** The balls are at a uniform temperature at the end of the process

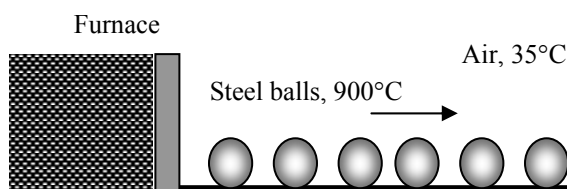
Properties The density and specific heat of the balls are given to be $\rho = 7833 \text{ kg/m}^3$ and $c_p = 0.465 \text{ kJ/kg}\cdot^\circ\text{C}$.

Analysis We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$



(b) The amount of heat transfer from a single ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (7833 \text{ kg/m}^3) \frac{\pi (0.008 \text{ m})^3}{6} = 0.00210 \text{ kg}$$

$$Q_{\text{out}} = mc_p(T_1 - T_2) = (0.0021 \text{ kg})(0.465 \text{ kJ/kg}\cdot^\circ\text{C})(900 - 100)^\circ\text{C} = 0.781 \text{ kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

$$\dot{Q}_{\text{out}} = \dot{n}_{\text{ball}} Q_{\text{out}} = (2500 \text{ balls/h}) \times (0.781 \text{ kJ/ball}) = 1,953 \text{ kJ/h} = \mathbf{542 \text{ W}}$$

5-92 An electronic device is on for 5 minutes, and off for several hours. The temperature of the device at the end of the 5-min operating period is to be determined for the cases of operation with and without a heat sink.

Assumptions 1 The device and the heat sink are isothermal. 2 The thermal properties of the device and of the sink are constant. 3 Heat loss from the device during on time is disregarded since the highest possible temperature is to be determined.

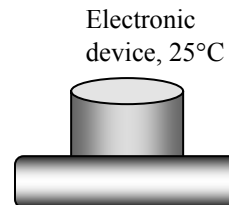
Properties The specific heat of the device is given to be $c_p = 850 \text{ J/kg} \cdot ^\circ\text{C}$. The specific heat of aluminum at room temperature of 300 K is $902 \text{ J/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis We take the device to be the system. Noting that electrical energy is supplied, the energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{e,in}} = \Delta U_{\text{device}} = m(u_2 - u_1)$$

$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)$$



Substituting, the temperature of the device at the end of the process is determined to be

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg} \cdot ^\circ\text{C})(T_2 - 25)^\circ\text{C} \rightarrow T_2 = \mathbf{554^\circ\text{C}} \text{ (without the heat sink)}$$

Case 2 When a heat sink is attached, the energy balance can be expressed as

$$W_{\text{e,in}} = \Delta U_{\text{device}} + \Delta U_{\text{heat sink}}$$

$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)_{\text{device}} + mc(T_2 - T_1)_{\text{heat sink}}$$

Substituting, the temperature of the device-heat sink combination is determined to be

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg} \cdot ^\circ\text{C})(T_2 - 25)^\circ\text{C} + (0.200 \text{ kg})(902 \text{ J/kg} \cdot ^\circ\text{C})(T_2 - 25)^\circ\text{C}$$

$$T_2 = \mathbf{70.6^\circ\text{C}} \text{ (with heat sink)}$$

Discussion These are the maximum temperatures. In reality, the temperatures will be lower because of the heat losses to the surroundings.

5-93 EES Problem 5-92 is reconsidered. The effect of the mass of the heat sink on the maximum device temperature as the mass of heat sink varies from 0 kg to 1 kg is to be investigated. The maximum temperature is to be plotted against the mass of heat sink.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns:"

"T_1 is the maximum temperature of the device"

Q_dot_out = 30 [W]

m_device=20 [g]

Cp_device=850 [J/kg-C]

A=5 [cm^2]

DELTA_t=5 [min]

T_amb=25 [C]

{m_sink=0.2 [kg]}

"Cp_al taken from Table A-3(b) at 300K"

Cp_al=0.902 [kJ/kg-C]

T_2=T_amb

"Solution:"

"The device without the heat sink is considered to be a closed system."

"Conservation of Energy for the closed system:"

"E_dot_in - E_dot_out = DELTAE_dot, we neglect DELTA KE and DELTA PE for the system, the device."

E_dot_in - E_dot_out = DELTAE_dot

E_dot_in = 0

E_dot_out = Q_dot_out

"Use the solid material approximation to find the energy change of the device."

DELTA E_dot = m_device*convert(g,kg)*Cp_device*(T_2-T_1_device)/(DELTA_t*convert(min,s))

"The device with the heat sink is considered to be a closed system."

"Conservation of Energy for the closed system:"

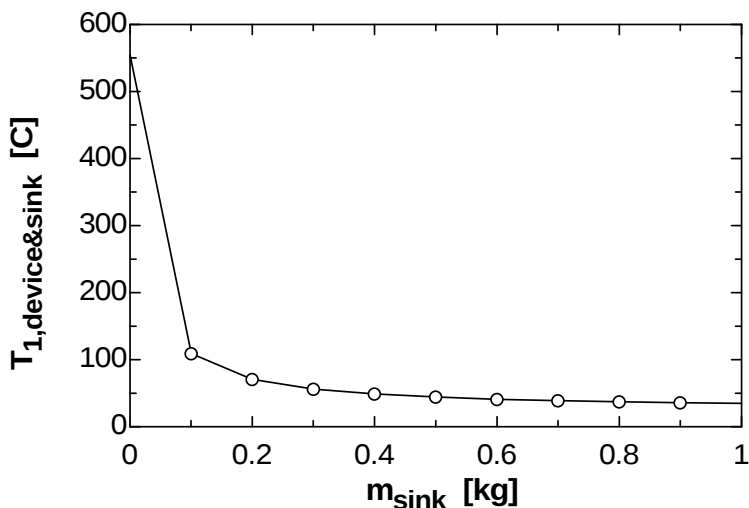
"E_dot_in - E_dot_out = DELTAE_dot, we neglect DELTA KE and DELTA PE for the device with the heat sink."

E_dot_in - E_dot_out = DELTAE_dot_combined

"Use the solid material approximation to find the energy change of the device."

DELTA E_dot_combined = (m_device*convert(g,kg)*Cp_device*(T_2-T_1_device&sink)+m_sink*Cp_al*(T_2-T_1_device&sink)*convert(kJ,J))/(DELTA_t*convert(min,s))

m _{sink} [kg]	T _{1,device&sink} [C]
0	554.4
0.1	109
0.2	70.59
0.3	56.29
0.4	48.82
0.5	44.23
0.6	41.12
0.7	38.88
0.8	37.19
0.9	35.86
1	34.79



5-94 An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked is to be determined.

Assumptions **1** The egg is spherical in shape with a radius of $r_0 = 2.75$ cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

Properties The density and specific heat of the egg are given to be $\rho = 1020$ kg/m³ and $c_p = 3.32$ kJ/kg·°C.

Analysis We take the egg as the system. This is a closed system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

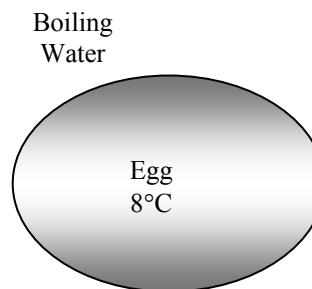
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{egg}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020 \text{ kg/m}^3) \frac{\pi (0.055 \text{ m})^3}{6} = 0.0889 \text{ kg}$$

$$Q_{\text{in}} = mc_p(T_2 - T_1) = (0.0889 \text{ kg})(3.32 \text{ kJ/kg} \cdot ^\circ\text{C})(80 - 8)^\circ\text{C} = \mathbf{21.2 \text{ kJ}}$$



5-95E Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven is to be determined.

Assumptions **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the brass are given to be $\rho = 532.5$ lbm/ft³ and $c_p = 0.091$ Btu/lbm·°F.

Analysis We take the plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

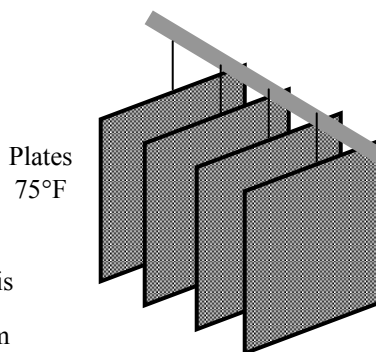
The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho V = \rho LA = (532.5 \text{ lbm/ft}^3) [(1.2 / 12 \text{ ft})(2 \text{ ft})(2 \text{ ft})] = 213 \text{ lbm}$$

$$Q_{\text{in}} = mc(T_2 - T_1) = (213 \text{ lbm/plate})(0.091 \text{ Btu/lbm} \cdot ^\circ\text{F})(1000 - 75)^\circ\text{F} = 17,930 \text{ Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{plate}} Q_{\text{in, per plate}} = (300 \text{ plates/min}) \times (17,930 \text{ Btu/plate}) = \mathbf{5,379,000 \text{ Btu/min} = 89,650 \text{ Btu/s}}$$



5-96 Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven is to be determined.

Assumptions 1 The thermal properties of the rods are constant. 2 The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the steel rods are given to be $\rho = 7833 \text{ kg/m}^3$ and $c_p = 0.465 \text{ kJ/kg} \cdot ^\circ\text{C}$.

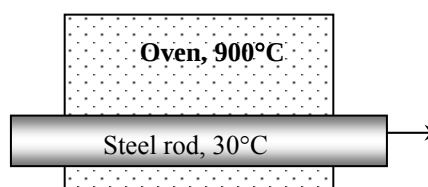
Analysis Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho V = \rho LA = \rho L(\pi D^2 / 4) = (7833 \text{ kg/m}^3)(3 \text{ m})[\pi(0.1 \text{ m})^2 / 4] = 184.6 \text{ kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6 \text{ kg})(0.465 \text{ kJ/kg} \cdot ^\circ\text{C})(700 - 30)^\circ\text{C} = 57,512 \text{ kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}} / \Delta t = (57,512 \text{ kJ}) / (1 \text{ min}) = 57,512 \text{ kJ/min} = \mathbf{958.5 \text{ kW}}$$

Review Problems

5-97 The compression work from P_1 to P_2 using a polytropic process is to be compared for neon and air.

Assumptions The process is quasi-equilibrium.

Properties The gas constants for neon and air $R = 0.4119$ and 0.287 kJ/kg·K, respectively (Table A-2a).

Analysis For a polytropic process,

$$P\nu^n = \text{Constant}$$

The boundary work during a polytropic process of an ideal gas is

$$w_{b,\text{out}} = \int_1^2 P d\nu = \text{Constant} \int_1^2 \nu^{-n} d\nu = \frac{P_1 \nu_1}{1-n} \left[\left(\frac{\nu_2}{\nu_1} \right)^{1-n} - 1 \right] = \frac{RT_1}{1-n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]$$

The negative of this expression gives the compression work during a polytropic process. Inspection of this equation reveals that the gas with the smallest gas constant (i.e., largest molecular weight) requires the least work for compression. In this problem, air will require the least amount of work.

5-98 Nitrogen is heated to experience a specified temperature change. The heat transfer is to be determined for two cases.

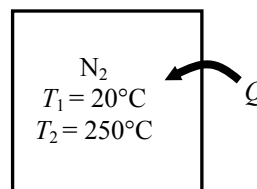
Assumptions 1 Nitrogen is an ideal gas since it is at a high temperature and probably low pressure relative to its critical point values of 126.2 K and 3.39 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for nitrogen.

Properties The specific heats of nitrogen at the average temperature of $(20+250)/2=135^\circ\text{C}=408\text{ K}$ are $c_p = 1.045\text{ kJ/kg}\cdot\text{K}$ and $c_v = 0.748\text{ kJ/kg}\cdot\text{K}$ (Table A-2b).

Analysis We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for a constant-volume process can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = \Delta U = mc_v(T_2 - T_1)$$



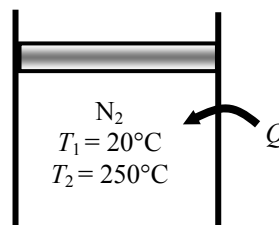
The energy balance during a constant-pressure process (such as in a piston-cylinder device) can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U$$

$$Q_{\text{in}} = W_{b,\text{out}} + \Delta U$$

$$Q_{\text{in}} = \Delta H = mc_p(T_2 - T_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting for both cases,

$$Q_{\text{in}, v=\text{const}} = mc_v(T_2 - T_1) = (10\text{ kg})(0.748\text{ kJ/kg}\cdot\text{K})(250 - 20)\text{K} = \mathbf{1720\text{ kJ}}$$

$$Q_{\text{in}, p=\text{const}} = mc_p(T_2 - T_1) = (10\text{ kg})(1.045\text{ kJ/kg}\cdot\text{K})(250 - 20)\text{K} = \mathbf{2404\text{ kJ}}$$

5-99E An insulated rigid vessel contains air. A paddle wheel supplies work to the air. The work supplied and final temperature are to be determined.

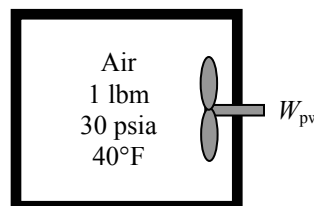
Assumptions 1 Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 238.5 R and 547 psia. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats can be used for air.

Properties The specific heats of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $c_v = 0.171$ Btu/lbm·R (Table A-2Ea).

Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{pw,in}} = \Delta U = mc_v(T_2 - T_1)$$



As the specific volume remains constant during this process, the ideal gas equation gives

$$T_2 = T_1 \frac{P_2}{P_1} = (500 \text{ R}) \frac{50 \text{ psia}}{30 \text{ psia}} = 833.3 \text{ R} = \mathbf{373.3^\circ\text{F}}$$

Substituting,

$$W_{\text{pw,in}} = mc_v(T_2 - T_1) = (1 \text{ lbm})(0.171 \text{ Btu/lbm} \cdot \text{R})(833.3 - 500) \text{ R} = \mathbf{57.0 \text{ Btu}}$$

5-100 During a phase change, a constant pressure process becomes a constant temperature process. Hence,

$$c_p = \left. \frac{\partial h}{\partial T} \right|_T = \frac{\text{finite}}{0} = \text{infinite}$$

and the specific heat at constant pressure has no meaning. The specific heat at constant volume does have a meaning since

$$c_v = \left. \frac{\partial u}{\partial T} \right|_v = \frac{\text{finite}}{\text{finite}} = \text{finite}$$

5-101 A gas mixture contained in a rigid tank is cooled. The heat transfer is to be determined.

Assumptions **1** The gas mixture is an ideal gas. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used.

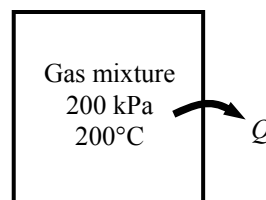
Properties The specific heat of gas mixture is given to be $c_v = 0.748 \text{ kJ/kg}\cdot\text{K}$.

Analysis We take the contents of the tank as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

$$q_{\text{out}} = c_v(T_1 - T_2)$$



As the specific volume remains constant during this process, the ideal gas equation gives

$$T_2 = T_1 \frac{P_2}{P_1} = (473 \text{ K}) \frac{100 \text{ kPa}}{200 \text{ kPa}} = 236.5 \text{ K}$$

Substituting,

$$q_{\text{out}} = c_v(T_1 - T_2) = (0.748 \text{ kJ/kg}\cdot\text{K})(473 - 236.5)\text{K} = \mathbf{177 \text{ kJ/kg}}$$

5-102 A well-insulated rigid vessel contains saturated liquid water. Electrical work is done on water. The final temperature is to be determined.

Assumptions 1 The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the tank itself is negligible.

Analysis We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1)$$

The amount of electrical work added during 30 minutes period is

$$W_{e,\text{in}} = VI\Delta t = (50 \text{ V})(10 \text{ A})(30 \times 60 \text{ s}) \left(\frac{1 \text{ W}}{1 \text{ VA}} \right) = 900,000 \text{ J} = 900 \text{ kJ}$$

The properties at the initial state are (Table A-4)

$$u_1 = u_{f@40^\circ\text{C}} = 167.53 \text{ kJ/kg}$$

$$\nu_1 = \nu_{f@40^\circ\text{C}} = 0.001008 \text{ m}^3/\text{kg}.$$

Substituting,

$$W_{e,\text{in}} = m(u_2 - u_1) \longrightarrow u_2 = u_1 + \frac{W_{e,\text{in}}}{m} = 167.53 \text{ kJ/kg} + \frac{900 \text{ kJ}}{3 \text{ kg}} = 467.53 \text{ kJ/kg}$$

The final state is compressed liquid. Noting that the specific volume is constant during the process, the final temperature is determined using EES to be

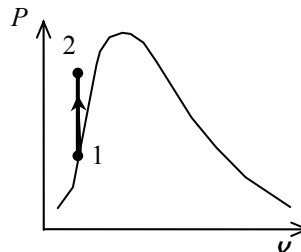
$$\left. \begin{array}{l} u_2 = 467.53 \text{ kJ/kg} \\ \nu_1 = \nu_2 = 0.001008 \text{ m}^3/\text{kg} \end{array} \right\} T_2 = \mathbf{118.9^\circ\text{C}} \quad (\text{from EES})$$

Discussion We cannot find this temperature directly from steam tables at the end of the book.

Approximating the final compressed liquid state as saturated liquid at the given internal energy, the final temperature is determined from Table A-4 to be

$$T_2 \cong T_{\text{sat}@u=467.53 \text{ kJ/kg}} = 111.5^\circ\text{C}$$

Note that this approximation resulted in an answer, which is not very close to the exact answer determined using EES.



5-103 The boundary work expression during a polytropic process of an ideal gas is to be derived.

Assumptions The process is quasi-equilibrium.

Analysis For a polytropic process,

$$P_1 \nu_1^n = P_2 \nu_2^n = \text{Constant}$$

Substituting this into the boundary work expression gives

$$\begin{aligned} w_{b,\text{out}} &= \int_1^2 P d\nu = P_1 \nu_1^n \int_1^2 \nu^{-n} d\nu = \frac{P_1 \nu_1}{1-n} (\nu_2^{1-n} - \nu_1^{1-n}) \\ &= \frac{P_1}{1-n} \left(\frac{\nu_2}{\nu_1^n} \nu_1^n - \frac{\nu_1}{\nu_1^n} \nu_1^n \right) \\ &= \frac{P_1 \nu_1}{1-n} (\nu_2^{1-n} \nu_1^{n-1} - 1) \\ &= \frac{RT_1}{1-n} \left[\left(\frac{\nu_2}{\nu_1} \right)^{1-n} - 1 \right] \end{aligned}$$

When the specific volume ratio is eliminated with the expression for the constant,

$$w_{b,\text{out}} = \frac{RT_1}{1-n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]$$

where $n \neq 1$

5-104 A cylinder equipped with an external spring is initially filled with air at a specified state. Heat is transferred to the air, and both the temperature and pressure rise. The total boundary work done by the air, and the amount of work done against the spring are to be determined, and the process is to be shown on a P - v diagram.

Assumptions 1 The process is quasi-equilibrium. 2 The spring is a linear spring.

Analysis (a) The pressure of the gas changes linearly with volume during this process, and thus the process curve on a P - V diagram will be a straight line. Then the boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

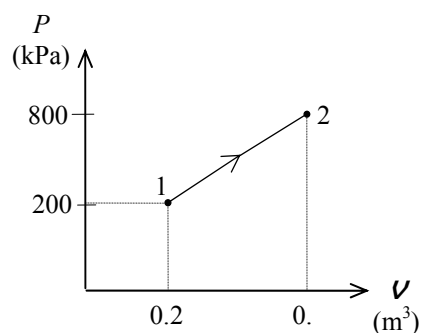
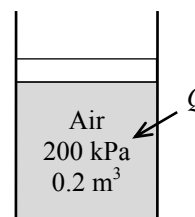
$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2} (v_2 - v_1) \\ &= \frac{(200 + 800)\text{kPa}}{2} (0.5 - 0.2)\text{m}^3 \left(\frac{1\text{ kJ}}{1\text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{150\text{ kJ}} \end{aligned}$$

(b) If there were no spring, we would have a constant pressure process at $P = 200\text{ kPa}$. The work done during this process is

$$\begin{aligned} W_{b,\text{out,no spring}} &= \int_1^2 P dV = P(v_2 - v_1) \\ &= (200\text{ kPa})(0.5 - 0.2)\text{m}^3 / \text{kg} \left(\frac{1\text{ kJ}}{1\text{ kPa} \cdot \text{m}^3} \right) \\ &= 60\text{ kJ} \end{aligned}$$

Thus,

$$W_{\text{spring}} = W_b - W_{b,\text{no spring}} = 150 - 60 = \mathbf{90\text{ kJ}}$$



5-105 A cylinder equipped with a set of stops for the piston is initially filled with saturated liquid-vapor mixture of water at a specified pressure. Heat is transferred to the water until the volume increases by 20%. The initial and final temperature, the mass of the liquid when the piston starts moving, and the work done during the process are to be determined, and the process is to be shown on a P - v diagram.

Assumptions The process is quasi-equilibrium.

Analysis (a) Initially the system is a saturated mixture at 125 kPa pressure, and thus the initial temperature is

$$T_1 = T_{\text{sat}@125 \text{ kPa}} = \mathbf{106.0^\circ\text{C}}$$

The total initial volume is

$$V_1 = m_f v_f + m_g v_g = 2 \times 0.001048 + 3 \times 1.3750 = 4.127 \text{ m}^3$$

Then the total and specific volumes at the final state are

$$V_3 = 1.2 V_1 = 1.2 \times 4.127 = 4.953 \text{ m}^3$$

$$v_3 = \frac{V_3}{m} = \frac{4.953 \text{ m}^3}{5 \text{ kg}} = 0.9905 \text{ m}^3/\text{kg}$$

Thus,

$$\left. \begin{array}{l} P_3 = 300 \text{ kPa} \\ v_3 = 0.9905 \text{ m}^3/\text{kg} \end{array} \right\} T_3 = \mathbf{373.6^\circ\text{C}}$$

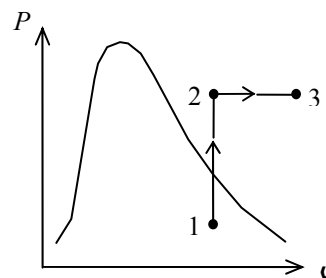
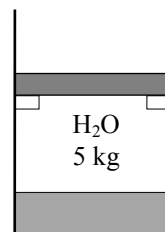
(b) When the piston first starts moving, $P_2 = 300 \text{ kPa}$ and $V_2 = V_1 = 4.127 \text{ m}^3$. The specific volume at this state is

$$v_2 = \frac{V_2}{m} = \frac{4.127 \text{ m}^3}{5 \text{ kg}} = 0.8254 \text{ m}^3/\text{kg}$$

which is greater than $v_g = 0.60582 \text{ m}^3/\text{kg}$ at 300 kPa. Thus **no liquid** is left in the cylinder when the piston starts moving.

(c) No work is done during process 1-2 since $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_b = \int_2^3 P dV = P_2 (V_3 - V_2) = (300 \text{ kPa})(4.953 - 4.127) \text{ m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{247.6 \text{ kJ}}$$



5-106E A spherical balloon is initially filled with air at a specified state. The pressure inside is proportional to the square of the diameter. Heat is transferred to the air until the volume doubles. The work done is to be determined.

Assumptions 1 Air is an ideal gas. 2 The process is quasi-equilibrium.

Properties The gas constant of air is $R = 0.06855 \text{ Btu/lbm} \cdot \text{R}$ (Table A-1E).

Analysis The dependence of pressure on volume can be expressed as

$$\mathcal{V} = \frac{1}{6} \pi D^3 \longrightarrow D = \left(\frac{6\mathcal{V}}{\pi} \right)^{1/3}$$

$$P \propto D^2 \longrightarrow P = kD^2 = k \left(\frac{6\mathcal{V}}{\pi} \right)^{2/3}$$

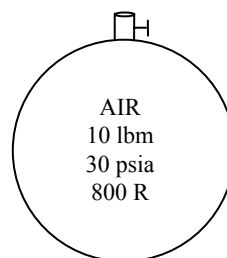
or,
$$k \left(\frac{6}{\pi} \right)^{2/3} = P_1 \mathcal{V}_1^{-2/3} = P_2 \mathcal{V}_2^{-2/3}$$

Also,
$$\frac{P_2}{P_1} = \left(\frac{\mathcal{V}_2}{\mathcal{V}_1} \right)^{2/3} = 2^{2/3} = 1.587$$

and
$$\frac{P_1 \mathcal{V}_1}{T_1} = \frac{P_2 \mathcal{V}_2}{T_2} \longrightarrow T_2 = \frac{P_2 \mathcal{V}_2}{P_1 \mathcal{V}_1} T_1 = 1.587 \times 2 \times (800 \text{ R}) = 2539 \text{ R}$$

Thus,

$$\begin{aligned} W_b &= \int_1^2 P d\mathcal{V} = \int_1^2 k \left(\frac{6\mathcal{V}}{\pi} \right)^{2/3} d\mathcal{V} = \frac{3k}{5} \left(\frac{6}{\pi} \right)^{2/3} (\mathcal{V}_2^{5/3} - \mathcal{V}_1^{5/3}) = \frac{3}{5} (P_2 \mathcal{V}_2 - P_1 \mathcal{V}_1) \\ &= \frac{3}{5} mR(T_2 - T_1) = \frac{3}{5} (10 \text{ lbm})(0.06855 \text{ Btu/lbm} \cdot \text{R})(2539 - 800) \text{ R} = \mathbf{715 \text{ Btu}} \end{aligned}$$



5-107E EES Problem 5-106E is reconsidered. Using the integration feature, the work done is to be determined and compared to the 'hand calculated' result.

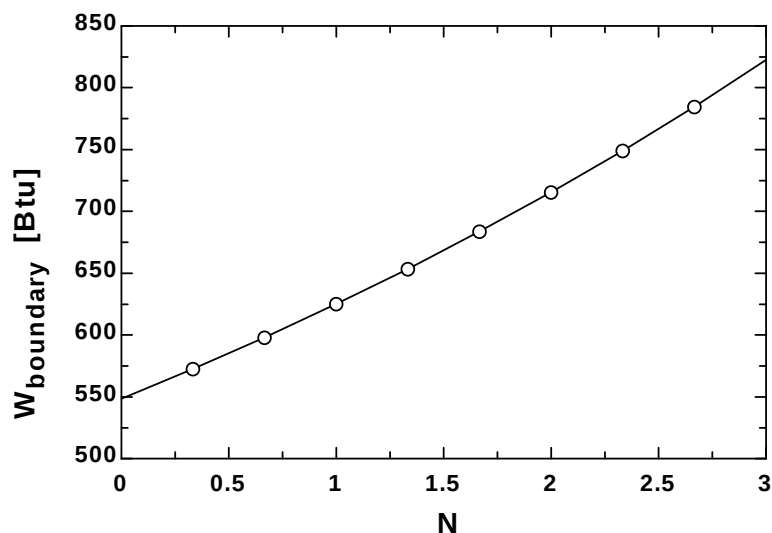
Analysis The problem is solved using EES, and the solution is given below.

```

N=2
m=10 [lbm]
P_1=30 [psia]
T_1=800 [R]
V_2=2*V_1
R=1545"[ft-lbf/lbmol-R]"/molar mass(air)"[ft-lbf/lbm-R]"
P_1*Convert(psia,lbf/ft^2)*V_1=m*R*T_1
V_1=4*pi*(D_1/2)^3/3"[ft^3]"
C=P_1/D_1^N
(D_1/D_2)^3=V_1/V_2
P_2=C*D_2^N"[psia]"
P_2*Convert(psia,lbf/ft^2)*V_2=m*R*T_2
P=C*D_1^N*Convert(psia,lbf/ft^2)"[ft^2]"
V=4*pi*(D/2)^3/3"[ft^3]"
W_boundary_EES=integral(P,V,V_1,V_2)*convert(ft-lbf,Btu)"[Btu]"
W_boundary_HAND=pi*C/(2*(N+3))*(D_2^(N+3)-D_1^(N+3))*convert(ft-lbf,Btu)*convert(ft^2,in^2)"[Btu]"

```

N	W _{boundary} [Btu]
0	548.3
0.3333	572.5
0.6667	598.1
1	625
1.333	653.5
1.667	683.7
2	715.5
2.333	749.2
2.667	784.8
3	822.5



5-108 A cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant is heated both electrically and by heat transfer at constant pressure for 6 min. The electric current is to be determined, and the process is to be shown on a T - ν diagram.

Assumptions 1 The cylinder is stationary and thus the kinetic and potential energy changes are negligible.

2 The thermal energy stored in the cylinder itself and the wires is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{e,\text{in}} - W_{b,\text{out}} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{in}} + W_{e,\text{in}} = m(h_2 - h_1)$$

$$Q_{\text{in}} + (VI\Delta t) = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)

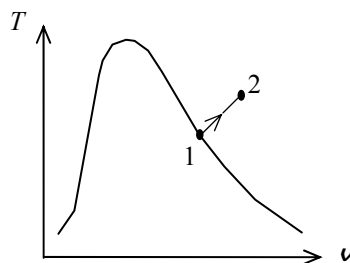
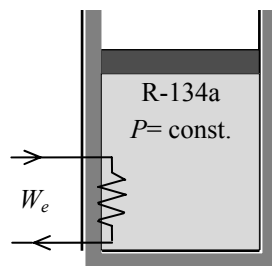
$$\left. \begin{array}{l} P_1 = 240 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} h_1 = h_{g@240\text{kPa}} = 247.28 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 240 \text{ kPa} \\ T_1 = 70^\circ\text{C} \end{array} \right\} h_2 = 314.51 \text{ kJ/kg}$$

Substituting,

$$300,000 \text{ VAs} + (110 \text{ V})(I)(6 \times 60 \text{ s}) = (12 \text{ kg})(314.51 - 247.28) \text{ kJ/kg} \left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}} \right)$$

$$I = \mathbf{12.8 \text{ A}}$$



5-109 A cylinder is initially filled with saturated liquid-vapor mixture of R-134a at a specified pressure. Heat is transferred to the cylinder until the refrigerant vaporizes completely at constant pressure. The initial volume, the work done, and the total heat transfer are to be determined, and the process is to be shown on a P - v diagram.

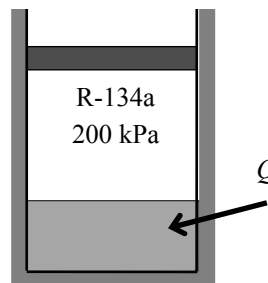
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis (a) Using property data from R-134a tables (Tables A-11 through A-13), the initial volume of the refrigerant is determined to be

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.25 \end{array} \right\} \begin{array}{l} v_f = 0.0007533, \quad v_g = 0.099867 \text{ m}^3/\text{kg} \\ u_f = 38.28, \quad u_g = 186.21 \text{ kJ/kg} \end{array}$$

$$\begin{aligned} v_1 &= v_f + x_1 v_{fg} \\ &= 0.0007533 + 0.25 \times (0.099867 - 0.0007533) = 0.02553 \text{ m}^3/\text{kg} \\ u_1 &= u_f + x_1 u_{fg} = 38.28 + 0.25 \times 186.21 = 84.83 \text{ kJ/kg} \end{aligned}$$

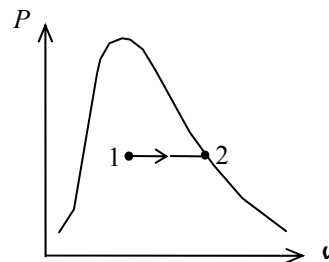
$$V_1 = m v_1 = (0.2 \text{ kg})(0.02553 \text{ m}^3/\text{kg}) = \mathbf{0.005106 \text{ m}^3}$$



(b) The work done during this constant pressure process is

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} v_2 = v_{g@200 \text{ kPa}} = 0.09987 \text{ m}^3/\text{kg} \\ u_2 = u_{g@200 \text{ kPa}} = 224.48 \text{ kJ/kg} \end{array}$$

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P dV = P(v_2 - v_1) = mP(v_2 - v_1) \\ &= (0.2 \text{ kg})(200 \text{ kPa})(0.09987 - 0.02553) \text{ m}^3/\text{kg} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{2.97 \text{ kJ}} \end{aligned}$$



(c) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ Q_{\text{in}} - W_{b,\text{out}} &= \Delta U \\ Q_{\text{in}} &= m(u_2 - u_1) + W_{b,\text{out}} \end{aligned}$$

Substituting,

$$Q_{\text{in}} = (0.2 \text{ kg})(224.48 - 84.83) \text{ kJ/kg} + 2.97 = \mathbf{30.9 \text{ kJ}}$$

5-110 A cylinder is initially filled with helium gas at a specified state. Helium is compressed polytropically to a specified temperature and pressure. The heat transfer during the process is to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The gas constant of helium is $R = 2.0769 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1). Also, $c_v = 3.1156 \text{ kJ/kg} \cdot \text{K}$ (Table A-2).

Analysis The mass of helium and the exponent n are determined to be

$$m = \frac{P_1 V_1}{RT_1} = \frac{(150 \text{ kPa})(0.5 \text{ m}^3)}{(2.0769 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 0.123 \text{ kg}$$

$$\frac{P_1 V_1}{RT_1} = \frac{P_2 V_2}{RT_2} \longrightarrow V_2 = \frac{T_2 P_1}{T_1 P_2} V_1 = \frac{413 \text{ K}}{293 \text{ K}} \times \frac{150 \text{ kPa}}{400 \text{ kPa}} \times 0.5 \text{ m}^3 = 0.264 \text{ m}^3$$

$$P_2 V_2^n = P_1 V_1^n \longrightarrow \left(\frac{P_2}{P_1} \right) = \left(\frac{V_1}{V_2} \right)^n \longrightarrow \frac{400}{150} = \left(\frac{0.5}{0.264} \right)^n \longrightarrow n = 1.536$$

Then the boundary work for this polytropic process can be determined from

$$W_{b,\text{in}} = -\int_1^2 P dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.123 \text{ kg})(2.0769 \text{ kJ/kg} \cdot \text{K})(413 - 293) \text{ K}}{1 - 1.536} = 57.2 \text{ kJ}$$

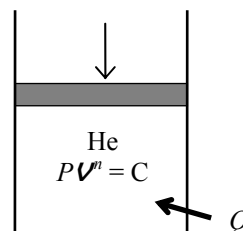
We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} + W_{b,\text{in}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} = m(u_2 - u_1) - W_{b,\text{in}}$$

$$= mc_v(T_2 - T_1) - W_{b,\text{in}}$$



Substituting,

$$Q_{\text{in}} = (0.123 \text{ kg})(3.1156 \text{ kJ/kg} \cdot \text{K})(413 - 293) \text{ K} - (57.2 \text{ kJ}) = \mathbf{-11.2 \text{ kJ}}$$

The negative sign indicates that heat is lost from the system.

5-111 Nitrogen gas is expanded in a polytropic process. The work done and the heat transfer are to be determined.

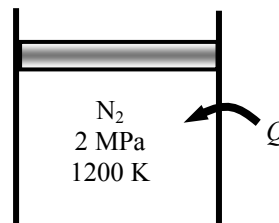
Assumptions 1 Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K and 3.39 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used.

Properties The properties of nitrogen are $R = 0.2968 \text{ kJ/kg}\cdot\text{K}$ and $c_v = 0.743 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis We take nitrogen in the cylinder as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$



Using the boundary work relation for the polytropic process of an ideal gas gives

$$w_{b,\text{out}} = \frac{RT_1}{1-n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] = \frac{(0.2968 \text{ kJ/kg}\cdot\text{K})(1200 \text{ K})}{1-1.35} \left[\left(\frac{120}{2000} \right)^{0.35/1.35} - 1 \right] = \mathbf{526.9 \text{ kJ/kg}}$$

The temperature at the final state is

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(n-1)/n} = (1200 \text{ K}) \left(\frac{120 \text{ kPa}}{2000 \text{ kPa}} \right)^{0.35/1.35} = 578.6 \text{ K}$$

Substituting into the energy balance equation,

$$q_{\text{in}} = w_{b,\text{out}} + c_v(T_2 - T_1) = 526.9 \text{ kJ/kg} + (0.743 \text{ kJ/kg}\cdot\text{K})(578.6 - 1200) \text{ K} = \mathbf{65.2 \text{ kJ/kg}}$$

5-112 The expansion work from P_1 to P_2 in a closed system polytropic process is to be compared for neon and air.

Assumptions The process is quasi-equilibrium.

Properties The gas constants for neon and air are $R = 0.4119$ and $0.287 \text{ kJ/kg}\cdot\text{K}$, respectively (Table A-2a).

Analysis For a polytropic process,

$$Pv^n = \text{Constant}$$

The boundary work during a polytropic process of an ideal gas is

$$w_{b,\text{out}} = \int_1^2 P dv = \text{Constant} \int_1^2 v^{-n} dv = \frac{P_1 v_1}{1-n} \left[\left(\frac{v_2}{v_1} \right)^{1-n} - 1 \right] = \frac{RT_1}{1-n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]$$

Inspection of this equation reveals that the gas with the largest gas constant (i.e., smallest molecular weight) will produce the greatest amount of work. Hence, the neon gas will produce the greatest amount of work.

5-113 A cylinder and a rigid tank initially contain the same amount of an ideal gas at the same state. The temperature of both systems is to be raised by the same amount. The amount of extra heat that must be transferred to the cylinder is to be determined.

Analysis In the absence of any work interactions, other than the boundary work, the ΔH and ΔU represent the heat transfer for ideal gases for constant pressure and constant volume processes, respectively. Thus the extra heat that must be supplied to the air maintained at constant pressure is

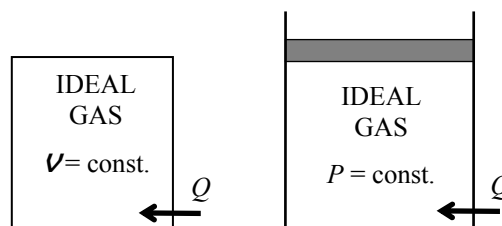
$$Q_{\text{in, extra}} = \Delta H - \Delta U = mc_p\Delta T - mc_v\Delta T = m(c_p - c_v)\Delta T = mR\Delta T$$

where

$$R = \frac{R_u}{M} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{25 \text{ kg/kmol}} = 0.3326 \text{ kJ/kg} \cdot \text{K}$$

Substituting,

$$Q_{\text{in, extra}} = (12 \text{ kg})(0.3326 \text{ kJ/kg} \cdot \text{K})(15 \text{ K}) = \mathbf{59.9 \text{ kJ}}$$



5-114 The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

Assumptions 1 Water is an incompressible substance with constant specific heats. 2 The energy stored in the glass containers themselves is negligible relative to the energy stored in water. 3 The house is maintained at 22°C at all times.

Properties The density and specific heat of water at room temperature are $\rho = 1 \text{ kg/L}$ and $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3).

Analysis (a) The total mass of water is

$$m_w = \rho V = (1 \text{ kg/L})(50 \times 20 \text{ L}) = 1000 \text{ kg}$$

Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

$$= (\Delta U)_{\text{water}} = mc(T_2 - T_1)_{\text{water}}$$

or, $\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$

Substituting,

$$(15 \text{ kJ/s})\Delta t - (50,000 \text{ kJ/h})(10 \text{ h}) = (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(22 - 80)^\circ\text{C}$$

It gives $\Delta t = 17,170 \text{ s} = \mathbf{4.77 \text{ h}}$

(b) If the house incorporated no solar heating, the energy balance relation above would simplify further to

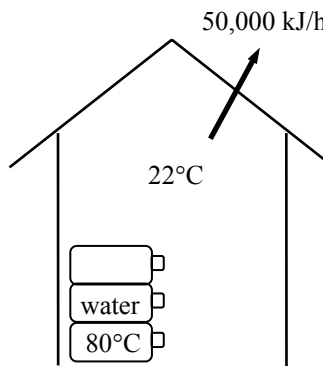
$$\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = 0$$

Substituting,

$$(15 \text{ kJ/s})\Delta t - (50,000 \text{ kJ/h})(10 \text{ h}) = 0$$

It gives

$$\Delta t = 33,333 \text{ s} = \mathbf{9.26 \text{ h}}$$



5-115 An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature is to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

Properties The specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3).

Analysis Taking the water in the container as the system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

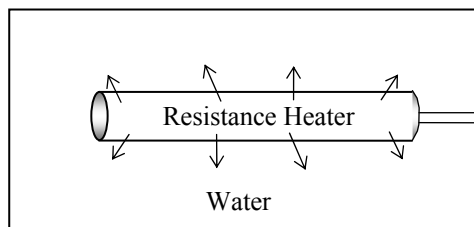
$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)_{\text{water}}$$

Substituting,

$$(1800 \text{ J/s})\Delta t = (40 \text{ kg})(4180 \text{ J/kg}\cdot^\circ\text{C})(80 - 20)^\circ\text{C}$$

Solving for Δt gives

$$\Delta t = 5573 \text{ s} = 92.9 \text{ min} = 1.55 \text{ h}$$



5-116 One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room is to be determined.

Assumptions 1 The room is well insulated and well sealed. 2 The thermal properties of water and air are constant.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1). The specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis The volume and the mass of the air in the room are

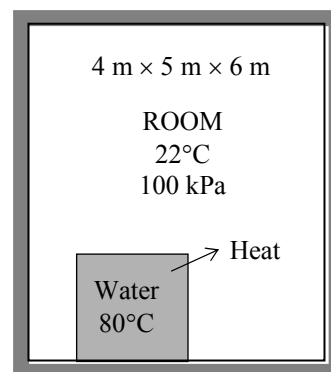
$$V = 4 \times 5 \times 6 = 120 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1 V_1}{R T_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.2870 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(295 \text{ K})} = 141.7 \text{ kg}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$



or

$$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting,

$$(1000 \text{ kg})(4.180 \text{ kJ/kg} \cdot ^\circ\text{C})(T_f - 80)^\circ\text{C} + (141.7 \text{ kg})(0.718 \text{ kJ/kg} \cdot ^\circ\text{C})(T_f - 22)^\circ\text{C} = 0$$

It gives

$$T_f = \mathbf{78.6^\circ\text{C}}$$

where T_f is the final equilibrium temperature in the room.

5-117 A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it is to meet the heating requirements of this room for a 25-h period.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 25-h period.

Properties The specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3).

Analysis Heat loss from the room during a 25-h period is

$$Q_{\text{loss}} = (8000 \text{ kJ/h})(24 \text{ h}) = 192,000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}} \quad \text{?}$$

or

$$-Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

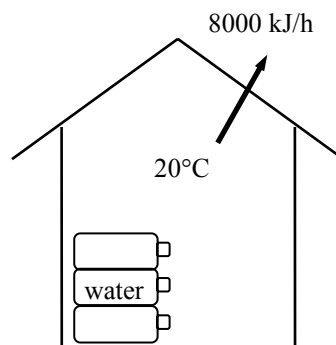
Substituting,

$$-192,000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(20 - T_1)$$

It gives

$$T_1 = \mathbf{65.9^\circ\text{C}}$$

where T_1 is the temperature of the water when it is first brought into the room.



5-118 A sample of a food is burned in a bomb calorimeter, and the water temperature rises by 3.2°C when equilibrium is established. The energy content of the food is to be determined.

Assumptions 1 Water is an incompressible substance with constant specific heats. 2 Air is an ideal gas with constant specific heats. 3 The energy stored in the reaction chamber is negligible relative to the energy stored in water. 4 The energy supplied by the mixer is negligible.

Properties The specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg}\cdot^{\circ}\text{C}$ (Table A-3). The constant volume specific heat of air at room temperature is $c_v = 0.718 \text{ kJ/kg}\cdot^{\circ}\text{C}$ (Table A-2).

Analysis The chemical energy released during the combustion of the sample is transferred to the water as heat. Therefore, disregarding the change in the sensible energy of the reaction chamber, the energy content of the food is simply the heat transferred to the water. Taking the water as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \rightarrow Q_{\text{in}} = \Delta U$$

or

$$Q_{\text{in}} = (\Delta U)_{\text{water}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$Q_{\text{in}} = (3 \text{ kg})(4.18 \text{ kJ/kg}\cdot^{\circ}\text{C})(3.2^{\circ}\text{C}) = 40.13 \text{ kJ}$$

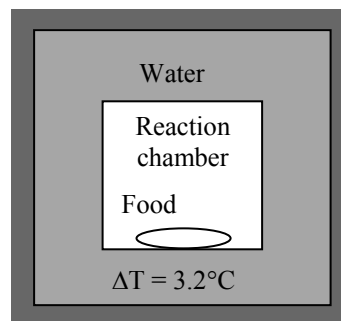
for a 2-g sample. Then the energy content of the food per unit mass is

$$\frac{40.13 \text{ kJ}}{2 \text{ g}} \left(\frac{1000 \text{ g}}{1 \text{ kg}} \right) = \mathbf{20,060 \text{ kJ/kg}}$$

To make a rough estimate of the error involved in neglecting the thermal energy stored in the reaction chamber, we treat the entire mass within the chamber as air and determine the change in sensible internal energy:

$$(\Delta U)_{\text{chamber}} = [mc_v(T_2 - T_1)]_{\text{chamber}} = (0.102 \text{ kg})(0.718 \text{ kJ/kg}\cdot^{\circ}\text{C})(3.2^{\circ}\text{C}) = 0.23 \text{ kJ}$$

which is less than 1% of the internal energy change of water. Therefore, it is reasonable to disregard the change in the sensible energy content of the reaction chamber in the analysis.



5-119 A man drinks one liter of cold water at 3°C in an effort to cool down. The drop in the average body temperature of the person under the influence of this cold water is to be determined.

Assumptions 1 Thermal properties of the body and water are constant. **2** The effect of metabolic heat generation and the heat loss from the body during that time period are negligible.

Properties The density of water is very nearly 1 kg/L, and the specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3). The average specific heat of human body is given to be $3.6 \text{ kJ/kg}\cdot^\circ\text{C}$.

Analysis. The mass of the water is

$$m_w = \rho V = (1 \text{ kg/L})(1 \text{ L}) = 1 \text{ kg}$$

We take the man and the water as our system, and disregard any heat and mass transfer and chemical reactions. Of course these assumptions may be acceptable only for very short time periods, such as the time it takes to drink the water. Then the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = \Delta U_{\text{body}} + \Delta U_{\text{water}}$$

or
$$[mc(T_2 - T_1)]_{\text{body}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting
$$(68 \text{ kg})(3.6 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 39)^\circ\text{C} + (1 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 3)^\circ\text{C} = 0$$

It gives

$$T_f = 38.4^\circ\text{C}$$

Then

$$\Delta T = 39 - 38.4 = \mathbf{0.6^\circ\text{C}}$$

Therefore, the average body temperature of this person should drop about half a degree celsius.



5-120 A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. The amount of ice or cold water that needs to be added to the water is to be determined.

Assumptions 1 Thermal properties of the ice and water are constant. 2 Heat transfer to the glass is negligible. 3 There is no stirring by hand or a mechanical device (it will add energy).

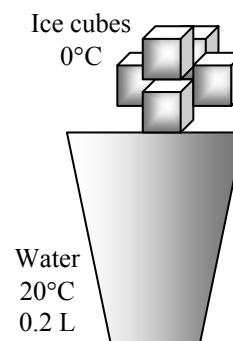
Properties The density of water is 1 kg/L, and the specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3). The specific heat of ice at about 0°C is $c = 2.11$ kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg,.

Analysis (a) The mass of the water is

$$m_w = \rho V = (1 \text{ kg/L})(0.2 \text{ L}) = 0.2 \text{ kg}$$

We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ 0 &= \Delta U \\ 0 &= \Delta U_{\text{ice}} + \Delta U_{\text{water}} \end{aligned}$$



$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{\text{if}} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Noting that $T_{1,\text{ice}} = 0^\circ\text{C}$ and $T_2 = 5^\circ\text{C}$ and substituting gives

$$m[0 + 333.7 \text{ kJ/kg} + (4.18 \text{ kJ/kg}\cdot^\circ\text{C})(5 - 0)^\circ\text{C}] + (0.2 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(5 - 20)^\circ\text{C} = 0$$

$$m = 0.0364 \text{ kg} = \mathbf{36.4 \text{ g}}$$

(b) When $T_{1,\text{ice}} = -8^\circ\text{C}$ instead of 0°C , substituting gives

$$\begin{aligned} m[(2.11 \text{ kJ/kg}\cdot^\circ\text{C})[0 - (-8)]^\circ\text{C} + 333.7 \text{ kJ/kg} + (4.18 \text{ kJ/kg}\cdot^\circ\text{C})(5 - 0)^\circ\text{C}] \\ + (0.2 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(5 - 20)^\circ\text{C} = 0 \end{aligned}$$

$$m = 0.0347 \text{ kg} = \mathbf{34.7 \text{ g}}$$

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by a term for cold water at 0°C :

$$\begin{aligned} (\Delta U)_{\text{cold water}} + (\Delta U)_{\text{water}} &= 0 \\ [mc(T_2 - T_1)]_{\text{cold water}} + [mc(T_2 - T_1)]_{\text{water}} &= 0 \end{aligned}$$

Substituting,

$$[m_{\text{cold water}} (4.18 \text{ kJ/kg}\cdot^\circ\text{C})(5 - 0)^\circ\text{C}] + (0.2 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(5 - 20)^\circ\text{C} = 0$$

It gives

$$m = 0.6 \text{ kg} = \mathbf{600 \text{ g}}$$

Discussion Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Also, the temperature of ice does not seem to make a significant difference.

5-121 EES Problem 5-120 is reconsidered. The effect of the initial temperature of the ice on the final mass of ice required as the ice temperature varies from -20°C to 0°C is to be investigated. The mass of ice is to be plotted against the initial temperature of ice.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns"

rho_water = 1 [kg/L]
 V = 0.2 [L]
 T_1_ice = 0 [C]
 T_1 = 20 [C]
 T_2 = 5 [C]
 C_ice = 2.11 [kJ/kg-C]
 C_water = 4.18 [kJ/kg-C]
 h_if = 333.7 [kJ/kg]
 T_1_ColdWater = 0 [C]

"The mass of the water is:"

m_water = rho_water*V "[kg]"

"The system is the water plus the ice. Assume a short time period and neglect any heat and mass transfer. The energy balance becomes:"

E_in - E_out = DELTAE_sys "[kJ]"

E_in = 0 "[kJ]"

E_out = 0 "[kJ]"

DELTAE_sys = DELTAU_water+DELTAE_ice "[kJ]"

DELTAE_water = m_water*C_water*(T_2 - T_1) "[kJ]"

DELTAE_ice = DELTAU_solid_ice+DELTAE_melted_ice "[kJ]"

DELTAE_solid_ice = m_ice*C_ice*(0-T_1_ice) + m_ice*h_if "[kJ]"

DELTAE_melted_ice = m_ice*C_water*(T_2 - 0) "[kJ]"

m_ice_grams = m_ice*convert(kg,g) "[g]"

"Cooling with Cold Water:"

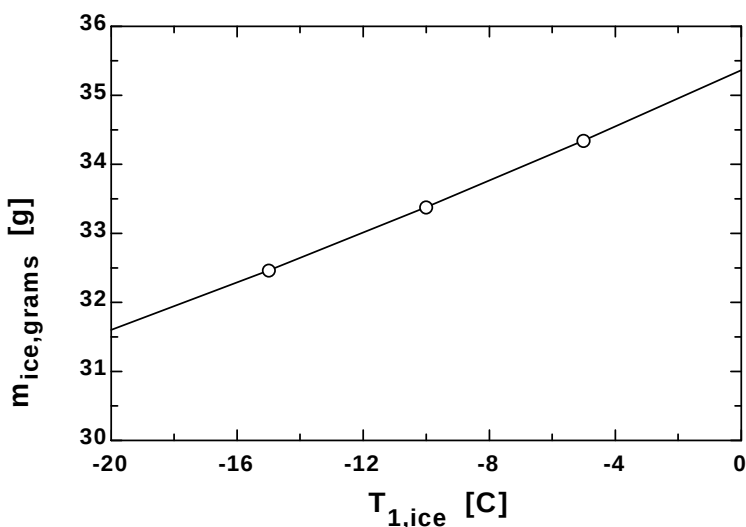
DELTAE_sys = DELTAU_water+DELTAE_ColdWater "[kJ]"

DELTAE_water = m_water*C_water*(T_2_ColdWater - T_1) "[kJ]"

DELTAE_ColdWater = m_ColdWater*C_water*(T_2_ColdWater - T_1_ColdWater) "[kJ]"

m_ColdWater_grams = m_ColdWater*convert(kg,g) "[g]"

m _{ice,grams} [g]	T _{1,ice} [C]
31.6	-20
32.47	-15
33.38	-10
34.34	-5
35.36	0



5-122 Carbon dioxide is compressed polytropically in a piston-cylinder device. The final temperature is to be determined treating the carbon dioxide as an ideal gas and a van der Waals gas.

Assumptions The process is quasi-equilibrium.

Properties The gas constant of carbon dioxide is $R = 0.1889 \text{ kJ/kg} \cdot \text{K}$ (Table A-1).

Analysis (a) The initial specific volume is

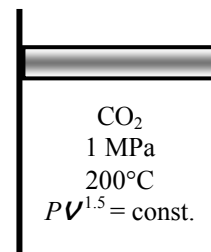
$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kJ/kg} \cdot \text{K})(473 \text{ K})}{1000 \text{ kPa}} = 0.08935 \text{ m}^3/\text{kg}$$

From polytropic process expression,

$$\nu_2 = \nu_1 \left(\frac{P_1}{P_2} \right)^{1/n} = (0.08935 \text{ m}^3/\text{kg}) \left(\frac{1000}{3000} \right)^{1/1.5} = 0.04295 \text{ m}^3/\text{kg}$$

The final temperature is then

$$T_2 = \frac{P_2 \nu_2}{R} = \frac{(3000 \text{ kPa})(0.04295 \text{ m}^3/\text{kg})}{0.1889 \text{ kJ/kg} \cdot \text{K}} = \mathbf{682.1 \text{ K}}$$



(b) The van der Waals equation of state for carbon dioxide is

$$\left(P + \frac{365.8}{\bar{\nu}^2} \right) (\bar{\nu} - 0.0428) = R_u T$$

When this is applied to the initial state, the result is

$$\left(1000 + \frac{365.8}{\bar{\nu}_1^2} \right) (\bar{\nu}_1 - 0.0428) = (8.314)(473)$$

whose solution by iteration or by EES is

$$\bar{\nu}_1 = 3.882 \text{ m}^3/\text{kmol}$$

The final molar specific volume is then

$$\bar{\nu}_2 = \bar{\nu}_1 \left(\frac{P_1}{P_2} \right)^{1/n} = (3.882 \text{ m}^3/\text{kmol}) \left(\frac{1000}{3000} \right)^{1/1.5} = 1.866 \text{ m}^3/\text{kmol}$$

Substitution of the final molar specific volume into the van der Waals equation of state produces

$$T_2 = \frac{1}{R_u} \left(P + \frac{365.8}{\bar{\nu}^2} \right) (\bar{\nu} - 0.0428) = \frac{1}{8.314} \left(3000 + \frac{365.8}{(1.866)^2} \right) (1.866 - 0.0428) = \mathbf{680.9 \text{ K}}$$

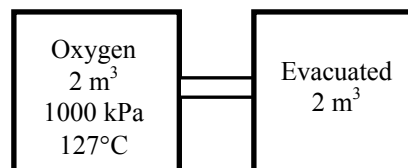
5-123 Two adiabatic chambers are connected by a valve. One chamber contains oxygen while the other one is evacuated. The valve is now opened until the oxygen fills both chambers and both tanks have the same pressure. The total internal energy change and the final pressure in the tanks are to be determined.

Assumptions 1 Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 154.8 K and 5.08 MPa. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 3 Constant specific heats at room temperature can be used. 4 Both chambers are insulated and thus heat transfer is negligible.

Analysis We take both chambers as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$0 = \Delta U = mc_v(T_2 - T_1)$$



Since the internal energy does not change, the temperature of the air will also not change. Applying the ideal gas equation gives

$$P_1 V_1 = P_2 V_2 \longrightarrow P_2 = P_1 \frac{V_1}{V_2} = (1000 \text{ kPa}) \frac{2 \text{ m}^3}{4 \text{ m}^3} = \mathbf{500 \text{ kPa}}$$

5-124 ... 5-129 Design and Essay Problems

5-128 A claim that fruits and vegetables are cooled by 6°C for each percentage point of weight loss as moisture during vacuum cooling is to be evaluated.

Analysis Assuming the fruits and vegetables are cooled from 30°C and 0°C, the average heat of vaporization can be taken to be 2466 kJ/kg, which is the value at 15°C, and the specific heat of products can be taken to be 4 kJ/kg·°C. Then the vaporization of 0.01 kg water will lower the temperature of 1 kg of produce by $24.66/4 = 6^\circ\text{C}$. Therefore, the vacuum cooled products will lose 1 percent moisture for each 6°C drop in temperature. Thus the claim is **reasonable**.



Solutions Manual

for

Introduction to Thermodynamics and Heat Transfer**Yunus A. Cengel****2nd Edition, 2008****Chapter 6****MASS AND ENERGY ANALYSIS OF
CONTROL VOLUMES****PROPRIETARY AND CONFIDENTIAL**

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Conservation of Mass

6-1C Mass, energy, momentum, and electric charge are conserved, and volume and entropy are not conserved during a process.

6-2C Mass flow rate is the amount of mass flowing through a cross-section per unit time whereas the volume flow rate is the amount of volume flowing through a cross-section per unit time.

6-3C The amount of mass or energy entering a control volume does not have to be equal to the amount of mass or energy leaving during an unsteady-flow process.

6-4C Flow through a control volume is steady when it involves no changes with time at any specified position.

6-5C No, a flow with the same volume flow rate at the inlet and the exit is not necessarily steady (unless the density is constant). To be steady, the mass flow rate through the device must remain constant.

6-6E A garden hose is used to fill a water bucket. The volume and mass flow rates of water, the filling time, and the discharge velocity are to be determined.

Assumptions 1 Water is an incompressible substance. 2 Flow through the hose is steady. 3 There is no waste of water by splashing.

Properties We take the density of water to be 62.4 lbm/ft^3 (Table A-3E).

Analysis (a) The volume and mass flow rates of water are

$$\dot{V} = AV = (\pi D^2 / 4)V = [\pi(1/12 \text{ ft})^2 / 4](8 \text{ ft/s}) = \mathbf{0.04363 \text{ ft}^3/\text{s}}$$

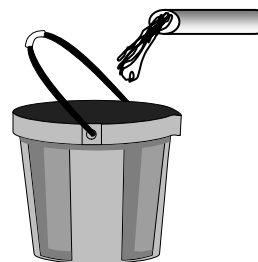
$$\dot{m} = \rho \dot{V} = (62.4 \text{ lbm/ft}^3)(0.04363 \text{ ft}^3/\text{s}) = \mathbf{2.72 \text{ lbm/s}}$$

(b) The time it takes to fill a 20-gallon bucket is

$$\Delta t = \frac{V}{\dot{V}} = \frac{20 \text{ gal}}{0.04363 \text{ ft}^3/\text{s}} \left(\frac{1 \text{ ft}^3}{7.4804 \text{ gal}} \right) = \mathbf{61.3 \text{ s}}$$

(c) The average discharge velocity of water at the nozzle exit is

$$V_e = \frac{\dot{V}}{A_e} = \frac{\dot{V}}{\pi D_e^2 / 4} = \frac{0.04363 \text{ ft}^3/\text{s}}{[\pi(0.5/12 \text{ ft})^2 / 4]} = \mathbf{32 \text{ ft/s}}$$



Discussion Note that for a given flow rate, the average velocity is inversely proportional to the square of the diameter. Therefore, when the diameter is reduced by half, the velocity quadruples.

6-7 Air is accelerated in a nozzle. The mass flow rate and the exit area of the nozzle are to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 2.21 kg/m^3 at the inlet, and 0.762 kg/m^3 at the exit.

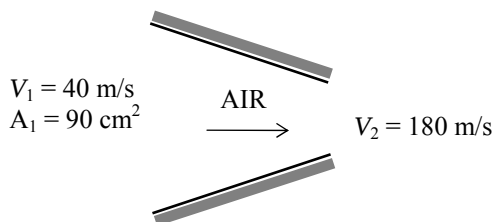
Analysis (a) The mass flow rate of air is determined from the inlet conditions to be

$$\dot{m} = \rho_1 A_1 V_1 = (2.21 \text{ kg/m}^3)(0.009 \text{ m}^2)(40 \text{ m/s}) = \mathbf{0.796 \text{ kg/s}}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

Then the exit area of the nozzle is determined to be

$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.796 \text{ kg/s}}{(0.762 \text{ kg/m}^3)(180 \text{ m/s})} = 0.0058 \text{ m}^2 = \mathbf{58 \text{ cm}^2}$$

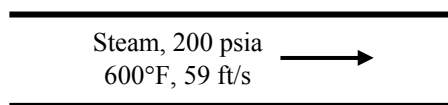


6-8E Steam flows in a pipe. The minimum diameter of the pipe for a given steam velocity is to be determined.

Assumptions Flow through the pipe is steady.

Properties The specific volume of steam at the given state is (Table A-6E)

$$\left. \begin{array}{l} P = 200 \text{ psia} \\ T = 600^\circ\text{F} \end{array} \right\} \nu_1 = 3.0586 \text{ ft}^3/\text{lbm}$$



Analysis The cross sectional area of the pipe is

$$\dot{m} = \frac{1}{\nu} A_c V \longrightarrow A = \frac{\dot{m} \nu}{V} = \frac{(200 \text{ lbm/s})(3.0586 \text{ ft}^3/\text{lbm})}{59 \text{ ft/s}} = 10.37 \text{ ft}^2$$

Solving for the pipe diameter gives

$$A = \frac{\pi D^2}{4} \longrightarrow D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(10.37 \text{ ft}^2)}{\pi}} = \mathbf{3.63 \text{ ft}}$$

Therefore, the diameter of the pipe must be at least 3.63 ft to ensure that the velocity does not exceed 59 ft/s.

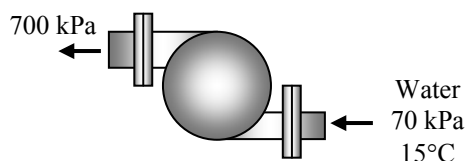
6-9 A water pump increases water pressure. The diameters of the inlet and exit openings are given. The velocity of the water at the inlet and outlet are to be determined.

Assumptions 1 Flow through the pump is steady. 2 The specific volume remains constant.

Properties The inlet state of water is compressed liquid. We approximate it as a saturated liquid at the given temperature. Then, at 15°C and 40°C, we have (Table A-4)

$$\left. \begin{array}{l} T = 15^\circ\text{C} \\ x = 0 \end{array} \right\} \nu_1 = 0.001001 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} T = 40^\circ\text{C} \\ x = 0 \end{array} \right\} \nu_1 = 0.001008 \text{ m}^3/\text{kg}$$



Analysis The velocity of the water at the inlet is

$$V_1 = \frac{\dot{m} \nu_1}{A_1} = \frac{4\dot{m} \nu_1}{\pi D_1^2} = \frac{4(0.5 \text{ kg/s})(0.001001 \text{ m}^3/\text{kg})}{\pi(0.01 \text{ m})^2} = \mathbf{6.37 \text{ m/s}}$$

Since the mass flow rate and the specific volume remains constant, the velocity at the pump exit is

$$V_2 = V_1 \frac{A_1}{A_2} = V_1 \left(\frac{D_1}{D_2} \right)^2 = (6.37 \text{ m/s}) \left(\frac{0.01 \text{ m}}{0.015 \text{ m}} \right)^2 = \mathbf{2.83 \text{ m/s}}$$

Using the specific volume at 40°C, the water velocity at the inlet becomes

$$V_1 = \frac{\dot{m} \nu_1}{A_1} = \frac{4\dot{m} \nu_1}{\pi D_1^2} = \frac{4(0.5 \text{ kg/s})(0.001008 \text{ m}^3/\text{kg})}{\pi(0.01 \text{ m})^2} = \mathbf{6.42 \text{ m/s}}$$

which is a 0.8% increase in velocity.

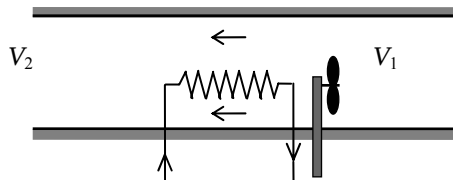
6-10 Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 1.20 kg/m^3 at the inlet, and 1.05 kg/m^3 at the exit.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\begin{aligned}\dot{m}_1 &= \dot{m}_2 \\ \rho_1 A V_1 &= \rho_2 A V_2 \\ \frac{V_2}{V_1} &= \frac{\rho_1}{\rho_2} = \frac{1.20 \text{ kg/m}^3}{1.05 \text{ kg/m}^3} = 1.14 \quad (\text{or, an increase of } \mathbf{14\%})\end{aligned}$$



Therefore, the air velocity increases 14% as it flows through the hair drier.

6-11 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

Properties The density of air is given to be 1.18 kg/m^3 at the beginning, and 7.20 kg/m^3 at the end.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. The mass balance for this system can be expressed as

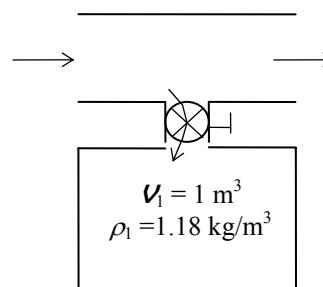
Mass balance:

$$m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1 = \rho_2 V - \rho_1 V$$

Substituting,

$$m_i = (\rho_2 - \rho_1)V = [(7.20 - 1.18) \text{ kg/m}^3](1 \text{ m}^3) = \mathbf{6.02 \text{ kg}}$$

Therefore, 6.02 kg of mass entered the tank.



6-12 A smoking lounge that can accommodate 15 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge and the diameter of the duct are to be determined.

Assumptions Infiltration of air into the smoking lounge is negligible.

Properties The minimum fresh air requirements for a smoking lounge is given to be 30 L/s per person.

Analysis The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

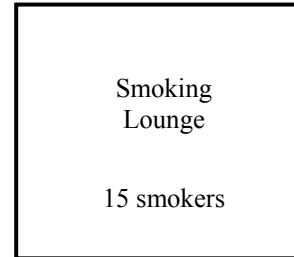
$$\begin{aligned}\dot{V}_{\text{air}} &= \dot{V}_{\text{air per person}} (\text{No. of persons}) \\ &= (30 \text{ L/s} \cdot \text{person})(15 \text{ persons}) = 450 \text{ L/s} = \mathbf{0.45 \text{ m}^3/\text{s}}\end{aligned}$$

The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2 / 4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.45 \text{ m}^3/\text{s})}{\pi(8 \text{ m/s})}} = \mathbf{0.268 \text{ m}}$$



Therefore, the diameter of the fresh air duct should be at least 26.8 cm if the velocity of air is not to exceed 8 m/s.

6-13 The minimum fresh air requirements of a residential building is specified to be 0.35 air changes per hour. The size of the fan that needs to be installed and the diameter of the duct are to be determined.

Analysis The volume of the building and the required minimum volume flow rate of fresh air are

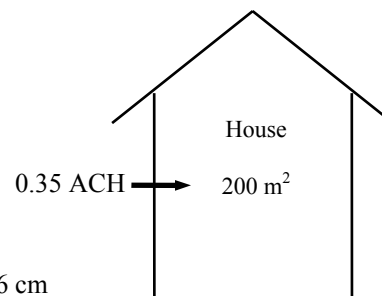
$$\begin{aligned}V_{\text{room}} &= (2.7 \text{ m})(200 \text{ m}^2) = 540 \text{ m}^3 \\ \dot{V} &= V_{\text{room}} \times \text{ACH} = (540 \text{ m}^3)(0.35/\text{h}) = 189 \text{ m}^3/\text{h} = 189,000 \text{ L/h} = \mathbf{3150 \text{ L/min}}\end{aligned}$$

The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2 / 4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(189 / 3600 \text{ m}^3/\text{s})}{\pi(6 \text{ m/s})}} = \mathbf{0.106 \text{ m}}$$



Therefore, the diameter of the fresh air duct should be at least 10.6 cm if the velocity of air is not to exceed 6 m/s.

6-14 A cyclone separator is used to remove fine solid particles that are suspended in a gas stream. The mass flow rates at the two outlets and the amount of fly ash collected per year are to be determined.

Assumptions Flow through the separator is steady.

Analysis Since the ash particles cannot be converted into the gas and vice-versa, the mass flow rate of ash into the control volume must equal that going out, and the mass flow rate of flue gas into the control volume must equal that going out. Hence, the mass flow rate of ash leaving is

$$\dot{m}_{\text{ash}} = y_{\text{ash}} \dot{m}_{\text{in}} = (0.001)(10 \text{ kg/s}) = \mathbf{0.01 \text{ kg/s}}$$

The mass flow rate of flue gas leaving the separator is then

$$\dot{m}_{\text{flue gas}} = \dot{m}_{\text{in}} - \dot{m}_{\text{ash}} = 10 - 0.01 = \mathbf{9.99 \text{ kg/s}}$$

The amount of fly ash collected per year is

$$m_{\text{ash}} = \dot{m}_{\text{ash}} \Delta t = (0.01 \text{ kg/s})(365 \times 24 \times 3600 \text{ s/year}) = \mathbf{315,400 \text{ kg/year}}$$

6-15 Air flows through an aircraft engine. The volume flow rate at the inlet and the mass flow rate at the exit are to be determined.

Assumptions 1 Air is an ideal gas. 2 The flow is steady.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis The inlet volume flow rate is

$$\dot{V}_1 = A_1 V_1 = (1 \text{ m}^2)(180 \text{ m/s}) = \mathbf{180 \text{ m}^3/\text{s}}$$

The specific volume at the inlet is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(20 + 273 \text{ K})}{100 \text{ kPa}} = 0.8409 \text{ m}^3/\text{kg}$$

Since the flow is steady, the mass flow rate remains constant during the flow. Then,

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{180 \text{ m}^3/\text{s}}{0.8409 \text{ m}^3/\text{kg}} = \mathbf{214.1 \text{ kg/s}}$$

6-16 A spherical hot-air balloon is considered. The time it takes to inflate the balloon is to be determined.

Assumptions 1 Air is an ideal gas.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis The specific volume of air entering the balloon is

$$\nu = \frac{RT}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(35 + 273 \text{ K})}{120 \text{ kPa}} = 0.7366 \text{ m}^3/\text{kg}$$

The mass flow rate at this entrance is

$$\dot{m} = \frac{A_c V}{\nu} = \frac{\pi D^2}{4} \frac{V}{\nu} = \frac{\pi (1 \text{ m})^2}{4} \frac{2 \text{ m/s}}{0.7366 \text{ m}^3/\text{kg}} = 2.132 \text{ kg/s}$$

The initial mass of the air in the balloon is

$$m_i = \frac{V_i}{\nu} = \frac{\pi D^3}{6\nu} = \frac{\pi (3 \text{ m})^3}{6(0.7366 \text{ m}^3/\text{kg})} = 19.19 \text{ kg}$$

Similarly, the final mass of air in the balloon is

$$m_f = \frac{V_f}{\nu} = \frac{\pi D^3}{6\nu} = \frac{\pi (15 \text{ m})^3}{6(0.7366 \text{ m}^3/\text{kg})} = 2399 \text{ kg}$$

The time it takes to inflate the balloon is determined from

$$\Delta t = \frac{m_f - m_i}{\dot{m}} = \frac{(2399 - 19.19) \text{ kg}}{2.132 \text{ kg/s}} = 1116 \text{ s} = \mathbf{18.6 \text{ min}}$$

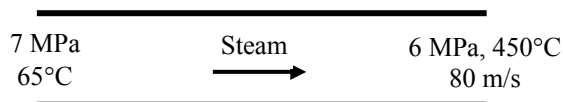
6-17 Water flows through the tubes of a boiler. The velocity and volume flow rate of the water at the inlet are to be determined.

Assumptions Flow through the boiler is steady.

Properties The specific volumes of water at the inlet and exit are (Tables A-6 and A-7)

$$\left. \begin{array}{l} P_1 = 7 \text{ MPa} \\ T_1 = 65^\circ\text{C} \end{array} \right\} \nu_1 = 0.001017 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} P_2 = 6 \text{ MPa} \\ T_2 = 450^\circ\text{C} \end{array} \right\} \nu_2 = 0.05217 \text{ m}^3/\text{kg}$$



Analysis The cross-sectional area of the tube is

$$A_c = \frac{\pi D^2}{4} = \frac{\pi (0.13 \text{ m})^2}{4} = 0.01327 \text{ m}^2$$

The mass flow rate through the tube is same at the inlet and exit. It may be determined from exit data to be

$$\dot{m} = \frac{A_c V_2}{\nu_2} = \frac{(0.01327 \text{ m}^2)(80 \text{ m/s})}{0.05217 \text{ m}^3/\text{kg}} = 20.35 \text{ kg/s}$$

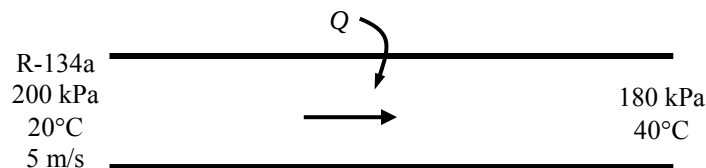
The water velocity at the inlet is then

$$V_1 = \frac{\dot{m} \nu_1}{A_c} = \frac{(20.35 \text{ kg/s})(0.001017 \text{ m}^3/\text{kg})}{0.01327 \text{ m}^2} = \mathbf{1.560 \text{ m/s}}$$

The volumetric flow rate at the inlet is

$$\dot{V}_1 = A_c V_1 = (0.01327 \text{ m}^2)(1.560 \text{ m/s}) = \mathbf{0.0207 \text{ m}^3/\text{s}}$$

6-18 Refrigerant-134a flows through a pipe. Heat is supplied to R-134a. The volume flow rates of air at the inlet and exit, the mass flow rate, and the velocity at the exit are to be determined.



Properties The specific volumes of R-134a at the inlet and exit are (Table A-13)

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 20^\circ\text{C} \end{array} \right\} \nu_1 = 0.1142 \text{ m}^3/\text{kg} \qquad \left. \begin{array}{l} P_1 = 180 \text{ kPa} \\ T_1 = 40^\circ\text{C} \end{array} \right\} \nu_2 = 0.1374 \text{ m}^3/\text{kg}$$

Analysis (a) (b) The volume flow rate at the inlet and the mass flow rate are

$$\dot{V}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28 \text{ m})^2}{4} (5 \text{ m/s}) = \mathbf{0.3079 \text{ m}^3/\text{s}}$$

$$\dot{m} = \frac{1}{\nu_1} A_c V_1 = \frac{1}{\nu_1} \frac{\pi D^2}{4} V_1 = \frac{1}{0.1142 \text{ m}^3/\text{kg}} \frac{\pi (0.28 \text{ m})^2}{4} (5 \text{ m/s}) = \mathbf{2.696 \text{ kg/s}}$$

(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{V}_2 = \dot{m} \nu_2 = (2.696 \text{ kg/s})(0.1374 \text{ m}^3/\text{kg}) = \mathbf{0.3705 \text{ m}^3/\text{s}}$$

$$V_2 = \frac{\dot{V}_2}{A_c} = \frac{0.3705 \text{ m}^3/\text{s}}{\frac{\pi (0.28 \text{ m})^2}{4}} = \mathbf{6.02 \text{ m/s}}$$

Flow Work and Energy Transfer by Mass

6-19C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass.

6-20C Flow energy or flow work is the energy needed to push a fluid into or out of a control volume. Fluids at rest do not possess any flow energy.

6-21C Flowing fluids possess flow energy in addition to the forms of energy a fluid at rest possesses. The total energy of a fluid at rest consists of internal, kinetic, and potential energies. The total energy of a flowing fluid consists of internal, kinetic, potential, and flow energies.

6-22E Steam is leaving a pressure cooker at a specified pressure. The velocity, flow rate, the total and flow energies, and the rate of energy transfer by mass are to be determined.

Assumptions 1 The flow is steady, and the initial start-up period is disregarded. 2 The kinetic and potential energies are negligible, and thus they are not considered. 3 Saturation conditions exist within the cooker at all times so that steam leaves the cooker as a saturated vapor at 30 psia.

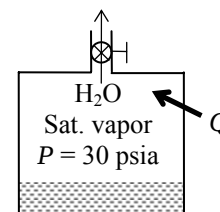
Properties The properties of saturated liquid water and water vapor at 30 psia are $\nu_f = 0.01700 \text{ ft}^3/\text{lbm}$, $\nu_g = 13.749 \text{ ft}^3/\text{lbm}$, $u_g = 1087.8 \text{ Btu/lbm}$, and $h_g = 1164.1 \text{ Btu/lbm}$ (Table A-5E).

Analysis (a) Saturation conditions exist in a pressure cooker at all times after the steady operating conditions are established. Therefore, the liquid has the properties of saturated liquid and the exiting steam has the properties of saturated vapor at the operating pressure. The amount of liquid that has evaporated, the mass flow rate of the exiting steam, and the exit velocity are

$$m = \frac{\Delta V_{\text{liquid}}}{\nu_f} = \frac{0.4 \text{ gal}}{0.01700 \text{ ft}^3/\text{lbm}} \left(\frac{0.13368 \text{ ft}^3}{1 \text{ gal}} \right) = 3.145 \text{ lbm}$$

$$\dot{m} = \frac{m}{\Delta t} = \frac{3.145 \text{ lbm}}{45 \text{ min}} = 0.0699 \text{ lbm/min} = \mathbf{1.165 \times 10^{-3} \text{ lbm/s}}$$

$$V = \frac{\dot{m}}{\rho_g A_c} = \frac{\dot{m} \nu_g}{A_c} = \frac{(1.165 \times 10^{-3} \text{ lbm/s})(13.749 \text{ ft}^3/\text{lbm})}{0.15 \text{ in}^2} \left(\frac{144 \text{ in}^2}{1 \text{ ft}^2} \right) = \mathbf{15.4 \text{ ft/s}}$$



(b) Noting that $h = u + P\nu$ and that the kinetic and potential energies are disregarded, the flow and total energies of the exiting steam are

$$e_{\text{flow}} = P\nu = h - u = 1164.1 - 1087.8 = \mathbf{76.3 \text{ Btu/lbm}}$$

$$\theta = h + ke + pe \cong h = \mathbf{1164.1 \text{ Btu/lbm}}$$

Note that the kinetic energy in this case is $ke = V^2/2 = (15.4 \text{ ft/s})^2 = 237 \text{ ft}^2/\text{s}^2 = 0.0095 \text{ Btu/lbm}$, which is very small compared to enthalpy.

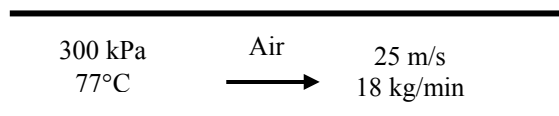
(c) The rate at which energy is leaving the cooker by mass is simply the product of the mass flow rate and the total energy of the exiting steam per unit mass,

$$\dot{E}_{\text{mass}} = \dot{m}\theta = (1.165 \times 10^{-3} \text{ lbm/s})(1164.1 \text{ Btu/lbm}) = \mathbf{1.356 \text{ Btu/s}}$$

Discussion The numerical value of the energy leaving the cooker with steam alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity is the difference between the enthalpies of the exiting vapor and the liquid inside (which is h_{fg}) since it relates directly to the amount of energy supplied to the cooker.

6-23 Air flows steadily in a pipe at a specified state. The diameter of the pipe, the rate of flow energy, and the rate of energy transport by mass are to be determined. Also, the error involved in the determination of energy transport by mass is to be determined.

Properties The properties of air are $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ and $c_p = 1.008 \text{ kJ/kg}\cdot\text{K}$ (at 350 K from Table A-2b)



Analysis (a) The diameter is determined as follows

$$\nu = \frac{RT}{P} = \frac{(0.287 \text{ kJ/kg}\cdot\text{K})(77 + 273 \text{ K})}{(300 \text{ kPa})} = 0.3349 \text{ m}^3/\text{kg}$$

$$A = \frac{\dot{m}\nu}{V} = \frac{(18/60 \text{ kg/s})(0.3349 \text{ m}^3/\text{kg})}{25 \text{ m/s}} = 0.004018 \text{ m}^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.004018 \text{ m}^2)}{\pi}} = \mathbf{0.0715 \text{ m}}$$

(b) The rate of flow energy is determined from

$$\dot{W}_{\text{flow}} = \dot{m}P\nu = (18/60 \text{ kg/s})(300 \text{ kPa})(0.3349 \text{ m}^3/\text{kg}) = \mathbf{30.14 \text{ kW}}$$

(c) The rate of energy transport by mass is

$$\begin{aligned} \dot{E}_{\text{mass}} &= \dot{m}(h + ke) = \dot{m}\left(c_p T + \frac{1}{2}V^2\right) \\ &= (18/60 \text{ kg/s})\left[(1.008 \text{ kJ/kg}\cdot\text{K})(77 + 273 \text{ K}) + \frac{1}{2}(25 \text{ m/s})^2\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] \\ &= \mathbf{105.94 \text{ kW}} \end{aligned}$$

(d) If we neglect kinetic energy in the calculation of energy transport by mass

$$\dot{E}_{\text{mass}} = \dot{m}h = \dot{m}c_p T = (18/60 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot\text{K})(77 + 273 \text{ K}) = 105.84 \text{ kW}$$

Therefore, the error involved if neglect the kinetic energy is only **0.09%**.

6-24E A water pump increases water pressure. The flow work required by the pump is to be determined.

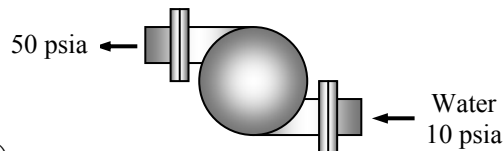
Assumptions 1 Flow through the pump is steady. 2 The state of water at the pump inlet is saturated liquid. 3 The specific volume remains constant.

Properties The specific volume of saturated liquid water at 10 psia is

$$\nu = \nu_{f@10 \text{ psia}} = 0.01659 \text{ ft}^3/\text{lbm} \quad (\text{Table A-5E})$$

Then the flow work relation gives

$$\begin{aligned} w_{\text{flow}} &= P_2 \nu_2 - P_1 \nu_1 = \nu(P_2 - P_1) \\ &= (0.01659 \text{ ft}^3/\text{lbm})(50 - 10) \text{ psia} \left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3} \right) \\ &= \mathbf{0.1228 \text{ Btu/lbm}} \end{aligned}$$

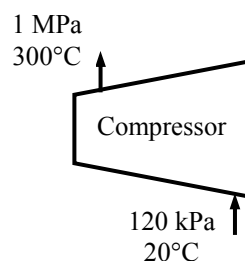


6-25 An air compressor compresses air. The flow work required by the compressor is to be determined.

Assumptions 1 Flow through the compressor is steady. 2 Air is an ideal gas.

Properties Combining the flow work expression with the ideal gas equation of state gives

$$\begin{aligned} w_{\text{flow}} &= P_2 \nu_2 - P_1 \nu_1 \\ &= R(T_2 - T_1) \\ &= (0.287 \text{ kJ/kg} \cdot \text{K})(300 - 20) \text{ K} \\ &= \mathbf{80.36 \text{ kJ/kg}} \end{aligned}$$



Steady Flow Energy Balance: Nozzles and Diffusers

6-26C A steady-flow system involves no changes with time anywhere within the system or at the system boundaries

6-27C No.

6-28C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

6-29C The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

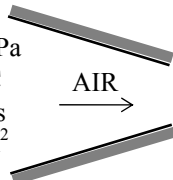
6-30C Heat transfer to the fluid as it flows through a nozzle is desirable since it will probably increase the kinetic energy of the fluid. Heat transfer from the fluid will decrease the exit velocity.

6-31 Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Air is an ideal gas with constant specific heats. 3 Potential energy changes are negligible. 4 The device is adiabatic and thus heat transfer is negligible. 5 There are no work interactions.

Properties The gas constant of air is 0.287 kPa·m³/kg·K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is $c_p = 1.02$ kJ/kg·°C (Table A-2).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be



$$\begin{array}{l} P_1 = 300 \text{ kPa} \\ T_1 = 200^\circ\text{C} \\ V_1 = 30 \text{ m/s} \\ A_1 = 80 \text{ cm}^2 \end{array} \quad \xrightarrow{\text{AIR}} \quad \begin{array}{l} P_2 = 100 \text{ kPa} \\ V_2 = 180 \text{ m/s} \end{array}$$

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(473 \text{ K})}{300 \text{ kPa}} = 0.4525 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 = \frac{1}{0.4525 \text{ m}^3/\text{kg}} (0.008 \text{ m}^2)(30 \text{ m/s}) = \mathbf{0.5304 \text{ kg/s}}$$

(b) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \equiv \dot{W} \equiv \Delta p e \equiv 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,\text{ave}}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Substituting,
$$0 = (1.02 \text{ kJ/kg} \cdot \text{K})(T_2 - 200^\circ\text{C}) + \frac{(180 \text{ m/s})^2 - (30 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right)$$

It yields $T_2 = \mathbf{184.6^\circ\text{C}}$

(c) The specific volume of air at the nozzle exit is

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(184.6 + 273 \text{ K})}{100 \text{ kPa}} = 1.313 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}} A_2 (180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = \mathbf{38.7 \text{ cm}^2}$$

6-32 EES Problem 6-31 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 cm^2 to 150 cm^2 is to be investigated, and the final results are to be plotted against the inlet area.

Analysis The problem is solved using EES, and the solution is given below.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
  If 'Air' = WorkFluid$ then
    HCal:=ENTHALPY('Air',T=Tx) "Ideal gas equ."
  else
    HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
  endif
end HCal

"System: control volume for the nozzle"
"Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 200 [C]
P[1] = 300 [kPa]
Vel[1] = 30 [m/s]
P[2] = 100 [kPa]
Vel[2] = 180 [m/s]
A[1]=80 [cm^2]
Am[1]=A[1]*convert(cm^2,m^2)

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

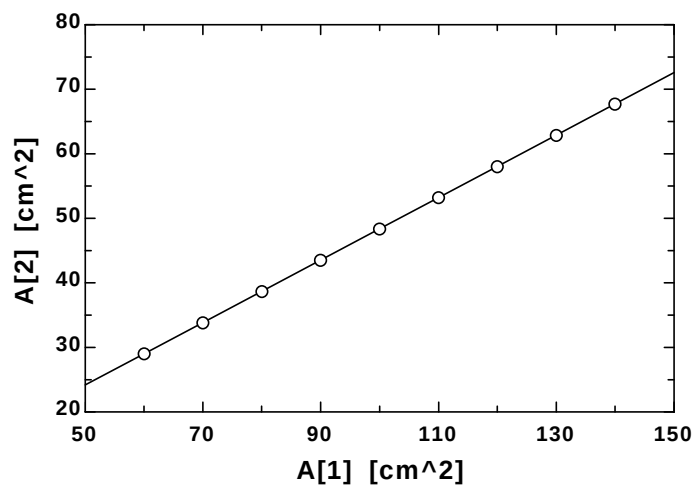
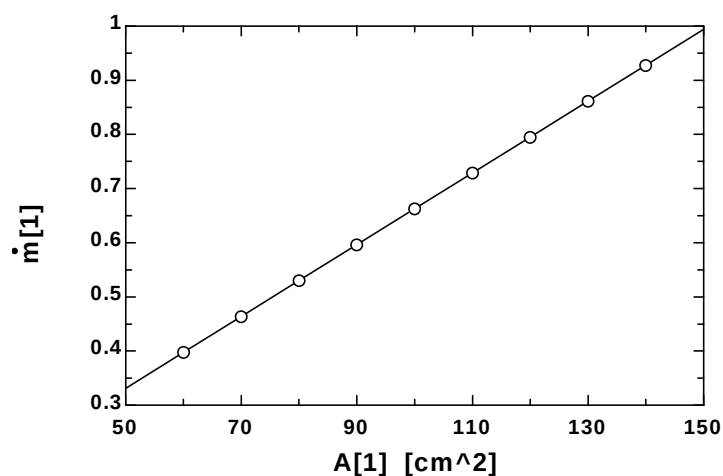
"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"Conservation of mass: "
m_dot[1]= m_dot[2]
"Mass flow rate"
m_dot[1]=Am[1]*Vel[1]/v[1]
m_dot[2]= Am[2]*Vel[2]/v[2]

"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)

"Definition"
A_ratio=A[1]/A[2]
A[2]=Am[2]*convert(m^2,cm^2)
```

A_1 [cm ²]	A_2 [cm ²]	m_1	T_2
50	24.19	0.3314	184.6
60	29.02	0.3976	184.6
70	33.86	0.4639	184.6
80	38.7	0.5302	184.6
90	43.53	0.5964	184.6
100	48.37	0.6627	184.6
110	53.21	0.729	184.6
120	58.04	0.7952	184.6
130	62.88	0.8615	184.6
140	67.72	0.9278	184.6
150	72.56	0.9941	184.6



6-33E Air is accelerated in an adiabatic nozzle. The velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The nozzle is adiabatic.

Properties The specific heat of air at the average temperature of $(700+645)/2=672.5^\circ\text{F}$ is $c_p = 0.253$ Btu/lbm·R (Table A-2Eb).

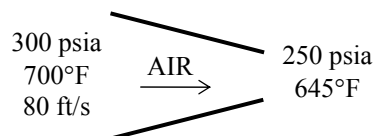
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2 / 2) = \dot{m}(h_2 + V_2^2 / 2)$$

$$h_1 + V_1^2 / 2 = h_2 + V_2^2 / 2$$



Solving for exit velocity,

$$\begin{aligned} V_2 &= \left[V_1^2 + 2(h_1 - h_2) \right]^{0.5} = \left[V_1^2 + 2c_p(T_1 - T_2) \right]^{0.5} \\ &= \left[(80 \text{ ft/s})^2 + 2(0.253 \text{ Btu/lbm} \cdot \text{R})(700 - 645) \text{R} \left(\frac{25,037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right) \right]^{0.5} \\ &= \mathbf{838.6 \text{ ft/s}} \end{aligned}$$

6-34 Air is decelerated in an adiabatic diffuser. The velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The diffuser is adiabatic.

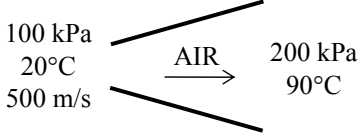
Properties The specific heat of air at the average temperature of $(20+90)/2=55^\circ\text{C}=328\text{ K}$ is $c_p = 1.007\text{ kJ/kg}\cdot\text{K}$ (Table A-2b).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$


Solving for exit velocity,

$$V_2 = [V_1^2 + 2(h_1 - h_2)]^{0.5} = [V_1^2 + 2c_p(T_1 - T_2)]^{0.5}$$

$$= \left[(500\text{ m/s})^2 + 2(1.007\text{ kJ/kg}\cdot\text{K})(20 - 90)\text{K} \left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}} \right) \right]^{0.5}$$

$$= \mathbf{330.2\text{ m/s}}$$

6-35 Steam is accelerated in a nozzle from a velocity of 80 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle are to be determined.

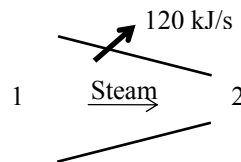
Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

Properties From the steam tables (Table A-6)

$$\left. \begin{array}{l} P_1 = 5 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.057838 \text{ m}^3/\text{kg} \\ h_1 = 3196.7 \text{ kJ/kg} \end{array}$$

and

$$\left. \begin{array}{l} P_2 = 2 \text{ MPa} \\ T_2 = 300^\circ\text{C} \end{array} \right\} \begin{array}{l} v_2 = 0.12551 \text{ m}^3/\text{kg} \\ h_2 = 3024.2 \text{ kJ/kg} \end{array}$$



Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The mass flow rate of steam is

$$\dot{m} = \frac{1}{v_1} V_1 A_1 = \frac{1}{0.057838 \text{ m}^3/\text{kg}} (80 \text{ m/s})(50 \times 10^{-4} \text{ m}^2) = \mathbf{6.92 \text{ kg/s}}$$

(b) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta p e \cong 0)$$

$$-\dot{Q}_{\text{out}} = \dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting, the exit velocity of the steam is determined to be

$$-120 \text{ kJ/s} = (6.916 \text{ kg/s}) \left(3024.2 - 3196.7 + \frac{V_2^2 - (80 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right)$$

It yields $V_2 = \mathbf{562.7 \text{ m/s}}$

(c) The exit area of the nozzle is determined from

$$\dot{m} = \frac{1}{v_2} V_2 A_2 \longrightarrow A_2 = \frac{\dot{m} v_2}{V_2} = \frac{(6.916 \text{ kg/s})(0.12551 \text{ m}^3/\text{kg})}{562.7 \text{ m/s}} = \mathbf{15.42 \times 10^{-4} \text{ m}^2}$$

6-36 CD EES Steam is accelerated in a nozzle from a velocity of 40 m/s to 300 m/s. The exit temperature and the ratio of the inlet-to-exit area of the nozzle are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Potential energy changes are negligible. 3 There are no work interactions. 4 The device is adiabatic and thus heat transfer is negligible.

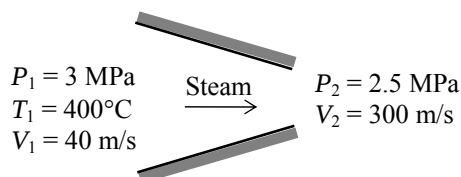
Properties From the steam tables (Table A-6),

$$\left. \begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_1 = 0.09938 \text{ m}^3/\text{kg} \\ h_1 = 3231.7 \text{ kJ/kg} \end{array}$$

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$



$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta p e \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 3231.7 \text{ kJ/kg} - \frac{(300 \text{ m/s})^2 - (40 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 3187.5 \text{ kJ/kg}$$

$$\text{Thus, } \left. \begin{array}{l} P_2 = 2.5 \text{ MPa} \\ h_2 = 3187.5 \text{ kJ/kg} \end{array} \right\} \begin{array}{l} T_2 = \mathbf{376.6^\circ\text{C}} \\ \nu_2 = 0.11533 \text{ m}^3/\text{kg} \end{array}$$

(b) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\nu_2} A_2 V_2 = \frac{1}{\nu_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\nu_1}{\nu_2} \frac{V_2}{V_1} = \frac{(0.09938 \text{ m}^3/\text{kg})(300 \text{ m/s})}{(0.11533 \text{ m}^3/\text{kg})(40 \text{ m/s})} = \mathbf{6.46}$$

6-37 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The enthalpy of air at the inlet temperature of 400 K is $h_1 = 400.98 \text{ kJ/kg}$ (Table A-21).

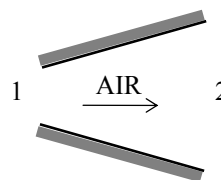
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad ,$$



or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 400.98 \text{ kJ/kg} - \frac{(30 \text{ m/s})^2 - (230 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 426.98 \text{ kJ/kg}$$

From Table A-21, $T_2 = 425.6 \text{ K}$

(b) The specific volume of air at the diffuser exit is

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow A_2 = \frac{\dot{m} \nu_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = 0.0678 \text{ m}^2$$

6-38E Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The enthalpy of air at the inlet temperature of 20°F is $h_1 = 114.69$ Btu/lbm (Table A-21E).

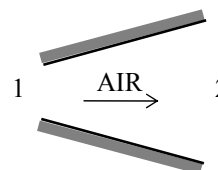
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta pe \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2},$$



or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 114.69 \text{ Btu/lbm} - \frac{0 - (600 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = 121.88 \text{ Btu/lbm}$$

From Table A-21E,

$$T_2 = \mathbf{510.0 \text{ R}}$$

(b) The exit velocity of air is determined from the conservation of mass relation,

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow \frac{1}{RT_2/P_2} A_2 V_2 = \frac{1}{RT_1/P_1} A_1 V_1$$

Thus,

$$V_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2} V_1 = \frac{1}{5} \frac{(510 \text{ R})(13 \text{ psia})}{(480 \text{ R})(14.5 \text{ psia})} (600 \text{ ft/s}) = \mathbf{114.3 \text{ ft/s}}$$

6-39 CO₂ gas is accelerated in a nozzle to 450 m/s. The inlet velocity and the exit temperature are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 CO₂ is an ideal gas with variable specific heats. 3 Potential energy changes are negligible. 4 The device is adiabatic and thus heat transfer is negligible. 5 There are no work interactions.

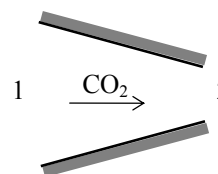
Properties The gas constant and molar mass of CO₂ are 0.1889 kPa·m³/kg·K and 44 kg/kmol (Table A-1). The enthalpy of CO₂ at 500°C is $\bar{h}_1 = 30,797$ kJ/kmol (from CO₂ ideal gas tables -not available in this text).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume is determined to be

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(773 \text{ K})}{1000 \text{ kPa}} = 0.146 \text{ m}^3/\text{kg}$$

Thus,

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 \longrightarrow V_1 = \frac{\dot{m} \nu_1}{A_1} = \frac{(6000/3600 \text{ kg/s})(0.146 \text{ m}^3/\text{kg})}{40 \times 10^{-4} \text{ m}^2} = \mathbf{60.8 \text{ m/s}}$$



(b) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta p \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$\begin{aligned} \bar{h}_2 &= \bar{h}_1 - \frac{V_2^2 - V_1^2}{2} M \\ &= 30,797 \text{ kJ/kmol} - \frac{(450 \text{ m/s})^2 - (60.8 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) (44 \text{ kg/kmol}) \\ &= 26,423 \text{ kJ/kmol} \end{aligned}$$

Then the exit temperature of CO₂ from the ideal gas tables is obtained to be $T_2 = \mathbf{685.8 \text{ K}}$

Alternative Solution Using constant specific heats for CO₂ at an anticipated average temperature of 700 K (Table A-2b), from the energy balance we obtain

$$\begin{aligned} 0 &= c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} \\ 0 &= (1.126 \text{ kJ/kg} \cdot \text{K})(T_2 - 500)^\circ\text{C} + \frac{(450 \text{ m/s})^2 - (60.8 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \\ T_2 &= 411.7^\circ\text{C} = 684.7 \text{ K} \end{aligned}$$

The result is practically identical to the result obtained earlier.

6-40 R-134a is accelerated in a nozzle from a velocity of 20 m/s. The exit velocity of the refrigerant and the ratio of the inlet-to-exit area of the nozzle are to be determined.

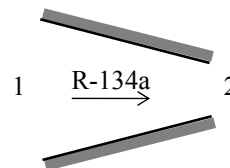
Assumptions 1 This is a steady-flow process since there is no change with time. 2 Potential energy changes are negligible. 3 There are no work interactions. 4 The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Table A-13)

$$\left. \begin{array}{l} P_1 = 700 \text{ kPa} \\ T_1 = 120^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.043358 \text{ m}^3/\text{kg} \\ h_1 = 358.90 \text{ kJ/kg} \end{array}$$

and

$$\left. \begin{array}{l} P_2 = 400 \text{ kPa} \\ T_2 = 30^\circ\text{C} \end{array} \right\} \begin{array}{l} v_2 = 0.056796 \text{ m}^3/\text{kg} \\ h_2 = 275.07 \text{ kJ/kg} \end{array}$$



Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta p e \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$0 = (275.07 - 358.90) \text{ kJ/kg} + \frac{V_2^2 - (20 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right)$$

It yields $V_2 = 409.9 \text{ m/s}$

(b) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2} \frac{V_2}{V_1} = \frac{(0.043358 \text{ m}^3/\text{kg})(409.9 \text{ m/s})}{(0.056796 \text{ m}^3/\text{kg})(20 \text{ m/s})} = \mathbf{15.65}$$

6-41 Nitrogen is decelerated in a diffuser from 200 m/s to a lower velocity. The exit velocity of nitrogen and the ratio of the inlet-to-exit area are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The molar mass of nitrogen is $M = 28 \text{ kg/kmol}$ (Table A-1). The enthalpies are (from nitrogen ideal gas tables-not available in this text)

$$T_1 = 7^\circ\text{C} = 280 \text{ K} \rightarrow \bar{h}_1 = 8141 \text{ kJ/kmol}$$

$$T_2 = 22^\circ\text{C} = 295 \text{ K} \rightarrow \bar{h}_2 = 8580 \text{ kJ/kmol}$$

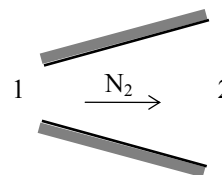
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = \frac{\bar{h}_2 - \bar{h}_1}{M} + \frac{V_2^2 - V_1^2}{2}$$



Substituting,

$$0 = \frac{(8580 - 8141) \text{ kJ/kmol}}{28 \text{ kg/kmol}} + \frac{V_2^2 - (200 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right)$$

It yields

$$V_2 = 93.0 \text{ m/s}$$

(b) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\nu_2} A_2 V_2 = \frac{1}{\nu_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\nu_1}{\nu_2} \frac{V_2}{V_1} = \left(\frac{RT_1/P_1}{RT_2/P_2} \right) \frac{V_2}{V_1}$$

or,

$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2} \right) \frac{V_2}{V_1} = \frac{(280 \text{ K}/60 \text{ kPa})(93.0 \text{ m/s})}{(295 \text{ K}/85 \text{ kPa})(200 \text{ m/s})} = 0.625$$

Alternative Solution Using constant specific heats for N_2 at room temperature (Table A-2a), from the energy balance we obtain

$$0 = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (1.039 \text{ kJ/kg} \cdot \text{K})(22 - 7)^\circ\text{C} + \frac{V_2^2 - (200 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right)$$

$$V_2 = 94.0 \text{ m/s}$$

The result is practically identical to the result obtained earlier.

6-42 EES Problem 6-41 is reconsidered. The effect of the inlet velocity on the exit velocity and the ratio of the inlet-to-exit area as the inlet velocity varies from 180 m/s to 260 m/s is to be investigated. The final results are to be plotted against the inlet velocity.

Analysis The problem is solved using EES, and the solution is given below.

```
Function HCal(WorkFluid$, Tx, Px)
```

```
"Function to calculate the enthalpy of an ideal gas or real gas"
```

```
  If 'N2' = WorkFluid$ then
```

```
    HCal:=ENTHALPY(WorkFluid$,T=Tx) "Ideal gas equ."
```

```
  else
```

```
    HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px) "Real gas equ."
```

```
  endif
```

```
end HCal
```

```
"System: control volume for the nozzle"
```

```
"Property relation: Nitrogen is an ideal gas"
```

```
"Process: Steady state, steady flow, adiabatic, no work"
```

```
"Knowns"
```

```
WorkFluid$ = 'N2'
```

```
T[1] = 7 [C]
```

```
P[1] = 60 [kPa]
```

```
{Vel[1] = 200 [m/s]}
```

```
P[2] = 85 [kPa]
```

```
T[2] = 22 [C]
```

```
"Property Data - since the Enthalpy function has different parameters  
for ideal gas and real fluids, a function was used to determine h."
```

```
h[1]=HCal(WorkFluid$,T[1],P[1])
```

```
h[2]=HCal(WorkFluid$,T[2],P[2])
```

```
"The Volume function has the same form for an ideal gas as for a real fluid."
```

```
v[1]=volume(workFluid$,T=T[1],p=P[1])
```

```
v[2]=volume(WorkFluid$,T=T[2],p=P[2])
```

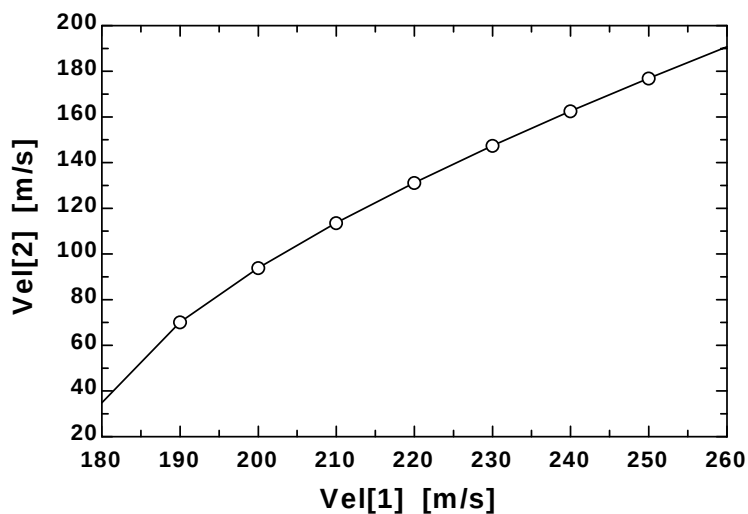
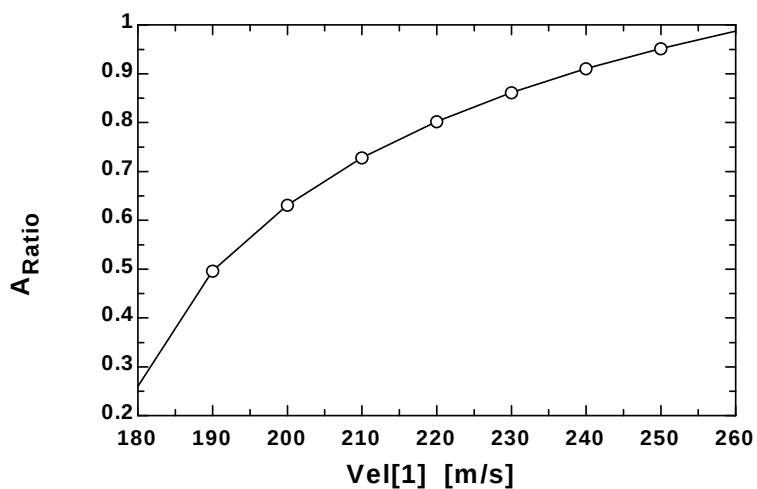
```
"From the definition of mass flow rate, m_dot = A*Vel/v and conservation of mass the area ratio  
A_Ratio = A_1/A_2 is:"
```

```
A_Ratio*Vel[1]/v[1] =Vel[2]/v[2]
```

```
"Conservation of Energy - SSSF energy balance"
```

```
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)
```

A_{Ratio}	Vel_1 [m/s]	Vel_2 [m/s]
0.2603	180	34.84
0.4961	190	70.1
0.6312	200	93.88
0.7276	210	113.6
0.8019	220	131.2
0.8615	230	147.4
0.9106	240	162.5
0.9518	250	177
0.9869	260	190.8



6-43 R-134a is decelerated in a diffuser from a velocity of 120 m/s. The exit velocity of R-134a and the mass flow rate of the R-134a are to be determined.

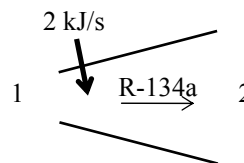
Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

Properties From the R-134a tables (Tables A-11 through A-13)

$$\left. \begin{array}{l} P_1 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} v_1 = 0.025621 \text{ m}^3/\text{kg} \\ h_1 = 267.29 \text{ kJ/kg} \end{array}$$

and

$$\left. \begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 40^\circ\text{C} \end{array} \right\} \begin{array}{l} v_2 = 0.023375 \text{ m}^3/\text{kg} \\ h_2 = 274.17 \text{ kJ/kg} \end{array}$$



Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity of R-134a is determined from the steady-flow mass balance to be

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow V_2 = \frac{v_2}{v_1} \frac{A_1}{A_2} V_1 = \frac{1}{1.8} \frac{(0.023375 \text{ m}^3/\text{kg})}{(0.025621 \text{ m}^3/\text{kg})} (120 \text{ m/s}) = \mathbf{60.8 \text{ m/s}}$$

(b) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta p e \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting, the mass flow rate of the refrigerant is determined to be

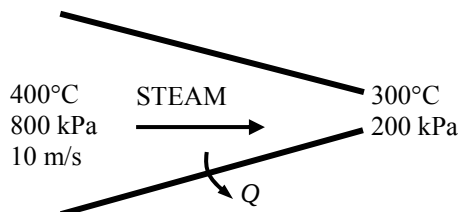
$$2 \text{ kJ/s} = \dot{m} \left((274.17 - 267.29) \text{ kJ/kg} + \frac{(60.8 \text{ m/s})^2 - (120 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right)$$

It yields $\dot{m} = \mathbf{1.308 \text{ kg/s}}$

6-44 Heat is lost from the steam flowing in a nozzle. The velocity and the volume flow rate at the nozzle exit are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions.

Analysis We take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as



Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right) + \dot{Q}_{\text{out}} \quad \text{since } \dot{W} \cong \Delta \text{pe} \cong 0$$

or

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + \frac{\dot{Q}_{\text{out}}}{\dot{m}}$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left. \begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.38429 \text{ m}^3/\text{kg} \\ h_1 = 3267.7 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 300^\circ\text{C} \end{array} \right\} \begin{array}{l} v_2 = 1.31623 \text{ m}^3/\text{kg} \\ h_2 = 3072.1 \text{ kJ/kg} \end{array}$$

The mass flow rate of the steam is

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.38429 \text{ m}^3/\text{kg}} (0.08 \text{ m}^2) (10 \text{ m/s}) = 2.082 \text{ kg/s}$$

Substituting,

$$3267.7 \text{ kJ/kg} + \frac{(10 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 3072.1 \text{ kJ/kg} + \frac{V_2^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) + \frac{25 \text{ kJ/s}}{2.082 \text{ kg/s}}$$

$$\longrightarrow V_2 = \mathbf{606 \text{ m/s}}$$

The volume flow rate at the exit of the nozzle is

$$\dot{V}_2 = \dot{m} v_2 = (2.082 \text{ kg/s})(1.31623 \text{ m}^3/\text{kg}) = \mathbf{2.74 \text{ m}^3/\text{s}}$$

Turbines and Compressors

6-45C Yes.

6-46C The volume flow rate at the compressor inlet will be greater than that at the compressor exit.

6-47C Yes. Because energy (in the form of shaft work) is being added to the air.

6-48C No.

6-49 Air is expanded in a turbine. The mass flow rate and outlet area are to be determined.

Assumptions 1 Air is an ideal gas. 2 The flow is steady.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis The specific volumes of air at the inlet and outlet are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(600 + 273 \text{ K})}{1000 \text{ kPa}} = 0.2506 \text{ m}^3/\text{kg}$$

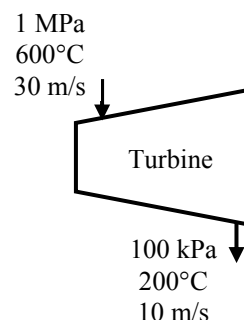
$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(200 + 273 \text{ K})}{100 \text{ kPa}} = 1.3575 \text{ m}^3/\text{kg}$$

The mass flow rate is

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{(0.1 \text{ m}^2)(30 \text{ m/s})}{0.2506 \text{ m}^3/\text{kg}} = \mathbf{11.97 \text{ kg/s}}$$

The outlet area is

$$A_2 = \frac{\dot{m} \nu_2}{V_2} = \frac{(11.97 \text{ kg/s})(1.3575 \text{ m}^3/\text{kg})}{10 \text{ m/s}} = \mathbf{1.605 \text{ m}^2}$$



6-50E Air is expanded in a gas turbine. The inlet and outlet mass flow rates are to be determined.

Assumptions 1 Air is an ideal gas. 2 The flow is steady.

Properties The gas constant of air is $R = 0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$ (Table A-1E).

Analysis The specific volumes of air at the inlet and outlet are

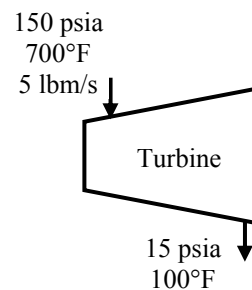
$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(700 + 460 \text{ R})}{150 \text{ psia}} = 2.864 \text{ ft}^3/\text{lbm}$$

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(100 + 460 \text{ R})}{15 \text{ psia}} = 13.83 \text{ ft}^3/\text{lbm}$$

The volume flow rates at the inlet and exit are then

$$\dot{V}_1 = \dot{m} \nu_1 = (5 \text{ lbm/s})(2.864 \text{ ft}^3/\text{lbm}) = \mathbf{14.32 \text{ ft}^3/\text{s}}$$

$$\dot{V}_2 = \dot{m} \nu_2 = (5 \text{ lbm/s})(13.83 \text{ ft}^3/\text{lbm}) = \mathbf{69.15 \text{ ft}^3/\text{s}}$$



6-51 Air is compressed at a rate of 10 L/s by a compressor. The work required per unit mass and the power required are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of $(20+300)/2=160^\circ\text{C}=433\text{ K}$ is $c_p = 1.018\text{ kJ/kg}\cdot\text{K}$ (Table A-2b). The gas constant of air is $R = 0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

$$w_{\text{in}} = c_p(T_2 - T_1) = (1.018\text{ kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{285.0\text{ kJ/kg}}$$

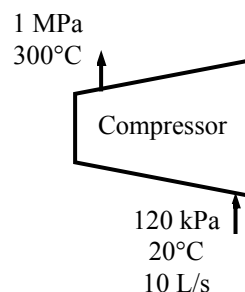
(b) The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273\text{ K})}{120\text{ kPa}} = 0.7008\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.010\text{ m}^3/\text{s}}{0.7008\text{ m}^3/\text{kg}} = 0.01427\text{ kg/s}$$

Then the power input is determined from the energy balance equation to be

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = (0.01427\text{ kg/s})(1.018\text{ kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{4.068\text{ kW}}$$



6-52 Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left. \begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 450^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.029782 \text{ m}^3/\text{kg} \\ h_1 = 3242.4 \text{ kJ/kg} \end{array}$$

and

$$\left. \begin{array}{l} P_2 = 10 \text{ kPa} \\ x_2 = 0.92 \end{array} \right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.92 \times 2392.1 = 2392.5 \text{ kJ/kg}$$

Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(50 \text{ m/s})^2 - (80 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = -1.95 \text{ kJ/kg}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta pe \cong 0)$$

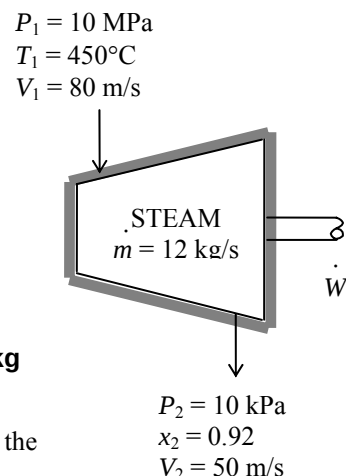
$$\dot{W}_{\text{out}} = -\dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{\text{out}} = -(12 \text{ kg/s})(2392.5 - 3242.4 - 1.95) \text{ kJ/kg} = \mathbf{10.2 \text{ MW}}$$

(c) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{v_1} A_1 V_1 \longrightarrow A_1 = \frac{\dot{m} v_1}{V_1} = \frac{(12 \text{ kg/s})(0.029782 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = \mathbf{0.00447 \text{ m}^2}$$



6-53 EES Problem 6-52 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns"

T[1] = 450 [C]
P[1] = 10000 [kPa]
Vel[1] = 80 [m/s]
P[2] = 10 [kPa]
X_2=0.92
Vel[2] = 50 [m/s]
m_dot[1]=12 [kg/s]
Fluid\$='Steam_IAPWS'

"Property Data"

h[1]=enthalpy(Fluid\$,T=T[1],P=P[1])
h[2]=enthalpy(Fluid\$,P=P[2],x=x_2)
T[2]=temperature(Fluid\$,P=P[2],x=x_2)
v[1]=volume(Fluid\$,T=T[1],p=P[1])
v[2]=volume(Fluid\$,P=P[2],x=x_2)

"Conservation of mass: "

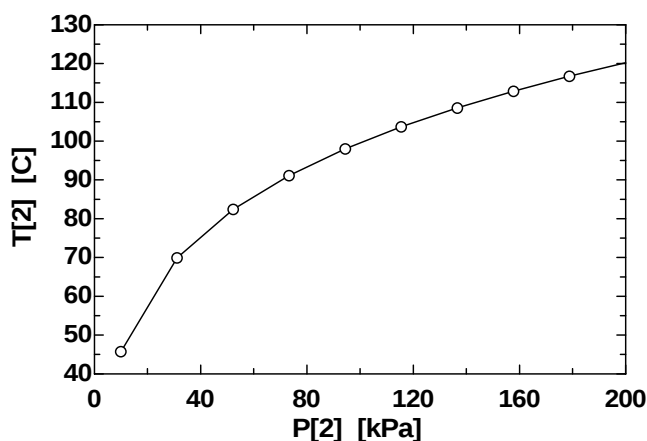
m_dot[1]= m_dot[2]

"Mass flow rate"

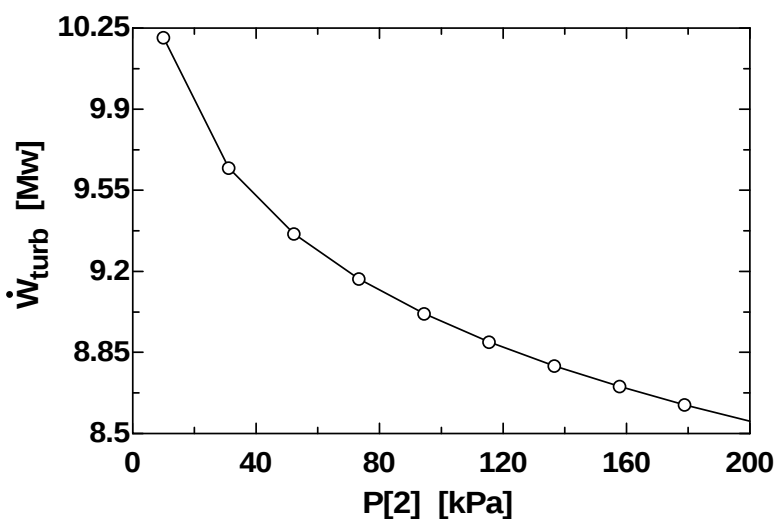
m_dot[1]=A[1]*Vel[1]/v[1]
m_dot[2]= A[2]*Vel[2]/v[2]

"Conservation of Energy - Steady Flow energy balance"

m_dot[1]*(h[1]+Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) =
m_dot[2]*(h[2]+Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+W_dot_turb*convert(MW,kJ/s)
DELTAke=Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)-Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)



P ₂ [kPa]	W _{turb} [MW]	T ₂ [C]
10	10.22	45.81
31.11	9.66	69.93
52.22	9.377	82.4
73.33	9.183	91.16
94.44	9.033	98.02
115.6	8.912	103.7
136.7	8.809	108.6
157.8	8.719	112.9
178.9	8.641	116.7
200	8.57	120.2



6-54 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left. \begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500^\circ\text{C} \end{array} \right\} h_1 = 3375.1 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 10 \text{ kPa} \\ x_2 = 0.90 \end{array} \right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.90 \times 2392.1 = 2344.7 \text{ kJ/kg}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

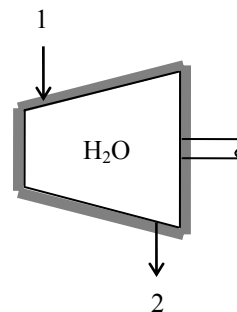
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}(h_2 - h_1)$$

Substituting, the required mass flow rate of the steam is determined to be

$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852 \text{ kg/s}}$$



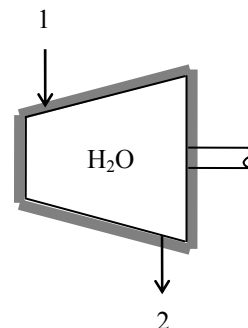
6-55E Steam expands in a turbine. The rate of heat loss from the steam for a power output of 4 MW is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the steam tables (Tables A-4E through 6E)

$$\left. \begin{array}{l} P_1 = 1000 \text{ psia} \\ T_1 = 900^\circ\text{F} \end{array} \right\} h_1 = 1448.6 \text{ Btu/lbm}$$

$$\left. \begin{array}{l} P_2 = 5 \text{ psia} \\ \text{sat. vapor} \end{array} \right\} h_2 = 1130.7 \text{ Btu/lbm}$$



Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{out}} = -\dot{m}(h_2 - h_1) - \dot{W}_{\text{out}}$$

Substituting,

$$\dot{Q}_{\text{out}} = -(45000/3600 \text{ lbm/s})(1130.7 - 1448.6) \text{ Btu/lbm} - 4000 \text{ kJ/s} \left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}} \right) = \mathbf{182.0 \text{ Btu/s}}$$

6-56 Steam expands in a turbine. The exit temperature of the steam for a power output of 2 MW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left. \begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500^\circ\text{C} \end{array} \right\} h_1 = 3399.5 \text{ kJ/kg}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

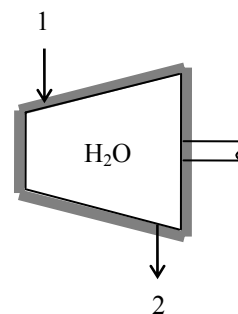
$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

Substituting,

$$\begin{aligned} 2500 \text{ kJ/s} &= (3 \text{ kg/s})(3399.5 - h_2) \text{ kJ/kg} \\ h_2 &= 2566.2 \text{ kJ/kg} \end{aligned}$$

Then the exit temperature becomes

$$\left. \begin{array}{l} P_2 = 20 \text{ kPa} \\ h_2 = 2566.2 \text{ kJ/kg} \end{array} \right\} T_2 = \mathbf{60.1^\circ\text{C}}$$



6-57 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

Properties The gas constant of Ar is $R = 0.2081 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$. The constant pressure specific heat of Ar is $c_p = 0.5203 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2a)

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of argon and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.2081 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(723 \text{ K})}{900 \text{ kPa}} = 0.167 \text{ m}^3/\text{kg}$$

Thus,

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.167 \text{ m}^3/\text{kg}} (0.006 \text{ m}^2)(80 \text{ m/s}) = 2.874 \text{ kg/s}$$

We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta p e \cong 0)$$

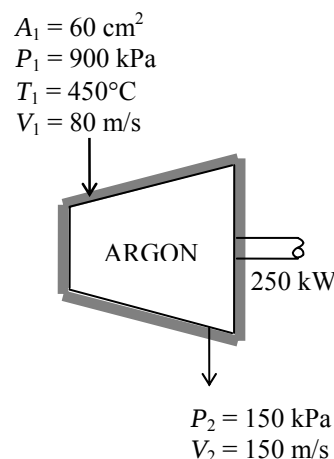
$$\dot{W}_{out} = -\dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting,

$$250 \text{ kJ/s} = -(2.874 \text{ kg/s}) \left[(0.5203 \text{ kJ/kg} \cdot ^\circ\text{C})(T_2 - 450^\circ\text{C}) + \frac{(150 \text{ m/s})^2 - (80 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right]$$

It yields

$$T_2 = 267.3^\circ\text{C}$$



6-58 Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is $c_p = 5.1926 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

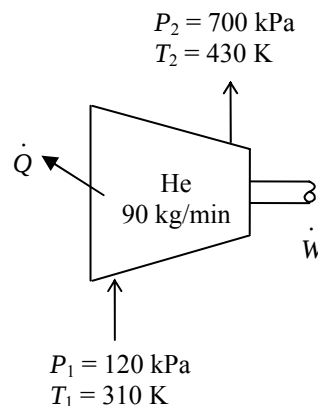
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{Q}_{\text{out}} + \dot{m}c_p(T_2 - T_1) \\ &= (90/60 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg}\cdot\text{K})(430 - 310)\text{K} \\ &= \mathbf{965 \text{ kW}} \end{aligned}$$



6-59 CO₂ is compressed by a compressor. The volume flow rate of CO₂ at the compressor inlet and the power input to the compressor are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 Helium is an ideal gas with variable specific heats. 4 The device is adiabatic and thus heat transfer is negligible.

Properties The gas constant of CO₂ is $R = 0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$, and its molar mass is $M = 44 \text{ kg/kmol}$ (Table A-1). The inlet and exit enthalpies of CO₂ are (from CO₂ ideal gas tables –not available in this text)

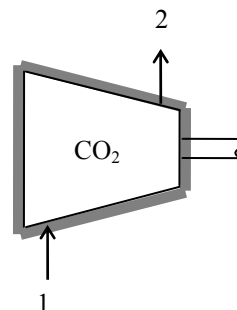
$$T_1 = 300 \text{ K} \rightarrow \bar{h}_1 = 9,431 \text{ kJ/kmol}$$

$$T_2 = 450 \text{ K} \rightarrow \bar{h}_2 = 15,483 \text{ kJ/kmol}$$

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its volume flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})}{100 \text{ kPa}} = 0.5667 \text{ m}^3/\text{kg}$$

$$\dot{V} = \dot{m}\nu_1 = (0.5 \text{ kg/s})(0.5667 \text{ m}^3/\text{kg}) = \mathbf{0.283 \text{ m}^3/\text{s}}$$



(b) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

Substituting

$$\dot{W}_{\text{in}} = \frac{(0.5 \text{ kg/s})(15,483 - 9,431 \text{ kJ/kmol})}{44 \text{ kg/kmol}} = \mathbf{68.8 \text{ kW}}$$

Alternative Solution Using constant specific heats for CO₂ at room temperature (Table A-2a), from the energy balance we obtain

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = (0.5 \text{ kg/s})(1.039 \text{ kJ/kg} \cdot \text{K})(450 - 300)\text{K} = 63.5 \text{ kW}$$

The result is close to the result obtained earlier.

6-60 Air is expanded in an adiabatic turbine. The mass flow rate of the air and the power produced are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The turbine is well-insulated, and thus there is no heat transfer. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of $(500+150)/2=325^\circ\text{C}=598\text{ K}$ is $c_p = 1.051\text{ kJ/kg}\cdot\text{K}$ (Table A-2b). The gas constant of air is $R = 0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).

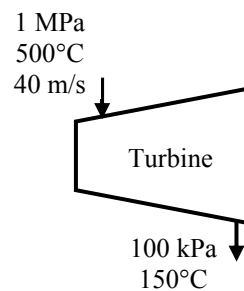
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right) + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m} \left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} \right) = \dot{m} \left(c_p (T_1 - T_2) + \frac{V_1^2 - V_2^2}{2} \right)$$



The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500+273\text{ K})}{1000\text{ kPa}} = 0.2219\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{(0.2\text{ m}^2)(40\text{ m/s})}{0.2219\text{ m}^3/\text{kg}} = \mathbf{36.06\text{ kg/s}}$$

Similarly at the outlet,

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(150+273\text{ K})}{100\text{ kPa}} = 1.214\text{ m}^3/\text{kg}$$

$$V_2 = \frac{\dot{m} \nu_2}{A_2} = \frac{(36.06\text{ kg/s})(1.214\text{ m}^3/\text{kg})}{1\text{ m}^2} = 43.78\text{ m/s}$$

(b) Substituting into the energy balance equation gives

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m} \left(c_p (T_1 - T_2) + \frac{V_1^2 - V_2^2}{2} \right) \\ &= (36.06\text{ kg/s}) \left[(1.051\text{ kJ/kg}\cdot\text{K})(500-150)\text{ K} + \frac{(40\text{ m/s})^2 - (43.78\text{ m/s})^2}{2} \left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2} \right) \right] \\ &= \mathbf{13,260\text{ kW}} \end{aligned}$$

6-61 Air is compressed in an adiabatic compressor. The mass flow rate of the air and the power input are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 The compressor is adiabatic. 3 Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of $(20+400)/2=210^\circ\text{C}=483\text{ K}$ is $c_p = 1.026\text{ kJ/kg}\cdot\text{K}$ (Table A-2b). The gas constant of air is $R = 0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).

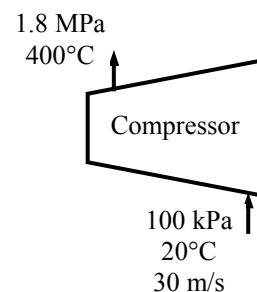
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) + \dot{W}_{\text{in}} = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$\dot{W}_{\text{in}} = \dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right) = \dot{m} \left(c_p (T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} \right)$$



The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273\text{ K})}{100\text{ kPa}} = 0.8409\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{(0.15\text{ m}^2)(30\text{ m/s})}{0.8409\text{ m}^3/\text{kg}} = \mathbf{5.351\text{ kg/s}}$$

Similarly at the outlet,

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400 + 273\text{ K})}{1800\text{ kPa}} = 0.1073\text{ m}^3/\text{kg}$$

$$V_2 = \frac{\dot{m} \nu_2}{A_2} = \frac{(5.351\text{ kg/s})(0.1073\text{ m}^3/\text{kg})}{0.08\text{ m}^2} = 7.177\text{ m/s}$$

(b) Substituting into the energy balance equation gives

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{m} \left(c_p (T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} \right) \\ &= (5.351\text{ kg/s}) \left[(1.026\text{ kJ/kg}\cdot\text{K})(400 - 20)\text{K} + \frac{(7.177\text{ m/s})^2 - (30\text{ m/s})^2}{2} \left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2} \right) \right] \\ &= \mathbf{2084\text{ kW}} \end{aligned}$$

6-62E Air is expanded in an adiabatic turbine. The mass flow rate of the air and the power produced are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 The turbine is well-insulated, and thus there is no heat transfer. 3 Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of $(800+250)/2=525^\circ\text{F}$ is $c_p = 0.2485 \text{ Btu/lbm}\cdot\text{R}$ (Table A-2Eb). The gas constant of air is $R = 0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$ (Table A-1E).

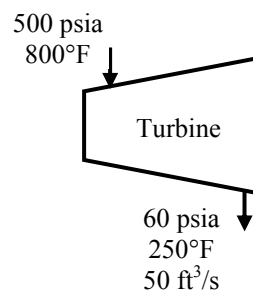
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right) + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m} \left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} \right) = \dot{m} \left(c_p (T_1 - T_2) + \frac{V_1^2 - V_2^2}{2} \right)$$



The specific volume of air at the exit and the mass flow rate are

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(250 + 460 \text{ R})}{60 \text{ psia}} = 4.383 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_2}{\nu_2} = \frac{50 \text{ ft}^3/\text{s}}{4.383 \text{ ft}^3/\text{lbm}} = 11.41 \text{ kg/s}$$

$$V_2 = \frac{\dot{m} \nu_2}{A_2} = \frac{(11.41 \text{ lbm/s})(4.383 \text{ ft}^3/\text{lbm})}{1.2 \text{ ft}^2} = 41.68 \text{ ft/s}$$

Similarly at the inlet,

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(800 + 460 \text{ R})}{500 \text{ psia}} = 0.9334 \text{ ft}^3/\text{lbm}$$

$$V_1 = \frac{\dot{m} \nu_1}{A_1} = \frac{(11.41 \text{ lbm/s})(0.9334 \text{ ft}^3/\text{lbm})}{0.6 \text{ ft}^2} = 17.75 \text{ ft/s}$$

Substituting into the energy balance equation gives

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m} \left(c_p (T_1 - T_2) + \frac{V_1^2 - V_2^2}{2} \right) \\ &= (11.41 \text{ lbm/s}) \left[(0.2485 \text{ Btu/lbm}\cdot\text{R})(800 - 250)\text{R} + \frac{(17.75 \text{ ft/s})^2 - (41.68 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) \right] \\ &= \mathbf{1559 \text{ kW}} \end{aligned}$$

6-63 Steam expands in a two-stage adiabatic turbine from a specified state to another state. Some steam is extracted at the end of the first stage. The power output of the turbine is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6)

$$\left. \begin{array}{l} P_1 = 12.5 \text{ MPa} \\ T_1 = 550^\circ\text{C} \end{array} \right\} h_1 = 3476.5 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 1 \text{ MPa} \\ T_2 = 200^\circ\text{C} \end{array} \right\} h_2 = 2828.3 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_3 = 100 \text{ kPa} \\ T_3 = 100^\circ\text{C} \end{array} \right\} h_3 = 2675.8 \text{ kJ/kg}$$

Analysis The mass flow rate through the second stage is

$$\dot{m}_3 = \dot{m}_1 - \dot{m}_2 = 20 - 1 = 19 \text{ kg/s}$$

We take the entire turbine, including the connection part between the two stages, as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters the turbine and two fluid streams leave, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\phi_0} (\text{steady})}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

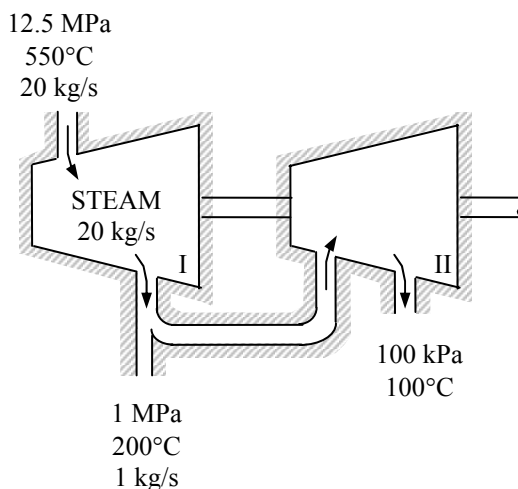
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

Substituting, the power output of the turbine is

$$\begin{aligned} \dot{W}_{\text{out}} &= (20 \text{ kg/s})(3476.5 \text{ kJ/kg}) - (1 \text{ kg/s})(2828.3 \text{ kJ/kg}) - (19 \text{ kg/s})(2675.8 \text{ kJ/kg}) \\ &= \mathbf{15,860 \text{ kW}} \end{aligned}$$



Throttling Valves

6-64C Yes.

6-65C No. Because air is an ideal gas and $h = h(T)$ for ideal gases. Thus if h remains constant, so does the temperature.

6-66C If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

6-67C The temperature of a fluid can increase, decrease, or remain the same during a throttling process. Therefore, this claim is valid since no thermodynamic laws are violated.

6-68 Refrigerant-134a is throttled by a capillary tube. The quality of the refrigerant at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

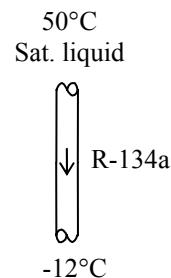
since $\dot{Q} \cong \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0$.

The inlet enthalpy of R-134a is, from the refrigerant tables (Table A-11),

$$\left. \begin{array}{l} T_1 = 50^\circ\text{C} \\ \text{sat. liquid} \end{array} \right\} h_1 = h_f = 123.49 \text{ kJ/kg}$$

The exit quality is

$$\left. \begin{array}{l} T_2 = -12^\circ\text{C} \\ h_2 = h_1 \end{array} \right\} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{123.49 - 35.92}{207.38} = \mathbf{0.422}$$



6-69 Steam is throttled from a specified pressure to a specified state. The quality at the inlet is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

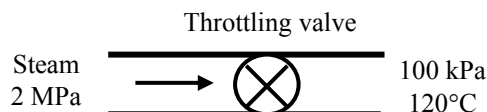
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0$.

The enthalpy of steam at the exit is (Table A-6),

$$\left. \begin{array}{l} P_2 = 100 \text{ kPa} \\ T_2 = 120^\circ\text{C} \end{array} \right\} h_2 = 2716.1 \text{ kJ/kg}$$

The quality of the steam at the inlet is (Table A-5)

$$\left. \begin{array}{l} P_2 = 2000 \text{ kPa} \\ h_1 = h_2 = 2716.1 \text{ kJ/kg} \end{array} \right\} x_1 = \frac{h_2 - h_f}{h_{fg}} = \frac{2716.1 - 908.47}{1889.8} = \mathbf{0.957}$$

6-70 CD EES Refrigerant-134a is throttled by a valve. The pressure and internal energy after expansion are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

$$\left. \begin{array}{l} P_1 = 0.8 \text{ MPa} \\ T_1 = 25^\circ\text{C} \end{array} \right\} h_1 \cong h_{f@25^\circ\text{C}} = 86.41 \text{ kJ/kg}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\text{net}} \quad (\text{steady}) = 0$$

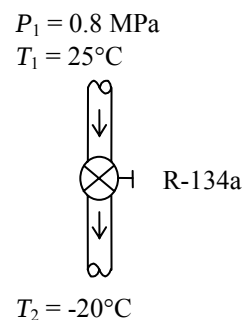
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left. \begin{array}{l} T_2 = -20^\circ\text{C} \\ (h_2 = h_1) \end{array} \right\} \begin{array}{ll} h_f = 25.49 \text{ kJ/kg}, & u_f = 25.39 \text{ kJ/kg} \\ h_g = 238.41 \text{ kJ/kg} & u_g = 218.84 \text{ kJ/kg} \end{array}$$



Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state, and thus

$$P_2 = P_{\text{sat}@-20^\circ\text{C}} = \mathbf{132.82 \text{ kPa}}$$

Also,

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{86.41 - 25.49}{212.91} = 0.2861$$

Thus,

$$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74 \text{ kJ/kg}}$$

6-71 Steam is throttled by a well-insulated valve. The temperature drop of the steam after the expansion is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of steam is (Tables A-6),

$$\left. \begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500^\circ\text{C} \end{array} \right\} h_1 = 3399.5 \text{ kJ/kg}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\text{net}} \text{ (steady)} = 0$$

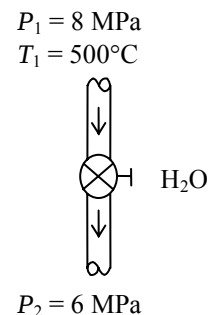
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then the exit temperature of steam becomes

$$\left. \begin{array}{l} P_2 = 6 \text{ MPa} \\ (h_2 = h_1) \end{array} \right\} T_2 = \mathbf{490.1^\circ\text{C}}$$



6-72 EES Problem 6-71 is reconsidered. The effect of the exit pressure of steam on the exit temperature after throttling as the exit pressure varies from 6 MPa to 1 MPa is to be investigated. The exit temperature of steam is to be plotted against the exit pressure.

Analysis The problem is solved using EES, and the solution is given below.

"Input information from Diagram Window"

{WorkingFluid\$='Steam_iapws' "WorkingFluid: can be changed to ammonia or other fluids"

P_in=8000 [kPa]

T_in=500 [C]

P_out=6000 [kPa]}

\$Warning off

"Analysis"

m_dot_in=m_dot_out "steady-state mass balance"

m_dot_in=1 "mass flow rate is arbitrary"

m_dot_in*h_in+Q_dot-W_dot-m_dot_out*h_out=0 "steady-state energy balance"

Q_dot=0 "assume the throttle to operate adiabatically"

W_dot=0 "throttles do not have any means of producing power"

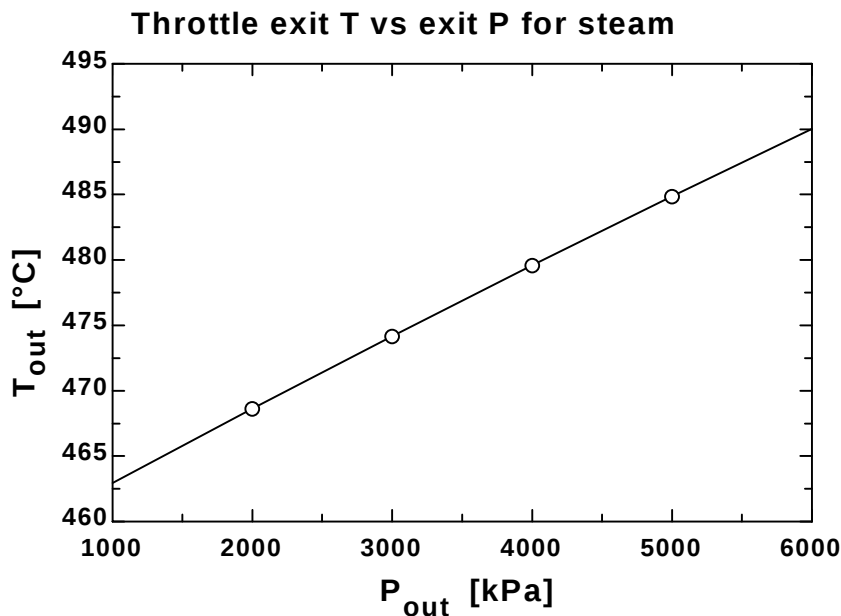
h_in=enthalpy(WorkingFluid\$,T=T_in,P=P_in) "property table lookup"

T_out=temperature(WorkingFluid\$,P=P_out,h=h_out) "property table lookup"

x_out=quality(WorkingFluid\$,P=P_out,h=h_out) "x_out is the quality at the outlet"

P[1]=P_in; P[2]=P_out; h[1]=h_in; h[2]=h_out "use arrays to place points on property plot"

P _{out} [kPa]	T _{out} [C]
1000	463.1
2000	468.8
3000	474.3
4000	479.7
5000	484.9
6000	490.1



6-73E High-pressure air is throttled to atmospheric pressure. The temperature of air after the expansion is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved. **5** Air is an ideal gas.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \xrightarrow{\text{steady}} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.

For an ideal gas,

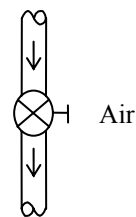
$$h = h(T).$$

Therefore,

$$T_2 = T_1 = \mathbf{90^\circ\text{F}}$$

$$P_1 = 200 \text{ psia}$$

$$T_1 = 90^\circ\text{F}$$



$$P_2 = 14.7 \text{ psia}$$

6-74 Carbon dioxide flows through a throttling valve. The temperature change of CO₂ is to be determined if CO₂ is assumed an ideal gas and a real gas.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

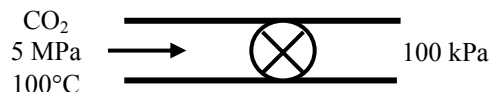
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \xrightarrow{\text{steady}} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.

(a) For an ideal gas, $h = h(T)$, and therefore,

$$T_2 = T_1 = 100^\circ\text{C} \longrightarrow \Delta T = T_1 - T_2 = 0^\circ\text{C}$$

(b) We obtain real gas properties of CO₂ from EES software as follows

$$\left. \begin{array}{l} P_1 = 5 \text{ MPa} \\ T_1 = 100^\circ\text{C} \end{array} \right\} h_1 = 34.77 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 100 \text{ kPa} \\ h_2 = h_1 = 34.77 \text{ kJ/kg} \end{array} \right\} T_2 = 66.0^\circ\text{C}$$

Note that EES uses a different reference state from the textbook for CO₂ properties. The temperature difference in this case becomes

$$\Delta T = T_1 - T_2 = 100 - 66.0 = 34.0^\circ\text{C}$$

That is, the temperature of CO₂ decreases by 34°C in a throttling process if its real gas properties are used.

Mixing Chambers and Heat Exchangers

6-75C Yes, if the mixing chamber is losing heat to the surrounding medium.

6-76C Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

6-77C Under the conditions of no heat and work interactions between the heat exchanger and the surrounding medium.

6-78 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant. 5 There are no work interactions.

Properties Noting that $T < T_{\text{sat @ 250 kPa}} = 127.41^\circ\text{C}$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,

$$h_1 \cong h_f @ 80^\circ\text{C} = 335.02 \text{ kJ/kg}$$

$$h_2 \cong h_f @ 20^\circ\text{C} = 83.915 \text{ kJ/kg}$$

$$h_3 \cong h_f @ 42^\circ\text{C} = 175.90 \text{ kJ/kg}$$

Analysis We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\text{no}}{\text{steady}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no}}{\text{steady}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

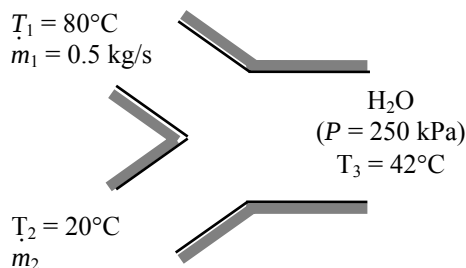
Combining the two relations and solving for $\dot{m}_2$ gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$$

Substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{(335.02 - 175.90) \text{ kJ/kg}}{(175.90 - 83.915) \text{ kJ/kg}} (0.5 \text{ kg/s}) = \mathbf{0.865 \text{ kg/s}}$$



6-79E Liquid water is heated in a chamber by mixing it with saturated water vapor. If both streams enter at the same rate, the temperature and quality (if saturated) of the exit stream is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From steam tables (Tables A-5E through A-6E),

$$h_1 \cong h_f @ 50^\circ\text{F} = 18.07 \text{ Btu/lbm}$$

$$h_2 = h_g @ 50 \text{ psia} = 1174.2 \text{ Btu/lbm}$$

Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

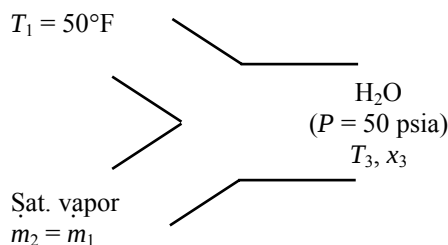
Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 2\dot{m}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$



Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two gives

$$\dot{m} h_1 + \dot{m} h_2 = 2\dot{m} h_3 \quad \text{or} \quad h_3 = (h_1 + h_2) / 2$$

Substituting,

$$h_3 = (18.07 + 1174.2) / 2 = 596.16 \text{ Btu/lbm}$$

At 50 psia, $h_f = 250.21 \text{ Btu/lbm}$ and $h_g = 1174.2 \text{ Btu/lbm}$. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\text{sat}} @ 50 \text{ psia} = \mathbf{280.99^\circ\text{F}}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{596.16 - 250.21}{924.03} = \mathbf{0.374}$$

6-80 Two streams of refrigerant-134a are mixed in a chamber. If the cold stream enters at twice the rate of the hot stream, the temperature and quality (if saturated) of the exit stream are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From R-134a tables (Tables A-11 through A-13),

$$h_1 \cong h_f @ 12^\circ\text{C} = 68.18 \text{ kJ/kg}$$

$$h_2 = h @ 1 \text{ MPa}, 60^\circ\text{C} = 293.38 \text{ kJ/kg}$$

Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\text{no}}{\text{}} \text{ (steady)} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2 \text{ since } \dot{m}_1 = 2\dot{m}_2$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no}}{\text{}} \text{ (steady)} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \text{ (since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two gives $2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3$ or $h_3 = (2h_1 + h_2)/3$

Substituting,

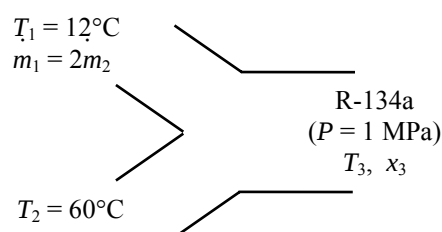
$$h_3 = (2 \times 68.18 + 293.38)/3 = 143.25 \text{ kJ/kg}$$

At 1 MPa, $h_f = 107.32 \text{ kJ/kg}$ and $h_g = 270.99 \text{ kJ/kg}$. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\text{sat}} @ 1 \text{ MPa} = 39.37^\circ\text{C}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{143.25 - 107.32}{163.67} = 0.220$$



6-81 EES Problem 6-80 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature and quality are to be plotted against the cold-to-hot mass flow rate ratio.

Analysis The problem is solved using EES, and the solution is given below.

"Input Data"

"m_frac = 2" "m_frac=m_dot_cold/m_dot_hot= m_dot_1/m_dot_2"

T[1]=12 [C]

P[1]=1000 [kPa]

T[2]=60 [C]

P[2]=1000 [kPa]

m_dot_1=m_frac*m_dot_2

P[3]=1000 [kPa]

m_dot_1=1

"Conservation of mass for the R134a: Sum of m_dot_in=m_dot_out"

m_dot_1+ m_dot_2 =m_dot_3

"Conservation of Energy for steady-flow: neglect changes in KE and PE"

"We assume no heat transfer and no work occur across the control surface."

E_dot_in - E_dot_out = DELTAE_dot_cv

DELTA E_dot_cv=0 "Steady-flow requirement"

E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]

E_dot_out=m_dot_3*h[3]

"Property data are given by:"

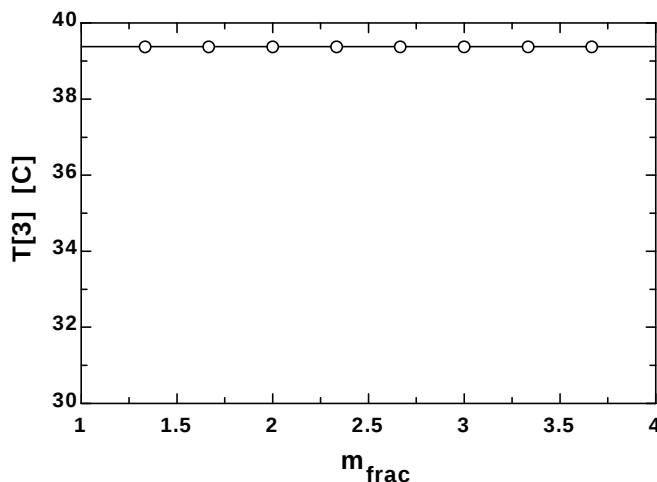
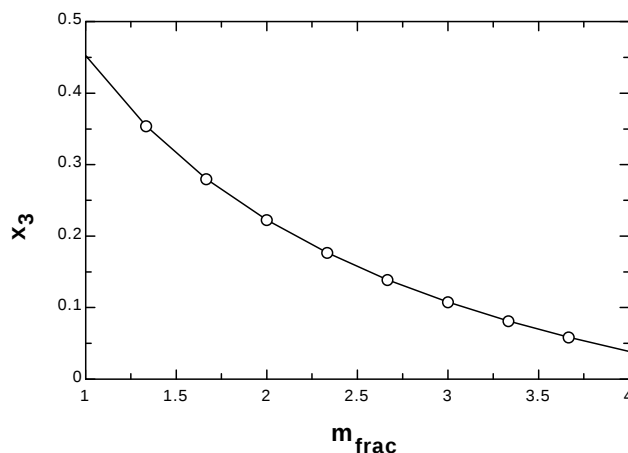
h[1] =enthalpy(R134a,T=T[1],P=P[1])

h[2] =enthalpy(R134a,T=T[2],P=P[2])

T[3] =temperature(R134a,P=P[3],h=h[3])

x_3=QUALITY(R134a,h=h[3],P=P[3])

m _{frac}	T ₃ [C]	x ₃
1	39.37	0.4491
1.333	39.37	0.3509
1.667	39.37	0.2772
2	39.37	0.2199
2.333	39.37	0.174
2.667	39.37	0.1365
3	39.37	0.1053
3.333	39.37	0.07881
3.667	39.37	0.05613
4	39.37	0.03649



6-82 Refrigerant-134a is condensed in a water-cooled condenser. The mass flow rate of the cooling water required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

Properties The enthalpies of R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left. \begin{array}{l} P_3 = 700 \text{ kPa} \\ T_3 = 70^\circ\text{C} \end{array} \right\} h_3 = 308.33 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_4 = 700 \text{ kPa} \\ \text{sat. liquid} \end{array} \right\} h_4 = h_f @ 700 \text{ kPa} = 88.82 \text{ kJ/kg}$$

Water exists as compressed liquid at both states, and thus (Table A-4)

$$h_1 \cong h_f @ 15^\circ\text{C} = 62.98 \text{ kJ/kg}$$

$$h_2 \cong h_f @ 25^\circ\text{C} = 104.83 \text{ kJ/kg}$$

Analysis We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_w \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

Energy balance (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

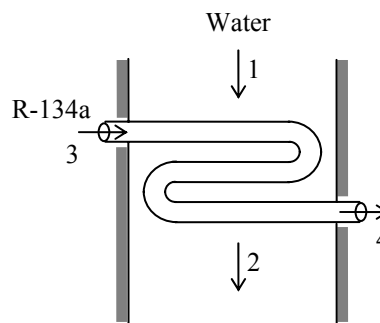
Combining the two, $\dot{m}_w (h_2 - h_1) = \dot{m}_R (h_3 - h_4)$

Solving for $\dot{m}_w$:

$$\dot{m}_w = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_R$$

Substituting,

$$\dot{m}_w = \frac{(308.33 - 88.82) \text{ kJ/kg}}{(104.83 - 62.98) \text{ kJ/kg}} (8 \text{ kg/min}) = \mathbf{42.0 \text{ kg/min}}$$



6-83E CD EES Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 There are no work interactions. 4 Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 5 Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$ (Table A-1E). The constant pressure specific heat of air is $c_p = 0.240 \text{ Btu/lbm}\cdot^\circ\text{F}$ (Table A-2E). The enthalpies of steam at the inlet and the exit states are (Tables A-4E through A-6E)

$$\left. \begin{array}{l} P_3 = 30 \text{ psia} \\ T_3 = 400^\circ\text{F} \end{array} \right\} h_3 = 1237.9 \text{ Btu/lbm}$$

$$\left. \begin{array}{l} P_4 = 25 \text{ psia} \\ T_4 = 212^\circ\text{F} \end{array} \right\} h_4 \cong h_{f@212^\circ\text{F}} = 180.21 \text{ Btu/lbm}$$

Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no}}{=} 0 \text{ (steady)} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_a \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

Energy balance (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no}}{=} 0 \text{ (steady)} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two, $\dot{m}_a (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$

Solving for $\dot{m}_a$:

$$\dot{m}_a = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \frac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$$

Substituting,

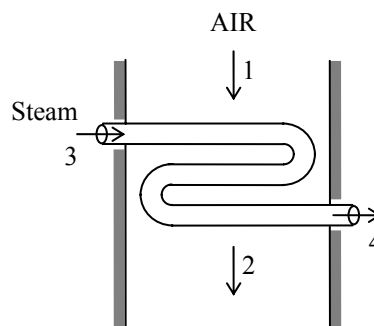
$$\dot{m}_a = \frac{(1237.9 - 180.21) \text{ Btu/lbm}}{(0.240 \text{ Btu/lbm}\cdot^\circ\text{F})(130 - 80)^\circ\text{F}} (15 \text{ lbm/min}) = 1322 \text{ lbm/min} = 22.04 \text{ lbm/s}$$

Also,

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540 \text{ R})}{14.7 \text{ psia}} = 13.61 \text{ ft}^3/\text{lbm}$$

Then the volume flow rate of air at the inlet becomes

$$\dot{V}_1 = \dot{m}_a \nu_1 = (22.04 \text{ lbm/s})(13.61 \text{ ft}^3/\text{lbm}) = \mathbf{300 \text{ ft}^3/\text{s}}$$



6-84 Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C , the minimum mass flow rate of the cooling water required is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 There are no work interactions. 4 Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 5 Liquid water is an incompressible substance with constant specific heats at room temperature.

Properties The cooling water exists as compressed liquid at both states, and its specific heat at room temperature is $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3). The enthalpies of the steam at the inlet and the exit states are (Tables A-5 and A-6)

$$\left. \begin{array}{l} P_3 = 20 \text{ kPa} \\ x_3 = 0.95 \end{array} \right\} h_3 = h_f + x_3 h_{fg} = 251.42 + 0.95 \times 2357.5 = 2491.1 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_4 = 20 \text{ kPa} \\ \text{sat. liquid} \end{array} \right\} h_4 \cong h_{f@20 \text{ kPa}} = 251.42 \text{ kJ/kg}$$

Analysis We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_w \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

Energy balance (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two,

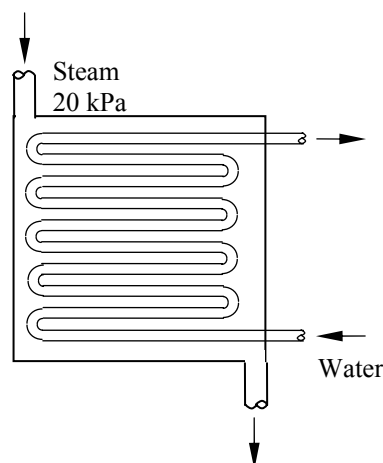
$$\dot{m}_w (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$$

Solving for $\dot{m}_w$:

$$\dot{m}_w = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \frac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$$

Substituting,

$$\dot{m}_w = \frac{(2491.1 - 251.42) \text{ kJ/kg}}{(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(10^\circ\text{C})} (20,000/3600 \text{ kg/s}) = \mathbf{297.7 \text{ kg/s}}$$



6-85 Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer in the heat exchanger and the mass flow rate of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg·°C, respectively.

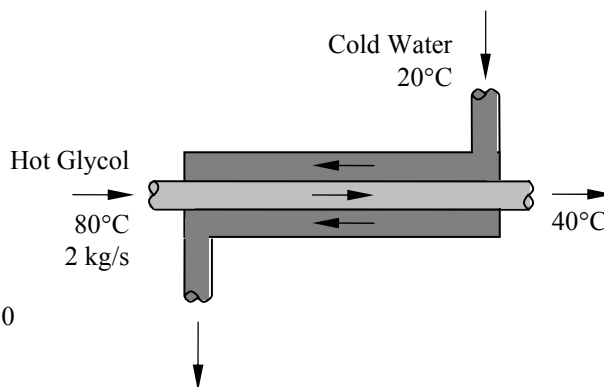
Analysis (a) We take the ethylene glycol tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{glycol}} = (2 \text{ kg/s})(2.56 \text{ kJ/kg} \cdot ^\circ\text{C})(80^\circ\text{C} - 40^\circ\text{C}) = \mathbf{204.8 \text{ kW}}$$

(b) The rate of heat transfer from glycol must be equal to the rate of heat transfer to the water. Then,

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{204.8 \text{ kJ/s}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(55^\circ\text{C} - 20^\circ\text{C})} = \mathbf{1.4 \text{ kg/s}}$$

6-86 EES Problem 6-85 is reconsidered. The effect of the inlet temperature of cooling water on the mass flow rate of water as the inlet temperature varies from 10°C to 40°C at constant exit temperature) is to be investigated. The mass flow rate of water is to be plotted against the inlet temperature.

Analysis The problem is solved using EES, and the solution is given below.

"Input Data"

```
{T_w[1]=20 [C]}
T_w[2]=55 [C] "w: water"
m_dot_eg=2 [kg/s] "eg: ethylene glycol"
T_eg[1]=80 [C]
T_eg[2]=40 [C]
C_p_w=4.18 [kJ/kg-K]
C_p_eg=2.56 [kJ/kg-K]
```

"Conservation of mass for the water: $m_{\dot{w}_{in}}=m_{\dot{w}_{out}}=m_{\dot{w}}$ "

"Conservation of mass for the ethylene glycol: $m_{\dot{eg}_{in}}=m_{\dot{eg}_{out}}=m_{\dot{eg}}$ "

"Conservation of Energy for steady-flow: neglect changes in KE and PE in each mass stream"

"We assume no heat transfer and no work occur across the control surface."

$E_{\dot{in}} - E_{\dot{out}} = \Delta E_{\dot{cv}}$

$\Delta E_{\dot{cv}}=0$ "Steady-flow requirement"

$E_{\dot{in}}=m_{\dot{w}}h_w[1] + m_{\dot{eg}}h_{eg}[1]$

$E_{\dot{out}}=m_{\dot{w}}h_w[2] + m_{\dot{eg}}h_{eg}[2]$

$Q_{\text{exchanged}}=m_{\dot{eg}}h_{eg}[1] - m_{\dot{eg}}h_{eg}[2]$

"Property data are given by:"

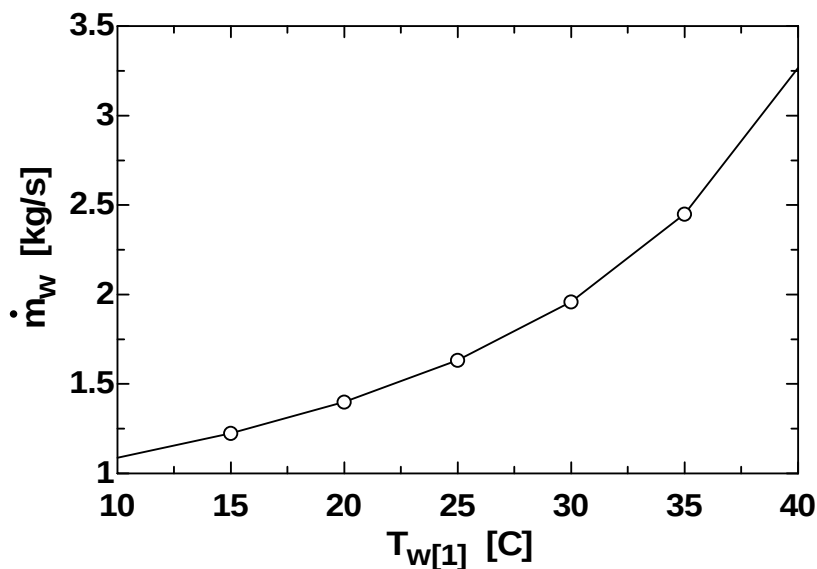
$h_w[1]=C_{p_w}T_w[1]$ "liquid approximation applied for water and ethylene glycol"

$h_w[2]=C_{p_w}T_w[2]$

$h_{eg}[1]=C_{p_{eg}}T_{eg}[1]$

$h_{eg}[2]=C_{p_{eg}}T_{eg}[2]$

m_w [kg/s]	$T_{w,1}$ [C]
1.089	10
1.225	15
1.4	20
1.633	25
1.96	30
2.45	35
3.266	40



6-87 Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg·°C, respectively.

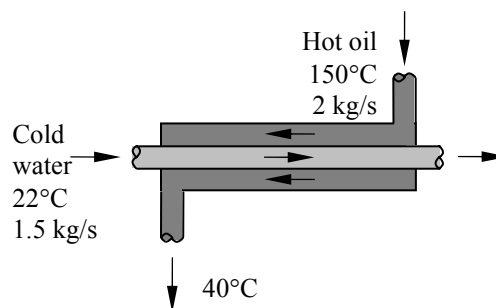
Analysis We take the oil tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} = (2 \text{ kg/s})(2.2 \text{ kJ/kg} \cdot ^\circ\text{C})(150^\circ\text{C} - 40^\circ\text{C}) = \mathbf{484 \text{ kW}}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}_{\text{water}}c_p} = 22^\circ\text{C} + \frac{484 \text{ kJ/s}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{99.2^\circ\text{C}}$$

6-88 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the exit temperature of hot water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg·°C, respectively.

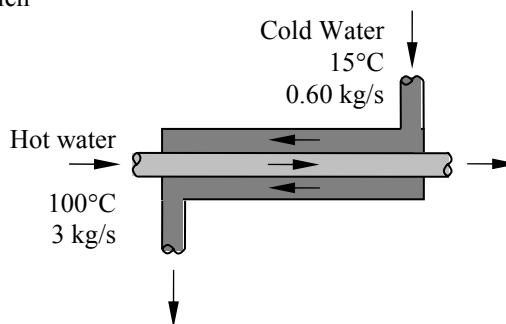
Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cold water}} = (0.60 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(45^\circ\text{C} - 15^\circ\text{C}) = \mathbf{75.24 \text{ kW}}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{hot water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 100^\circ\text{C} - \frac{75.24 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{94.0^\circ\text{C}}$$

6-89 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg·°C, respectively.

Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the exhaust gases becomes

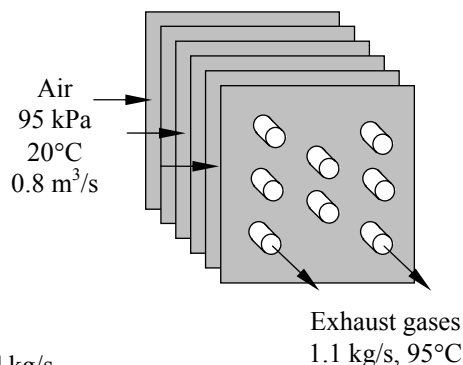
$$\begin{aligned} \dot{Q} &= [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas}} \\ &= (1.1 \text{ kg/s})(1.1 \text{ kJ/kg} \cdot ^\circ\text{C})(180^\circ\text{C} - 95^\circ\text{C}) \\ &= \mathbf{102.85 \text{ kW}} \end{aligned}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95 \text{ kPa})(0.8 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}) \times 293 \text{ K}} = 0.904 \text{ kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = \dot{m}c_p(T_{\text{c,out}} - T_{\text{c,in}}) \longrightarrow T_{\text{c,out}} = T_{\text{c,in}} + \frac{\dot{Q}}{\dot{m}c_p} = 20^\circ\text{C} + \frac{102.85 \text{ kW}}{(0.904 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{133.2^\circ\text{C}}$$



6-90 An adiabatic open feedwater heater mixes steam with feedwater. The outlet mass flow rate and the outlet velocity are to be determined.

Assumptions Steady operating conditions exist.

Properties From a mass balance

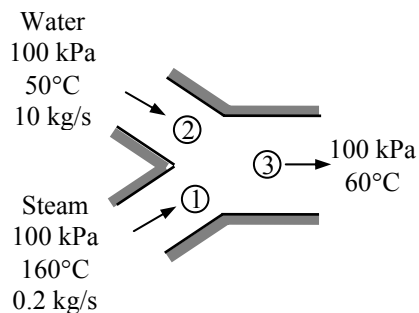
$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.2 + 10 = \mathbf{10.2 \text{ kg/s}}$$

The specific volume at the exit is (Table A-4)

$$\left. \begin{array}{l} P_3 = 100 \text{ kPa} \\ T_3 = 60^\circ\text{C} \end{array} \right\} \nu_3 \cong \nu_f @ 60^\circ\text{C} = 0.001017 \text{ m}^3/\text{kg}$$

The exit velocity is then

$$\begin{aligned} V_3 &= \frac{\dot{m}_3 \nu_3}{A_3} = \frac{4\dot{m}_3 \nu_3}{\pi D^2} \\ &= \frac{4(10.2 \text{ kg/s})(0.001017 \text{ m}^3/\text{kg})}{\pi(0.03 \text{ m})^2} \\ &= \mathbf{14.68 \text{ m/s}} \end{aligned}$$



6-91E An adiabatic open feedwater heater mixes steam with feedwater. The outlet mass flow rate and the outlet velocity are to be determined for two exit temperatures.

Assumptions Steady operating conditions exist.

Properties From a mass balance

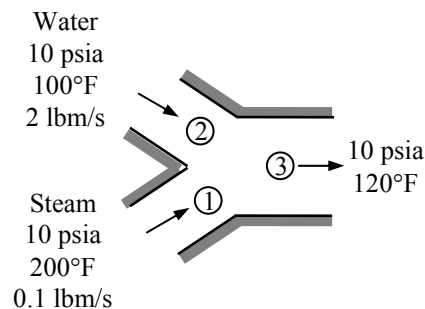
$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.1 + 2 = \mathbf{2.1 \text{ lbm/s}}$$

The specific volume at the exit is (Table A-4E)

$$\left. \begin{array}{l} P_3 = 10 \text{ psia} \\ T_3 = 120^\circ\text{F} \end{array} \right\} \nu_3 \cong \nu_f @ 120^\circ\text{F} = 0.01620 \text{ ft}^3/\text{lbm}$$

The exit velocity is then

$$\begin{aligned} V_3 &= \frac{\dot{m}_3 \nu_3}{A_3} = \frac{4 \dot{m}_3 \nu_3}{\pi D^2} \\ &= \frac{4(2.1 \text{ lbm/s})(0.01620 \text{ ft}^3/\text{lbm})}{\pi(0.5 \text{ ft})^2} = \mathbf{0.1733 \text{ ft/s}} \end{aligned}$$



When the temperature at the exit is 180°F, we have

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.1 + 2 = \mathbf{2.1 \text{ lbm/s}}$$

$$\left. \begin{array}{l} P_3 = 10 \text{ psia} \\ T_3 = 180^\circ\text{F} \end{array} \right\} \nu_3 \cong \nu_f @ 180^\circ\text{F} = 0.01651 \text{ ft}^3/\text{lbm}$$

$$V_3 = \frac{\dot{m}_3 \nu_3}{A_3} = \frac{4 \dot{m}_3 \nu_3}{\pi D^2} = \frac{4(2.1 \text{ lbm/s})(0.01651 \text{ ft}^3/\text{lbm})}{\pi(0.5 \text{ ft})^2} = \mathbf{0.1766 \text{ ft/s}}$$

The mass flow rate at the exit is same while the exit velocity slightly increases when the exit temperature is 180°F instead of 120°F.

6-92 Water is heated by hot oil in a heat exchanger. The rate of heat transfer in the heat exchanger and the outlet temperature of oil are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg·°C, respectively.

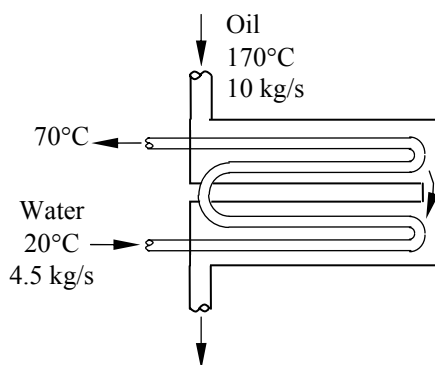
Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \approx \Delta pe \approx 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(70^\circ\text{C} - 20^\circ\text{C}) = \mathbf{940.5 \text{ kW}}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170^\circ\text{C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{129.1^\circ\text{C}}$$

6-93E Steam is condensed by cooling water in a condenser. The rate of heat transfer in the heat exchanger and the rate of condensation of steam are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy of vaporization of water at 85°F is 1045.2 Btu/lbm (Table A-4E).

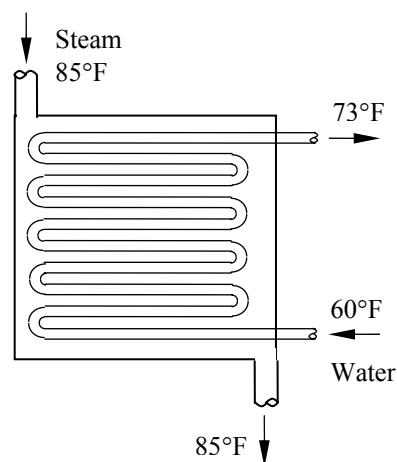
Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (138 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73^\circ\text{F} - 60^\circ\text{F}) = \mathbf{1794 \text{ Btu/s}}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1794 \text{ Btu/s}}{1045.2 \text{ Btu/lbm}} = \mathbf{1.72 \text{ lbm/s}}$$

6-94 Two streams of cold and warm air are mixed in a chamber. If the ratio of hot to cold air is 1.6, the mixture temperature and the rate of heat gain of the room are to be determined.

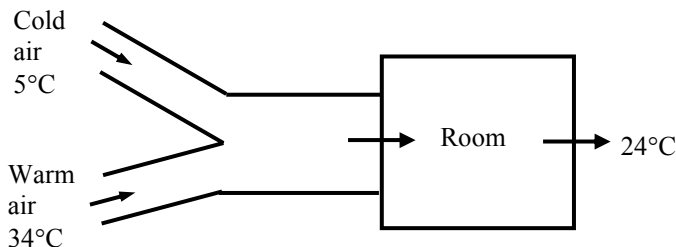
Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 There are no work interactions. 4 The device is adiabatic and thus heat transfer is negligible.

Properties The gas constant of air is $R = 0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$. The enthalpies of air are obtained from air table (Table A-21) as

$$h_1 = h_{@278 \text{ K}} = 278.13 \text{ kJ/kg}$$

$$h_2 = h_{@307 \text{ K}} = 307.23 \text{ kJ/kg}$$

$$h_{\text{room}} = h_{@297 \text{ K}} = 297.18 \text{ kJ/kg}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + 1.6\dot{m}_1 = \dot{m}_3 = 2.6\dot{m}_1 \text{ since } \dot{m}_2 = 1.6\dot{m}_1$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two gives $\dot{m}_1 h_1 + 1.6\dot{m}_1 h_2 = 2.6\dot{m}_1 h_3$ or $h_3 = (h_1 + 1.6h_2)/2.6$

Substituting,

$$h_3 = (278.13 + 1.6 \times 307.23)/2.6 = 296.04 \text{ kJ/kg}$$

From air table at this enthalpy, the mixture temperature is

$$T_3 = T_{@h=296.04 \text{ kJ/kg}} = 295.9 \text{ K} = \mathbf{22.9^\circ\text{C}}$$

(b) The mass flow rates are determined as follows

$$\nu_1 = \frac{RT_1}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(5 + 273 \text{ K})}{105 \text{ kPa}} = 0.7599 \text{ m}^3/\text{kg}$$

$$\dot{m}_1 = \frac{\dot{\mathcal{V}}_1}{\nu_1} = \frac{1.25 \text{ m}^3/\text{s}}{0.7599 \text{ m}^3/\text{kg}} = 1.645 \text{ kg/s}$$

$$\dot{m}_3 = 2.6\dot{m}_1 = 2.6(1.645 \text{ kg/s}) = 4.277 \text{ kg/s}$$

The rate of heat gain of the room is determined from

$$\dot{Q}_{\text{cool}} = \dot{m}_3(h_{\text{room}} - h_3) = (4.277 \text{ kg/s})(297.18 - 296.04) \text{ kJ/kg} = \mathbf{4.88 \text{ kW}}$$

6-95 A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 There are no work interactions. 4 Exhaust gases are assumed to have air properties with constant specific heats.

Properties The constant pressure specific heat of the exhaust gases is taken to be $c_p = 1.045 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2). The inlet and exit enthalpies of water are (Tables A-4 and A-5)

$$\left. \begin{array}{l} T_{w,\text{in}} = 15^\circ\text{C} \\ x = 0 \text{ (sat. liq.)} \end{array} \right\} h_{w,\text{in}} = 62.98 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_{w,\text{out}} = 2 \text{ MPa} \\ x = 1 \text{ (sat. vap.)} \end{array} \right\} h_{w,\text{out}} = 2798.3 \text{ kJ/kg}$$

Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0 \longrightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

Energy balance (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_{\text{exh}} h_{\text{exh},\text{in}} + \dot{m}_w h_{w,\text{in}} = \dot{m}_{\text{exh}} h_{\text{exh},\text{out}} + \dot{m}_w h_{w,\text{out}} + \dot{Q}_{\text{out}} \quad (\text{since } \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

or
$$\dot{m}_{\text{exh}} c_p T_{\text{exh},\text{in}} + \dot{m}_w h_{w,\text{in}} = \dot{m}_{\text{exh}} c_p T_{\text{exh},\text{out}} + \dot{m}_w h_{w,\text{out}} + \dot{Q}_{\text{out}}$$

Noting that the mass flow rate of exhaust gases is 15 times that of the water, substituting gives

$$\begin{aligned} 15\dot{m}_w (1.045 \text{ kJ/kg} \cdot ^\circ\text{C})(400^\circ\text{C}) + \dot{m}_w (62.98 \text{ kJ/kg}) \\ = 15\dot{m}_w (1.045 \text{ kJ/kg} \cdot ^\circ\text{C})T_{\text{exh},\text{out}} + \dot{m}_w (2798.3 \text{ kJ/kg}) + \dot{Q}_{\text{out}} \end{aligned} \quad (1)$$

The heat given up by the exhaust gases and heat picked up by the water are

$$\dot{Q}_{\text{exh}} = \dot{m}_{\text{exh}} c_p (T_{\text{exh},\text{in}} - T_{\text{exh},\text{out}}) = 15\dot{m}_w (1.045 \text{ kJ/kg} \cdot ^\circ\text{C})(400 - T_{\text{exh},\text{out}})^\circ\text{C} \quad (2)$$

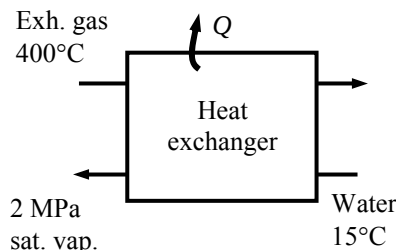
$$\dot{Q}_w = \dot{m}_w (h_{w,\text{out}} - h_{w,\text{in}}) = \dot{m}_w (2798.3 - 62.98) \text{ kJ/kg} \quad (3)$$

The heat loss is

$$\dot{Q}_{\text{out}} = f_{\text{heat loss}} \dot{Q}_{\text{exh}} = 0.1 \dot{Q}_{\text{exh}} \quad (4)$$

The solution may be obtained by a trial-error approach. Or, solving the above equations simultaneously using EES software, we obtain

$$T_{\text{exh},\text{out}} = \mathbf{206.1^\circ\text{C}}, \dot{Q}_w = \mathbf{97.26 \text{ kW}}, \dot{m}_w = 0.03556 \text{ kg/s}, \dot{m}_{\text{exh}} = 0.5333 \text{ kg/s}$$



6-96 Two streams of water are mixed in an insulated chamber. The temperature of the exit stream is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-4),

$$h_1 \cong h_{f@90^\circ\text{C}} = 377.04 \text{ kJ/kg}$$

$$h_2 \cong h_{f@50^\circ\text{C}} = 209.34 \text{ kJ/kg}$$

Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

Mass balance: $\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

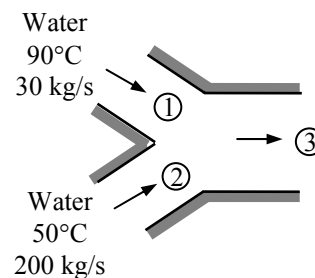
$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Solving for the exit enthalpy,

$$h_3 = \frac{\dot{m}_1 h_1 + \dot{m}_2 h_2}{\dot{m}_1 + \dot{m}_2} = \frac{(30)(377.04) + (200)(209.34)}{30 + 200} = 231.21 \text{ kJ/kg}$$

The temperature corresponding to this enthalpy is

$$T_3 = \mathbf{55.2^\circ\text{C}} \quad (\text{Table A-4})$$



6-97 A chilled-water heat-exchange unit is designed to cool air by water. The maximum water outlet temperature is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1). The constant pressure specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2a). The specific heat of water is $4.18 \text{ kJ/kg} \cdot \text{K}$ (Table A-3).

Analysis The water temperature at the heat exchanger exit will be maximum when all the heat released by the air is picked up by the water. First, the inlet specific volume and the mass flow rate of air are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(303 \text{ K})}{100 \text{ kPa}} = 0.8696 \text{ m}^3/\text{kg}$$

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{5 \text{ m}^3/\text{s}}{0.8696 \text{ m}^3/\text{kg}} = 5.750 \text{ kg/s}$$

We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_3 = \dot{m}_a \text{ and } \dot{m}_2 = \dot{m}_4 = \dot{m}_w$$

Energy balance (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_a (h_1 - h_3) = \dot{m}_w (h_4 - h_2)$$

$$\dot{m}_a c_{p,a} (T_1 - T_3) = \dot{m}_w c_{p,w} (T_4 - T_2)$$

Solving for the exit temperature of water,

$$T_4 = T_2 + \frac{\dot{m}_a c_{p,a} (T_1 - T_3)}{\dot{m}_w c_{p,w}} = 8^\circ\text{C} + \frac{(5.750 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})(30 - 18)^\circ\text{C}}{(2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{16.3^\circ\text{C}}$$

Pipe and duct Flow

6-98 Saturated liquid water is heated in a steam boiler at a specified rate. The rate of heat transfer in the boiler is to be determined.

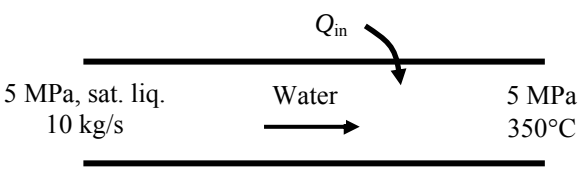
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which the water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{Q}_{\text{in}} = \dot{m}h_2$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1)$$


The enthalpies of water at the inlet and exit of the boiler are (Table A-5, A-6).

$$\left. \begin{array}{l} P_1 = 5 \text{ MPa} \\ x = 0 \end{array} \right\} h_1 \cong h_f @ 5 \text{ MPa} = 1154.5 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 5 \text{ MPa} \\ T_2 = 350^\circ\text{C} \end{array} \right\} h_2 = 3069.3 \text{ kJ/kg}$$

Substituting,

$$\dot{Q}_{\text{in}} = (10 \text{ kg/s})(3069.3 - 1154.5) \text{ kJ/kg} = \mathbf{19,150 \text{ kW}}$$

6-99 Air at a specified rate is heated by an electrical heater. The current is to be determined.

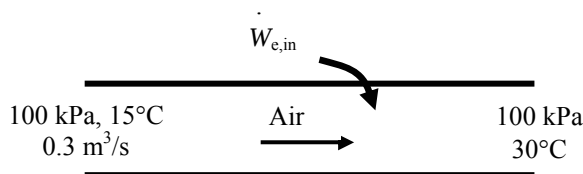
Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The heat losses from the air is negligible.

Properties The gas constant of air is $0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1). The constant pressure specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2a).

Analysis We take the pipe in which the air is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

$$\begin{aligned}\dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}h_1 + \dot{W}_{\text{e,in}} &= \dot{m}h_2 \\ \dot{W}_{\text{e,in}} &= \dot{m}(h_2 - h_1) \\ \mathbf{VI} &= \dot{m}c_p(T_2 - T_1)\end{aligned}$$



The inlet specific volume and the mass flow rate of air are

$$\begin{aligned}\nu_1 &= \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})}{100 \text{ kPa}} = 0.8266 \text{ m}^3/\text{kg} \\ \dot{m} &= \frac{\dot{V}_1}{\nu_1} = \frac{0.3 \text{ m}^3/\text{s}}{0.8266 \text{ m}^3/\text{kg}} = 0.3629 \text{ kg/s}\end{aligned}$$

Substituting into the energy balance equation and solving for the current gives

$$I = \frac{\dot{m}c_p(T_2 - T_1)}{\mathbf{V}} = \frac{(0.3629 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot \text{K})(30 - 15)\text{K}}{110 \text{ V}} \left(\frac{1000 \text{ VI}}{1 \text{ kJ/s}} \right) = \mathbf{49.7 \text{ Amperes}}$$

6-100E The cooling fan of a computer draws air, which is heated in the computer by absorbing the heat of PC circuits. The electrical power dissipated by the circuits is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** All the heat dissipated by the circuits are picked up by the air drawn by the fan.

Properties The gas constant of air is $0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$ (Table A-1E). The constant pressure specific heat of air at room temperature is $c_p = 0.240 \text{ Btu/lbm} \cdot ^\circ\text{F}$ (Table A-2Ea).

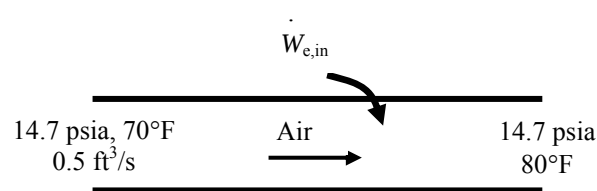
Analysis We take the pipe in which the air is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{e,\text{in}} = \dot{m}h_2$$

$$\dot{W}_{e,\text{in}} = \dot{m}(h_2 - h_1)$$

$$\dot{W}_{e,\text{in}} = \dot{m}c_p(T_2 - T_1)$$


The inlet specific volume and the mass flow rate of air are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(530 \text{ R})}{14.7 \text{ psia}} = 13.35 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.5 \text{ ft}^3/\text{s}}{13.35 \text{ ft}^3/\text{lbm}} = 0.03745 \text{ lbm/s}$$

Substituting,

$$\dot{W}_{e,\text{out}} = (0.03745 \text{ lbm/s})(0.240 \text{ Btu/lbm} \cdot \text{R})(80 - 70) \text{ Btu/lbm} \left(\frac{1 \text{ kW}}{0.94782 \text{ Btu}} \right) = \mathbf{0.0948 \text{ kW}}$$

6-101 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions 1 Steady operation under worst conditions is considered. 2 Air is an ideal gas with constant specific heats. 3 Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of $T_{\text{avg}} = (45+60)/2 = 52.5^\circ\text{C} = 325.5\text{ K}$ is $c_p = 1.0065\text{ kJ/kg}\cdot^\circ\text{C}$. The gas constant for air is $R = 0.287\text{ kJ/kg}\cdot\text{K}$ (Table A-2).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C , and leave at 60°C .

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the required mass flow rate of air to absorb heat at a rate of 60 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{60\text{ W}}{(1006.5\text{ J/kg}\cdot^\circ\text{C})(60 - 45)^\circ\text{C}}$$

$$= 0.00397\text{ kg/s} = 0.238\text{ kg/min}$$



The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(60 + 273)\text{ K}} = 0.6972\text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.238\text{ kg/min}}{0.6972\text{ kg/m}^3} = \mathbf{0.341\text{ m}^3/\text{min}}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.341\text{ m}^3/\text{min})}{\pi(110\text{ m/min})}} = 0.063\text{ m} = \mathbf{6.3\text{ cm}}$$

6-102 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions 1 Steady operation under worst conditions is considered. 2 Air is an ideal gas with constant specific heats. 3 Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of $T_{\text{ave}} = (45+60)/2 = 52.5^\circ\text{C}$ is $c_p = 1.0065 \text{ kJ/kg}\cdot^\circ\text{C}$. The gas constant for air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-2).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C , and leave at 60°C .

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{steady}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the required mass flow rate of air to absorb heat at a rate of 100 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}})$$

$$\dot{m} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{100 \text{ W}}{(1006.5 \text{ J/kg}\cdot^\circ\text{C})(60 - 45)^\circ\text{C}} = 0.006624 \text{ kg/s} = 0.397 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.57 \text{ m}^3/\text{min}}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.57 \text{ m}^3/\text{min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1 \text{ cm}}$$

6-103E Electronic devices mounted on a cold plate are cooled by water. The amount of heat generated by the electronic devices is to be determined.

Assumptions 1 Steady operating conditions exist. 2 About 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation. 3 Kinetic and potential energy changes are negligible.

Properties The properties of water at room temperature are $\rho = 62.1 \text{ lbm/ft}^3$ and $c_p = 1.00 \text{ Btu/lbm}\cdot^\circ\text{F}$ (Table A-3E).

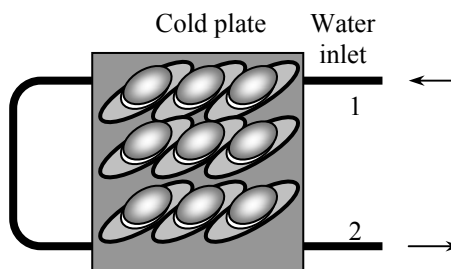
Analysis We take the tubes of the cold plate to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho AV = \rho \frac{\pi D^2}{4} V = (62.1 \text{ lbm/ft}^3) \frac{\pi (0.25/12 \text{ ft})^2}{4} (60 \text{ ft/min}) = 1.270 \text{ lbm/min} = 76.2 \text{ lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) = (76.2 \text{ lbm/h})(1.00 \text{ Btu/lbm}\cdot^\circ\text{F})(105 - 95)^\circ\text{F} = 762 \text{ Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

$$\dot{Q} = \frac{762 \text{ Btu/h}}{0.85} = \mathbf{896 \text{ Btu/h} = 263 \text{ W}}$$

6-104 CD EES The components of an electronic device located in a horizontal duct of rectangular cross section are cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant specific heats at room temperature. 3 Kinetic and potential energy changes are negligible

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-1). The specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(30 + 273) \text{ K}} = 1.165 \text{ kg/m}^3$$

$$\dot{m} = \rho \dot{V} = (1.165 \text{ kg/m}^3)(0.6 \text{ m}^3/\text{min}) = 0.700 \text{ kg/min}$$

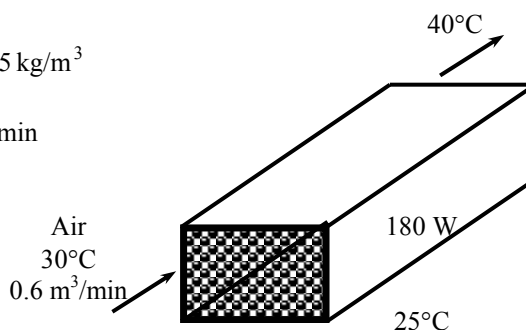
We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{\text{air}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} = (0.700/60 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})(40 - 30)^\circ\text{C} = 0.117 \text{ kW} = 117 \text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63 \text{ W}}$$

6-105 The components of an electronic device located in a horizontal duct of circular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant specific heats at room temperature. 3 Kinetic and potential energy changes are negligible

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-1). The specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(30 + 273) \text{ K}} = 1.165 \text{ kg/m}^3$$

$$\dot{m} = \rho \dot{V} = (1.165 \text{ kg/m}^3)(0.6 \text{ m}^3/\text{min}) = 0.700 \text{ kg/min}$$

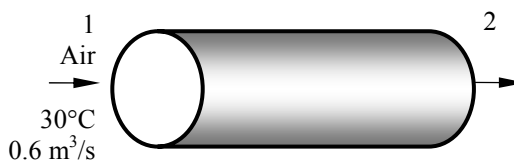
We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{\text{air}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} = (0.700/60 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})(40 - 30)^\circ\text{C} = 0.117 \text{ kW} = 117 \text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63 \text{ W}}$$

6-106E Water is heated in a parabolic solar collector. The required length of parabolic collector is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat loss from the tube is negligible so that the entire solar energy incident on the tube is transferred to the water. 3 Kinetic and potential energy changes are negligible

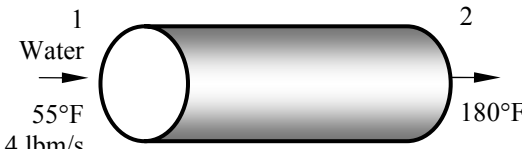
Properties The specific heat of water at room temperature is $c_p = 1.00 \text{ Btu/lbm} \cdot ^\circ\text{F}$ (Table A-2E).

Analysis We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$


Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{\text{total}} = \dot{m} c_p (T_e - T_i) = (4 \text{ lbm/s})(1.00 \text{ Btu/lbm} \cdot ^\circ\text{F})(180 - 55)^\circ\text{F} = 500 \text{ Btu/s} = 1,800,000 \text{ Btu/h}$$

The length of the tube required is

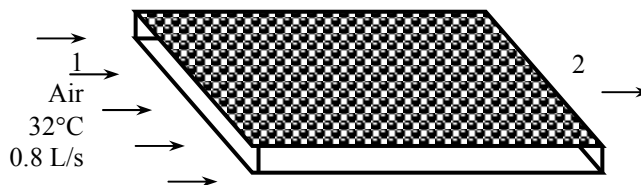
$$L = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{1,800,000 \text{ Btu/h}}{400 \text{ Btu/h} \cdot \text{ft}} = \mathbf{4500 \text{ ft}}$$

6-107 Air enters a hollow-core printed circuit board. The exit temperature of the air is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant specific heats at room temperature. 3 The local atmospheric pressure is 1 atm. 4 Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-1). The specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are



$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(32 + 273) \text{ K}} = 1.16 \text{ kg/m}^3$$

$$\dot{m} = \rho \dot{V} = (1.16 \text{ kg/m}^3)(0.0008 \text{ m}^3/\text{s}) = 0.000928 \text{ kg/s}$$

We take the hollow core to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the exit temperature of air leaving the hollow core becomes

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{\dot{Q}_{\text{in}}}{\dot{m}c_p} = 32^\circ\text{C} + \frac{20 \text{ J/s}}{(0.000928 \text{ kg/s})(1005 \text{ J/kg} \cdot ^\circ\text{C})} = 53.4^\circ\text{C}$$

6-108 A computer is cooled by a fan blowing air through the case of the computer. The required flow rate of the air and the fraction of the temperature rise of air that is due to heat generated by the fan are to be determined.

Assumptions 1 Steady flow conditions exist. 2 Air is an ideal gas with constant specific heats. 3 The pressure of air is 1 atm. 4 Kinetic and potential energy changes are negligible

Properties The specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2).

Analysis (a) We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Noting that the fan power is 25 W and the 8 PCBs transfer a total of 80 W of heat to air, the mass flow rate of air is determined to be

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q}_{\text{in}} + \dot{W}_{\text{in}}}{c_p(T_e - T_i)} = \frac{(8 \times 10) \text{ W} + 25 \text{ W}}{(1005 \text{ J/kg} \cdot ^\circ\text{C})(10^\circ\text{C})} = \mathbf{0.0104 \text{ kg/s}}$$

(b) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor can be determined from

$$\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{25 \text{ W}}{(0.0104 \text{ kg/s})(1005 \text{ J/kg} \cdot ^\circ\text{C})} = 2.4^\circ\text{C}$$

$$f = \frac{2.4^\circ\text{C}}{10^\circ\text{C}} = 0.24 = \mathbf{24\%}$$

6-109 Hot water enters a pipe whose outer surface is exposed to cold air in a basement. The rate of heat loss from the water is to be determined.

Assumptions 1 Steady flow conditions exist. 2 Water is an incompressible substance with constant specific heats. 3 The changes in kinetic and potential energies are negligible.

Properties The properties of water at the average temperature of $(90+88)/2 = 89^\circ\text{C}$ are $\rho = 965 \text{ kg/m}^3$ and $c_p = 4.21 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3).

Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965 \text{ kg/m}^3) \frac{\pi(0.04 \text{ m})^2}{4} (0.8 \text{ m/s}) = 0.970 \text{ kg/s}$$

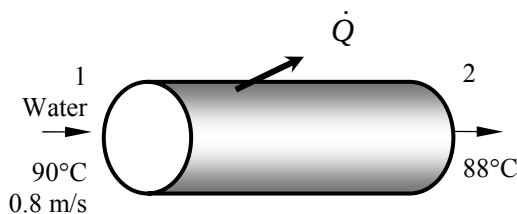
We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer from the hot water to the surrounding air becomes

$$\dot{Q}_{\text{out}} = \dot{m}c_p[T_{\text{in}} - T_{\text{out}}]_{\text{water}} = (0.970 \text{ kg/s})(4.21 \text{ kJ/kg}\cdot^\circ\text{C})(90 - 88)^\circ\text{C} = \mathbf{8.17 \text{ kW}}$$

6-110 EES Problem 6-109 is reconsidered. The effect of the inner pipe diameter on the rate of heat loss as the pipe diameter varies from 1.5 cm to 7.5 cm is to be investigated. The rate of heat loss is to be plotted against the diameter.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns:"

{D = 0.04 [m]}
 $\rho = 965 \text{ [kg/m}^3\text{]}$
 $\text{Vel} = 0.8 \text{ [m/s]}$
 $T_1 = 90 \text{ [C]}$
 $T_2 = 88 \text{ [C]}$
 $C_P = 4.21 \text{ [kJ/kg-C]}$

"Analysis:"

"The mass flow rate of water is:"

$\text{Area} = \pi D^2/4$
 $\dot{m} = \rho \cdot \text{Area} \cdot \text{Vel}$

"We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as"

$$\dot{E}_{\text{dot_in}} - \dot{E}_{\text{dot_out}} = \Delta \dot{E}_{\text{dot_sys}}$$

$$\Delta \dot{E}_{\text{dot_sys}} = 0 \text{ "Steady-flow assumption"}$$

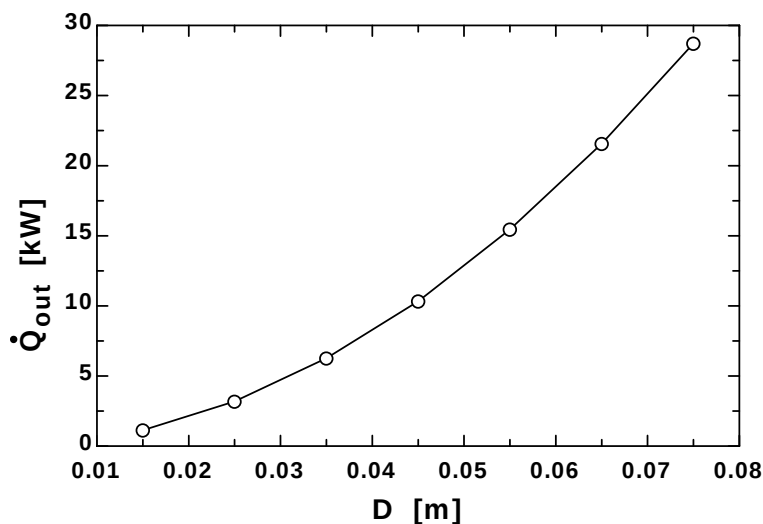
$$\dot{E}_{\text{dot_in}} = \dot{m} \cdot h_{\text{in}}$$

$$\dot{E}_{\text{dot_out}} = \dot{Q}_{\text{dot_out}} + \dot{m} \cdot h_{\text{out}}$$

$$h_{\text{in}} = C_P \cdot T_1$$

$$h_{\text{out}} = C_P \cdot T_2$$

D [m]	$\dot{Q}_{\text{out}}$ [kW]
0.015	1.149
0.025	3.191
0.035	6.254
0.045	10.34
0.055	15.44
0.065	21.57
0.075	28.72



6-111 A room is to be heated by an electric resistance heater placed in a duct in the room. The power rating of the electric heater and the temperature rise of air as it passes through the heater are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant specific heats at room temperature. 3 Kinetic and potential energy changes are negligible. 4 The heating duct is adiabatic, and thus heat transfer through it is negligible. 5 No air leaks in and out of the room.

Properties The gas constant of air is $0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1). The specific heats of air at room temperature are $c_p = 1.005$ and $c_v = 0.718 \text{ kJ/kg} \cdot \text{K}$ (Table A-2).

Analysis (a) The total mass of air in the room is

$$\begin{aligned} V &= 5 \times 6 \times 8 \text{ m}^3 = 240 \text{ m}^3 \\ m &= \frac{P_1 V}{RT_1} = \frac{(98 \text{ kPa})(240 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 284.6 \text{ kg} \end{aligned}$$

We first take the *entire room* as our system, which is a closed system since no mass leaks in or out. The power rating of the electric heater is determined by applying the conservation of energy relation to this constant volume closed system:

$$\begin{aligned} \underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} &= \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\ W_{e,in} + W_{fan,in} - Q_{out} &= \Delta U \quad (\text{since } \Delta KE = \Delta PE = 0) \\ \Delta t(\dot{W}_{e,in} + \dot{W}_{fan,in} - \dot{Q}_{out}) &= mc_{v,avg}(T_2 - T_1) \end{aligned}$$

Solving for the electrical work input gives

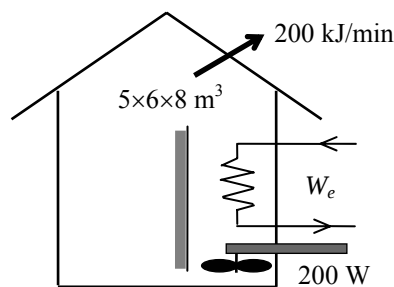
$$\begin{aligned} \dot{W}_{e,in} &= \dot{Q}_{out} - \dot{W}_{fan,in} + mc_v(T_2 - T_1)/\Delta t \\ &= (200/60 \text{ kJ/s}) - (0.2 \text{ kJ/s}) + (284.6 \text{ kg})(0.718 \text{ kJ/kg} \cdot ^\circ\text{C})(25 - 15)^\circ\text{C}/(15 \times 60 \text{ s}) \\ &= \mathbf{5.40 \text{ kW}} \end{aligned}$$

(b) We now take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this adiabatic steady-flow system can be expressed in the rate form as

$$\begin{aligned} \underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} &= \underbrace{\dot{\Delta E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \stackrel{\text{no (steady)}}{=} 0 \\ \dot{E}_{in} &= \dot{E}_{out} \\ \dot{W}_{e,in} + \dot{W}_{fan,in} + \dot{m}h_1 &= \dot{m}h_2 \quad (\text{since } \dot{Q} = \dot{\Delta ke} \cong \dot{\Delta pe} \cong 0) \\ \dot{W}_{e,in} + \dot{W}_{fan,in} &= \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1) \end{aligned}$$

Thus,

$$\Delta T = T_2 - T_1 = \frac{\dot{W}_{e,in} + \dot{W}_{fan,in}}{\dot{m}c_p} = \frac{(5.40 + 0.2) \text{ kJ/s}}{(50/60 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot \text{K})} = \mathbf{6.7^\circ\text{C}}$$



6-112 A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The constant pressure specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot \text{K}$ (Table A-2)

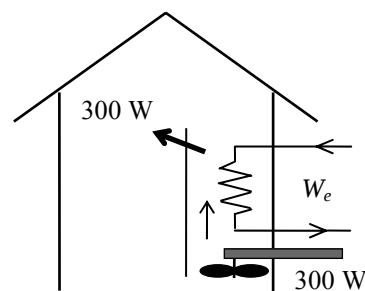
Analysis We take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan},\text{in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan},\text{in}} = \dot{Q}_{\text{out}} + \dot{m}(h_2 - h_1) = \dot{Q}_{\text{out}} + \dot{m}c_p(T_2 - T_1)$$



Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{e,\text{in}} = \dot{Q}_{\text{out}} + \dot{m}c_p\Delta T - \dot{W}_{\text{fan},\text{in}} = (0.3 \text{ kJ/s}) + (0.6 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})(7^\circ\text{C}) - 0.3 \text{ kW} = \mathbf{4.22 \text{ kW}}$$

6-113 A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Air is an ideal gas with constant specific heats at room temperature. 3 Kinetic and potential energy changes are negligible. 4 The power consumed by the fan and the heat losses are negligible.

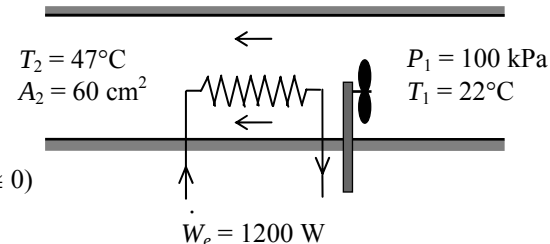
Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The constant pressure specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ (Table A-2)

Analysis We take the *hair dryer* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{e,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{\text{out}} \equiv \Delta ke \equiv \Delta pe \equiv 0)$$

$$\dot{W}_{e,\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$


Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{e,\text{in}}}{c_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(47 - 22)^\circ\text{C}} = 0.04776 \text{ kg/s}$$

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})}{(100 \text{ kPa})} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}\nu_1 = (0.04776 \text{ kg/s})(0.8467 \text{ m}^3/\text{kg}) = \mathbf{0.0404 \text{ m}^3/\text{s}}$$

(b) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320 \text{ K})}{(100 \text{ kPa})} = 0.9184 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(0.04776 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60 \times 10^{-4} \text{ m}^2} = \mathbf{7.31 \text{ m/s}}$$

6-114 EES Problem 6-113 is reconsidered. The effect of the exit cross-sectional area of the hair drier on the exit velocity as the exit area varies from 25 cm² to 75 cm² is to be investigated. The exit velocity is to be plotted against the exit cross-sectional area.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Knowns:"

$$R = 0.287 \text{ [kJPa}\cdot\text{m}^3/\text{kg}\cdot\text{K]}$$

$$P = 100 \text{ [kPa]}$$

$$T_1 = 22 \text{ [}^\circ\text{C]}$$

$$T_2 = 47 \text{ [}^\circ\text{C]}$$

$$\{A_2 = 60 \text{ [cm}^2\}]$$

$$A_1 = 53.35 \text{ [cm}^2\text{]}$$

$$\dot{W}_{\text{ele}} = 1200 \text{ [W]}$$

"Analysis:"

We take the hair dryer as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit. Thus, the energy balance for this steady-flow system can be expressed in the rate form as:"

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{E}_{\text{in}} = \dot{W}_{\text{ele}} \cdot \text{convert}(\text{W}, \text{kW}) + \dot{m}_1 (h_1 + \text{Vel}_1^2/2 \cdot \text{convert}(\text{m}^2/\text{s}^2, \text{kJ/kg}))$$

$$\dot{E}_{\text{out}} = \dot{m}_2 (h_2 + \text{Vel}_2^2/2 \cdot \text{convert}(\text{m}^2/\text{s}^2, \text{kJ/kg}))$$

$$h_2 = \text{enthalpy}(\text{air}, T = T_2)$$

$$h_1 = \text{enthalpy}(\text{air}, T = T_1)$$

"The volume flow rates of air are determined to be:"

$$\dot{V}_1 = \dot{m}_1 v_1$$

$$P v_1 = R(T_1 + 273)$$

$$\dot{V}_2 = \dot{m}_2 v_2$$

$$P v_2 = R(T_2 + 273)$$

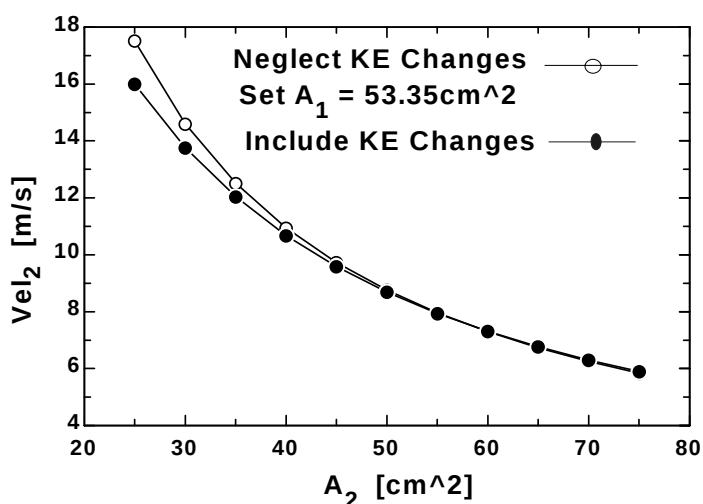
$$\dot{m}_1 = \dot{m}_2$$

$$\text{Vel}_1 = \dot{V}_1 / (A_1 \cdot \text{convert}(\text{cm}^2, \text{m}^2))$$

"(b) The exit velocity of air is determined from the mass balance to be"

$$\text{Vel}_2 = \dot{V}_2 / (A_2 \cdot \text{convert}(\text{cm}^2, \text{m}^2))$$

$A_2 \text{ [cm}^2\text{]}$	$\text{Vel}_2 \text{ [m/s]}$
25	16
30	13.75
35	12.03
40	10.68
45	9.583
50	8.688
55	7.941
60	7.31
65	6.77
70	6.303
75	5.896



6-115 The ducts of a heating system pass through an unheated area. The rate of heat loss from the air in the ducts is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

Properties The constant pressure specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ (Table A-2)

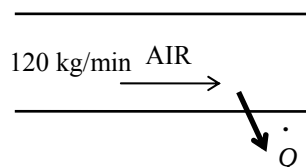
Analysis We take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$



Substituting,

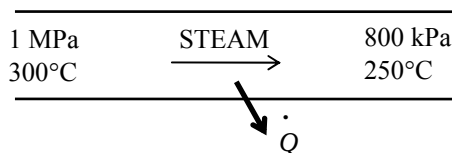
$$\dot{Q}_{\text{out}} = (120 \text{ kg/min})(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(4^\circ\text{C}) = \mathbf{482 \text{ kJ/min}}$$

6-116 Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

Properties From the steam tables (Table A-6),

$$\begin{aligned} P_1 = 1 \text{ MPa} & \left\{ \begin{aligned} v_1 &= 0.25799 \text{ m}^3/\text{kg} \\ h_1 &= 3051.6 \text{ kJ/kg} \end{aligned} \right. \\ P_2 = 800 \text{ kPa} & \left\{ \begin{aligned} h_2 &= 2950.4 \text{ kJ/kg} \end{aligned} \right. \\ T_1 = 300^\circ\text{C} & \\ T_2 = 250^\circ\text{C} & \end{aligned}$$



Analysis (a) The mass flow rate of steam is determined directly from

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.25799 \text{ m}^3/\text{kg}} [\pi (0.06 \text{ m})^2] (2 \text{ m/s}) = \mathbf{0.0877 \text{ kg/s}}$$

(b) We take the *steam pipe* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2)$$

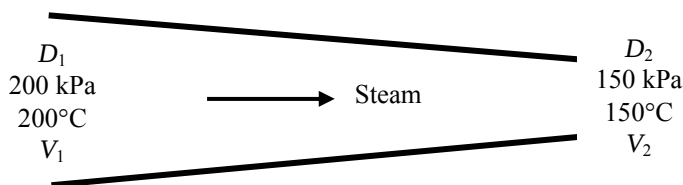
Substituting, the rate of heat loss is determined to be

$$\dot{Q}_{\text{loss}} = (0.0877 \text{ kg/s})(3051.6 - 2950.4) \text{ kJ/kg} = \mathbf{8.87 \text{ kJ/s}}$$

6-117 Steam flows through a non-constant cross-section pipe. The inlet and exit velocities of the steam are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Analysis We take the pipe as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as



Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow A_1 \frac{V_1}{\nu_1} = A_2 \frac{V_2}{\nu_2} \longrightarrow \frac{\pi D_1^2}{4} \frac{V_1}{\nu_1} = \frac{\pi D_2^2}{4} \frac{V_2}{\nu_2}$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_1 = 1.0805 \text{ m}^3/\text{kg} \\ h_1 = 2870.7 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} P_2 = 150 \text{ kPa} \\ T_2 = 150^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_2 = 1.2855 \text{ m}^3/\text{kg} \\ h_2 = 2772.9 \text{ kJ/kg} \end{array}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting,

$$\frac{\pi(1.8 \text{ m})^2}{4} \frac{V_1}{(1.0805 \text{ m}^3/\text{kg})} = \frac{\pi(1.0 \text{ m})^2}{4} \frac{V_2}{(1.2855 \text{ m}^3/\text{kg})} \quad (1)$$

$$2870.7 \text{ kJ/kg} + \frac{V_1^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 2772.9 \text{ kJ/kg} + \frac{V_2^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \quad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

$$V_1 = \mathbf{118.8 \text{ m/s}}$$

$$V_2 = \mathbf{458.0 \text{ m/s}}$$

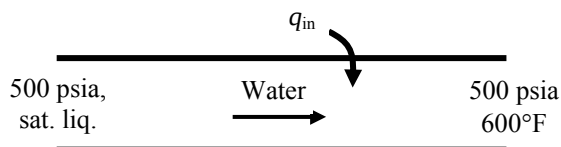
6-118E Saturated liquid water is heated in a steam boiler. The heat transfer per unit mass is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which the water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\begin{aligned}\dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}h_1 + \dot{Q}_{\text{in}} &= \dot{m}h_2 \\ \dot{Q}_{\text{in}} &= \dot{m}(h_2 - h_1) \\ q_{\text{in}} &= h_2 - h_1\end{aligned}$$



The enthalpies of water at the inlet and exit of the boiler are (Table A-5E, A-6E).

$$\left. \begin{array}{l} P_1 = 500 \text{ psia} \\ x = 0 \end{array} \right\} h_1 \cong h_f @ 500 \text{ psia} = 449.51 \text{ Btu/lbm}$$

$$\left. \begin{array}{l} P_2 = 500 \text{ psia} \\ T_2 = 600^\circ\text{F} \end{array} \right\} h_2 = 1298.6 \text{ Btu/lbm}$$

Substituting,

$$q_{\text{in}} = 1298.6 - 449.51 = \mathbf{849.1 \text{ Btu/lbm}}$$

6-119 R-134a is condensed in a condenser. The heat transfer per unit mass is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which R-134a is condensed as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

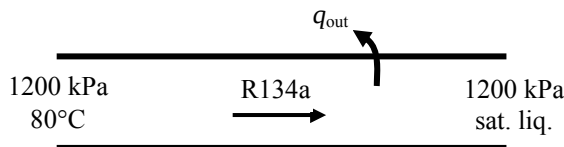
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2 + \dot{Q}_{\text{out}}$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2)$$

$$q_{\text{out}} = h_1 - h_2$$



The enthalpies of R-134a at the inlet and exit of the condenser are (Table A-12, A-13).

$$\left. \begin{array}{l} P_1 = 1200 \text{ kPa} \\ T_1 = 80^\circ\text{C} \end{array} \right\} h_1 = 311.39 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 1200 \text{ kPa} \\ x = 0 \end{array} \right\} h_2 \cong h_f @ 1200 \text{ kPa} = 117.77 \text{ kJ/kg}$$

Substituting,

$$q_{\text{out}} = 311.39 - 117.77 = \mathbf{193.6 \text{ kJ/kg}}$$

6-120 Water is heated at constant pressure so that it changes a state from saturated liquid to saturated vapor. The heat transfer per unit mass is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

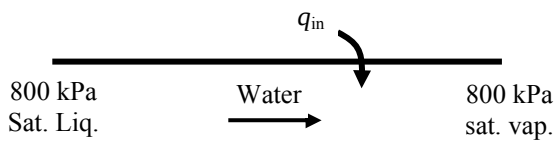
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{Q}_{\text{in}} = \dot{m}h_2$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1)$$

$$q_{\text{in}} = h_2 - h_1 = h_{fg}$$



where $h_{fg} @ 800 \text{ kPa} = 2047.5 \text{ kJ/kg}$ (Table A-5)

Thus,

$$q_{\text{in}} = \mathbf{2047.5 \text{ kJ/kg}}$$

Charging and Discharging Processes

6-121 Helium flows from a supply line to an initially evacuated tank. The flow work of the helium in the supply line and the final temperature of the helium in the tank are to be determined.

Properties The properties of helium are $R = 2.0769 \text{ kJ/kg}\cdot\text{K}$, $c_p = 5.1926 \text{ kJ/kg}\cdot\text{K}$, $c_v = 3.1156 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis The flow work is determined from its definition but we first determine the specific volume

$$\nu = \frac{RT_{\text{line}}}{P} = \frac{(2.0769 \text{ kJ/kg}\cdot\text{K})(120 + 273 \text{ K})}{(200 \text{ kPa})} = 4.0811 \text{ m}^3/\text{kg}$$

$$w_{\text{flow}} = P\nu = (200 \text{ kPa})(4.0811 \text{ m}^3/\text{kg}) = \mathbf{816.2 \text{ kJ/kg}}$$

Noting that the flow work in the supply line is converted to sensible internal energy in the tank, the final helium temperature in the tank is determined as follows

$$u_{\text{tank}} = h_{\text{line}}$$

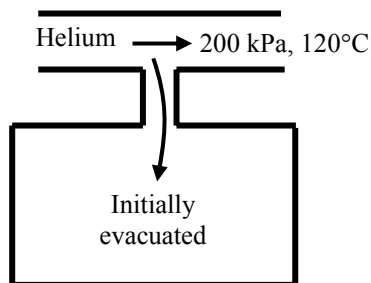
$$h_{\text{line}} = c_p T_{\text{line}} = (5.1926 \text{ kJ/kg}\cdot\text{K})(120 + 273 \text{ K}) = 2040.7 \text{ kJ/kg}$$

$$u_{\text{tank}} = c_v T_{\text{tank}} \longrightarrow 2040.7 \text{ kJ/kg} = (3.1156 \text{ kJ/kg}\cdot\text{K})T_{\text{tank}} \longrightarrow T_{\text{tank}} = \mathbf{655.0 \text{ K}}$$

Alternative Solution: Noting the definition of specific heat ratio, the final temperature in the tank can also be determined from

$$T_{\text{tank}} = kT_{\text{line}} = 1.667(120 + 273 \text{ K}) = \mathbf{655.1 \text{ K}}$$

which is practically the same result.



6-122 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).

Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

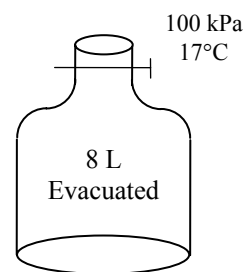
Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$



Combining the two balances:

$$Q_{\text{in}} = m_2 (u_2 - h_i)$$

where

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(100 \text{ kPa})(0.008 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})} = 0.0096 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{matrix} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{matrix}$$

Substituting,

$$Q_{\text{in}} = (0.0096 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -0.8 \text{ kJ}$$

or

$$Q_{\text{out}} = \mathbf{0.8 \text{ kJ}}$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

6-123 An insulated rigid tank is evacuated. A valve is opened, and air is allowed to fill the tank until mechanical equilibrium is established. The final temperature in the tank is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. 2 Air is an ideal gas with constant specific heats. 3 Kinetic and potential energies are negligible. 4 There are no work interactions involved. 5 The device is adiabatic and thus heat transfer is negligible.

Properties The specific heat ratio for air at room temperature is $k = 1.4$ (Table A-2).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

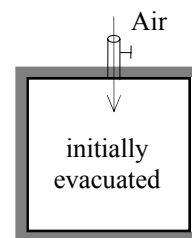
$$m_i h_i = m_2 u_2 \quad (\text{since } Q \cong W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$

Combining the two balances:

$$u_2 = h_i \rightarrow c_v T_2 = c_p T_i \rightarrow T_2 = (c_p / c_v) T_i = k T_i$$

Substituting,

$$T_2 = 1.4 \times 290 \text{ K} = 406 \text{ K} = \mathbf{133^\circ \text{C}}$$



6-124 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until mechanical equilibrium is established. The mass of air that entered and the amount of heat transfer are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the tank (will be verified).

Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The properties of air are (Table A-21)

$$\begin{aligned} T_i &= 295 \text{ K} \longrightarrow h_i = 295.17 \text{ kJ/kg} \\ T_1 &= 295 \text{ K} \longrightarrow u_1 = 210.49 \text{ kJ/kg} \\ T_2 &= 350 \text{ K} \longrightarrow u_2 = 250.02 \text{ kJ/kg} \end{aligned}$$

Analysis (a) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

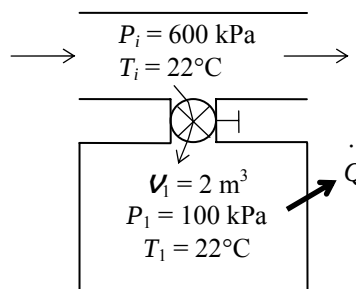
Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$



The initial and the final masses in the tank are

$$\begin{aligned} m_1 &= \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 2.362 \text{ kg} \\ m_2 &= \frac{P_2 V}{RT_2} = \frac{(600 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(350 \text{ K})} = 11.946 \text{ kg} \end{aligned}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = 11.946 - 2.362 = \mathbf{9.584 \text{ kg}}$$

(b) The heat transfer during this process is determined from

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(9.584 \text{ kg})(295.17 \text{ kJ/kg}) + (11.946 \text{ kg})(250.02 \text{ kJ/kg}) - (2.362 \text{ kg})(210.49 \text{ kJ/kg}) \\ &= -339 \text{ kJ} \rightarrow Q_{\text{out}} = \mathbf{339 \text{ kJ}} \end{aligned}$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

6-125 A rigid tank initially contains saturated R-134a liquid-vapor mixture. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The final temperature in the tank, the mass of R-134a that entered, and the heat transfer are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\begin{aligned} T_1 = 8^\circ\text{C} \quad \left\{ \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.0007887 + 0.7 \times (0.052762 - 0.0007887) = 0.03717 \text{ m}^3/\text{kg} \\ x_1 = 0.7 \end{array} \right. \quad \left\{ \begin{array}{l} u_1 = u_f + x_1 u_{fg} = 62.39 + 0.7 \times 172.19 = 182.92 \text{ kJ/kg} \end{array} \right. \\ P_2 = 800 \text{ kPa} \quad \left\{ \begin{array}{l} v_2 = v_{g@800 \text{ kPa}} = 0.02562 \text{ m}^3/\text{kg} \\ \text{sat. vapor} \end{array} \right. \quad \left\{ \begin{array}{l} u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array} \right. \\ P_i = 1.0 \text{ MPa} \quad \left\{ \begin{array}{l} h_i = 335.06 \text{ kJ/kg} \\ T_i = 100^\circ\text{C} \end{array} \right. \end{aligned}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(a) The tank contains saturated vapor at the final state at 800 kPa, and thus the final temperature is the saturation temperature at this pressure,

$$T_2 = T_{\text{sat @ 800 kPa}} = \mathbf{31.31^\circ\text{C}}$$

(b) The initial and the final masses in the tank are

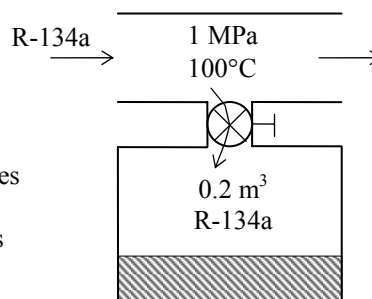
$$\begin{aligned} m_1 &= \frac{v}{v_1} = \frac{0.2 \text{ m}^3}{0.03717 \text{ m}^3/\text{kg}} = 5.38 \text{ kg} \\ m_2 &= \frac{v}{v_2} = \frac{0.2 \text{ m}^3}{0.02562 \text{ m}^3/\text{kg}} = 7.81 \text{ kg} \end{aligned}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 7.81 - 5.38 = \mathbf{2.43 \text{ kg}}$$

(c) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(2.43 \text{ kg})(335.06 \text{ kJ/kg}) + (7.81 \text{ kg})(246.79 \text{ kJ/kg}) - (5.38 \text{ kg})(182.92 \text{ kJ/kg}) \\ &= \mathbf{130 \text{ kJ}} \end{aligned}$$



6-126 A cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

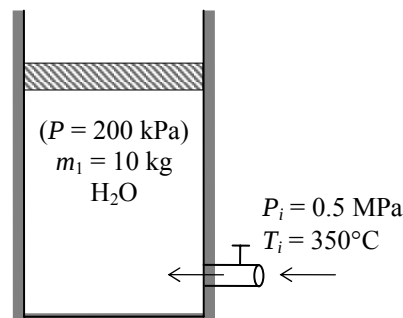
Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.

Properties The properties of steam are (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.6 \end{array} \right\} h_1 = h_f + x_1 h_{fg} \\ = 504.71 + 0.6 \times 2201.6 = 1825.6 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} h_2 = h_{g@200 \text{ kPa}} = 2706.3 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_i = 0.5 \text{ MPa} \\ T_i = 350^\circ\text{C} \end{array} \right\} h_i = 3168.1 \text{ kJ/kg}$$



Analysis (a) The cylinder contains saturated vapor at the final state at a pressure of 200 kPa, thus the final temperature in the cylinder must be

$$T_2 = T_{\text{sat @ 200 kPa}} = \mathbf{120.2^\circ\text{C}}$$

(b) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

$$\text{Mass balance: } m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\ m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

$$\text{Combining the two relations gives } 0 = W_{\text{b,out}} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$$

$$\text{or, } 0 = -(m_2 - m_1)h_i + m_2 h_2 - m_1 h_1$$

since the boundary work and ΔU combine into ΔH for constant pressure expansion and compression processes. Solving for m_2 and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3168.1 - 1825.6) \text{ kJ/kg}}{(3168.1 - 2706.3) \text{ kJ/kg}} (10 \text{ kg}) = 29.07 \text{ kg}$$

Thus,

$$m_i = m_2 - m_1 = 29.07 - 10 = \mathbf{19.07 \text{ kg}}$$

6-127E A scuba diver's air tank is to be filled with air from a compressed air line. The temperature and mass in the tank at the final state are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The gas constant of air is $0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$ (Table A-1E). The specific heats of air at room temperature are $c_p = 0.240 \text{ Btu/lbm} \cdot \text{R}$ and $c_v = 0.171 \text{ Btu/lbm} \cdot \text{R}$ (Table A-2Ea).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

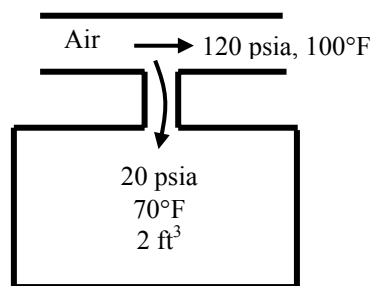
$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$



Combining the two balances:

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(20 \text{ psia})(2 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(70 + 460 \text{ R})} = 0.2038 \text{ lbm}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(120 \text{ psia})(2 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})T_2} = \frac{647.9}{T_2}$$

Substituting,

$$\left(\frac{647.9}{T_2} - 0.2038 \right) (0.24)(560) = \frac{647.9}{T_2} (0.171)T_2 - (0.2038)(0.171)(530)$$

whose solution is

$$T_2 = \mathbf{727.4 \text{ R} = 267.4^\circ \text{F}}$$

The final mass is then

$$m_2 = \frac{647.9}{T_2} = \frac{647.9}{727.4} = \mathbf{0.890 \text{ lbm}}$$

6-128 R-134a from a tank is discharged to an air-conditioning line in an isothermal process. The final quality of the R-134a in the tank and the total heat transfer are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the exit remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

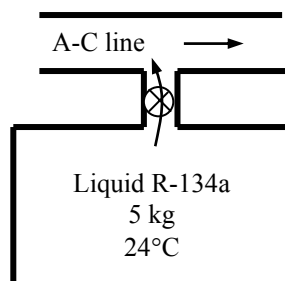
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$\begin{aligned} m_{\text{in}} - m_{\text{out}} &= \Delta m_{\text{system}} \\ -m_e &= m_2 - m_1 \\ m_e &= m_1 - m_2 \end{aligned}$$

Energy balance:

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ Q_{\text{in}} - m_e h_e &= m_2 u_2 - m_1 u_1 \\ Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + m_e h_e \end{aligned}$$



Combining the two balances:

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The initial state properties of R-134a in the tank are

$$\left. \begin{array}{l} T_1 = 24^\circ\text{C} \\ x = 0 \end{array} \right\} \begin{array}{l} v_1 = 0.0008261 \text{ m}^3/\text{kg} \\ u_1 = 84.44 \text{ kJ/kg} \\ h_e = 84.98 \text{ kJ/kg} \end{array} \quad (\text{Table A-11})$$

Note that we assumed that the refrigerant leaving the tank is at saturated liquid state, and found the exiting enthalpy accordingly. The volume of the tank is

$$V = m_1 v_1 = (5 \text{ kg})(0.0008261 \text{ m}^3/\text{kg}) = 0.004131 \text{ m}^3$$

The final specific volume in the container is

$$v_2 = \frac{V}{m_2} = \frac{0.004131 \text{ m}^3}{0.25 \text{ kg}} = 0.01652 \text{ m}^3/\text{kg}$$

The final state is now fixed. The properties at this state are (Table A-11)

$$\left. \begin{array}{l} T_2 = 24^\circ\text{C} \\ v_2 = 0.01652 \text{ m}^3/\text{kg} \end{array} \right\} \begin{array}{l} x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.01652 - 0.0008261}{0.031834 - 0.0008261} = \mathbf{0.5061} \\ u_2 = u_f + x_2 u_{fg} = 84.44 \text{ kJ/kg} + (0.5061)(158.65 \text{ kJ/kg}) = 164.73 \text{ kJ/kg} \end{array}$$

Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e \\ &= (0.25 \text{ kg})(164.73 \text{ kJ/kg}) - (5 \text{ kg})(84.44 \text{ kJ/kg}) + (4.75 \text{ kg})(84.98 \text{ kJ/kg}) \\ &= \mathbf{22.64 \text{ kJ}} \end{aligned}$$

6-129E Oxygen is supplied to a medical facility from 10 compressed oxygen tanks in an isothermal process. The mass of oxygen used and the total heat transfer to the tanks are to be determined.

Assumptions 1 This is an unsteady process but it can be analyzed as a uniform-flow process. **2** Oxygen is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved.

Properties The gas constant of oxygen is $0.3353 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$ (Table A-1E). The specific heats of oxygen at room temperature are $c_p = 0.219 \text{ Btu/lbm} \cdot \text{R}$ and $c_v = 0.157 \text{ Btu/lbm} \cdot \text{R}$ (Table A-2Ea).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$

$$-m_e = m_2 - m_1$$

$$m_e = m_1 - m_2$$

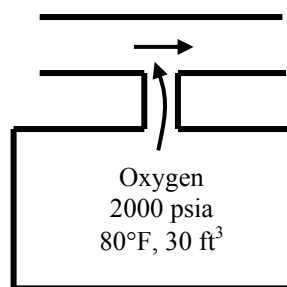
Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + m_e h_e$$

$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$



Combining the two balances:

$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses, and the mass used are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(2000 \text{ psia})(30 \text{ ft}^3)}{(0.3353 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(80 + 460 \text{ R})} = 331.4 \text{ lbm}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(100 \text{ psia})(30 \text{ ft}^3)}{(0.3353 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(80 + 460 \text{ R})} = 16.57 \text{ lbm}$$

$$m_e = m_1 - m_2 = 331.4 - 16.57 = \mathbf{314.8 \text{ lbm}}$$

Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e \\ &= (16.57)(0.157)(540) - (331.4)(0.157)(540) + (314.8)(0.219)(540) \\ &= \mathbf{10,540 \text{ Btu}} \end{aligned}$$

6-130 The air in an insulated, rigid compressed-air tank is released until the pressure in the tank reduces to a specified value. The final temperature of the air in the tank is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The specific heats of air at room temperature are $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ and $c_v = 0.718 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

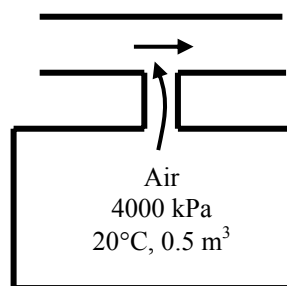
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$\begin{aligned} m_{\text{in}} - m_{\text{out}} &= \Delta m_{\text{system}} \\ -m_e &= m_2 - m_1 \\ m_e &= m_1 - m_2 \end{aligned}$$

Energy balance:

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ -m_e h_e &= m_2 u_2 - m_1 u_1 \\ 0 &= m_2 u_2 - m_1 u_1 + m_e h_e \\ 0 &= m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e \end{aligned}$$



Combining the two balances:

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses are given by

$$\begin{aligned} m_1 &= \frac{P_1 V}{RT_1} = \frac{(4000 \text{ kPa})(0.5 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})} = 23.78 \text{ kg} \\ m_2 &= \frac{P_2 V}{RT_2} = \frac{(2000 \text{ kPa})(0.5 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{3484}{T_2} \end{aligned}$$

The temperature of air leaving the tank changes from the initial temperature in the tank to the final temperature during the discharging process. We assume that the temperature of the air leaving the tank is the average of initial and final temperatures in the tank. Substituting into the energy balance equation gives

$$\begin{aligned} 0 &= m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e \\ 0 &= \frac{3484}{T_2} (0.718) T_2 - (23.78)(0.718)(293) + \left(23.78 - \frac{3484}{T_2} \right) (1.005) \left(\frac{293 + T_2}{2} \right) \end{aligned}$$

whose solution by trial-error or by an equation solver such as EES is

$$T_2 = \mathbf{241\text{K}} = \mathbf{-32^\circ\text{C}}$$

6-131E Steam is supplied from a line to a weighted piston-cylinder device. The final temperature (and quality if appropriate) of the steam in the piston cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$

$$m_i = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

The boundary work is determined from

$$W_{b,\text{out}} = P(\mathcal{V}_2 - \mathcal{V}_1) = P(m_2 \nu_2 - m_1 \nu_1) = P m_2 \nu_2$$

Substituting, the energy balance equation simplifies into

$$P m_2 \nu_2 = m_2 (h_i - u_2)$$

$$P \nu_2 = h_i - u_2$$

The enthalpy of steam at the inlet is

$$\left. \begin{array}{l} P_1 = 300 \text{ psia} \\ T_1 = 450^\circ\text{F} \end{array} \right\} h_i = 1226.4 \text{ Btu/lbm} \quad (\text{Table A - 6E})$$

Substituting this value into the energy balance equation and using an iterative solution of this equation gives (or better yet using EES software)

$$T_2 = \mathbf{425.1^\circ\text{F}}$$

$$u_2 = 1135.5 \text{ Btu/lbm}$$

$$\nu_2 = 2.4575 \text{ ft}^3/\text{lbm}$$

The final mass is

$$m_2 = \frac{\mathcal{V}_2}{\nu_2} = \frac{10 \text{ ft}^3}{2.4575 \text{ ft}^3/\text{lbm}} = 4.069 \text{ lbm}$$

and the work produced is

$$W_{b,\text{out}} = P \mathcal{V}_2 = (200 \text{ psia})(10 \text{ ft}^3) \left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3} \right) = \mathbf{370.1 \text{ Btu}}$$

6-132E Oxygen is supplied from a line to a weighted piston-cylinder device. The final temperature of the oxygen in the piston cylinder and the total work produced as the device is filled are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic. **4** Oxygen is an ideal gas with constant specific heats.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$

$$m_i = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

The boundary work is determined from

$$W_{b,\text{out}} = P(\mathcal{V}_2 - \mathcal{V}_1) = P(m_2 \nu_2 - m_1 \nu_1) = P m_2 \nu_2$$

Substituting, the energy balance equation simplifies into

$$P m_2 \nu_2 = m_2 (h_i - u_2)$$

$$P \nu_2 = h_i - u_2$$

$$R T_2 = c_p T_i - c_v T_2$$

Solving for the final temperature,

$$R T_2 = c_p T_i - c_v T_2 \longrightarrow T_2 = \frac{c_p}{R + c_v} T_i = \frac{c_p}{c_p} T_i = T_i = \mathbf{450^\circ\text{F}}$$

The work produced is

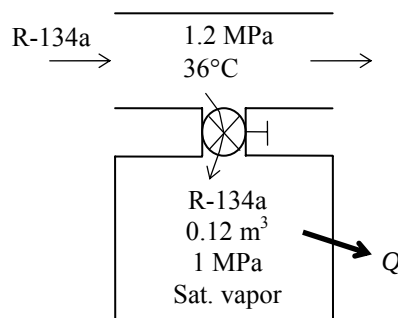
$$W_{b,\text{out}} = P \mathcal{V}_2 = (300 \text{ psia})(10 \text{ ft}^3) \left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3} \right) = \mathbf{555 \text{ Btu}}$$

6-133 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered and the heat transfer are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\begin{aligned} P_1 = 1 \text{ MPa} \quad \left\{ \begin{array}{l} \nu_1 = \nu_{g@1 \text{ MPa}} = 0.02031 \text{ m}^3/\text{kg} \\ u_1 = u_{g@1 \text{ MPa}} = 250.68 \text{ kJ/kg} \end{array} \right. \\ \text{sat. vapor} \\ P_2 = 1.2 \text{ MPa} \quad \left\{ \begin{array}{l} \nu_2 = \nu_{f@1.2 \text{ MPa}} = 0.0008934 \text{ m}^3/\text{kg} \\ u_2 = u_{f@1.2 \text{ MPa}} = 116.70 \text{ kJ/kg} \end{array} \right. \\ \text{sat. liquid} \\ P_i = 1.2 \text{ MPa} \quad \left\{ \begin{array}{l} h_i = h_{f@36^\circ\text{C}} = 102.30 \text{ kJ/kg} \\ T_i = 36^\circ\text{C} \end{array} \right. \end{aligned}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(a) The initial and the final masses in the tank are

$$m_1 = \frac{\nu_1}{\nu_1} = \frac{0.12 \text{ m}^3}{0.02031 \text{ m}^3/\text{kg}} = 5.91 \text{ kg}$$

$$m_2 = \frac{\nu_2}{\nu_2} = \frac{0.12 \text{ m}^3}{0.0008934 \text{ m}^3/\text{kg}} = 134.31 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 134.31 - 5.91 = \mathbf{128.4 \text{ kg}}$$

(c) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(128.4 \text{ kg})(102.30 \text{ kJ/kg}) + (134.31 \text{ kg})(116.70 \text{ kJ/kg}) - (5.91 \text{ kg})(250.68 \text{ kJ/kg}) \\ &= \mathbf{1057 \text{ kJ}} \end{aligned}$$

6-134 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\left. \begin{array}{l} T_1 = 200^\circ\text{C} \\ \text{sat. liquid} \end{array} \right\} \begin{array}{l} \nu_1 = \nu_{f@200^\circ\text{C}} = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = u_{f@200^\circ\text{C}} = 850.46 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} T_e = 200^\circ\text{C} \\ \text{sat. liquid} \end{array} \right\} h_e = h_{f@200^\circ\text{C}} = 852.26 \text{ kJ/kg}$$

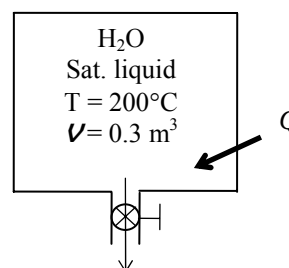
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$



The initial and the final masses in the tank are

$$m_1 = \frac{\nu_1}{\nu_f} = \frac{0.3 \text{ m}^3}{0.001157 \text{ m}^3/\text{kg}} = 259.4 \text{ kg}$$

$$m_2 = \frac{1}{2} m_1 = \frac{1}{2} (259.4 \text{ kg}) = 129.7 \text{ kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 259.4 - 129.7 = 129.7 \text{ kg}$$

Now we determine the final internal energy,

$$\nu_2 = \frac{\nu}{m_2} = \frac{0.3 \text{ m}^3}{129.7 \text{ kg}} = 0.002313 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{\nu_2 - \nu_f}{\nu_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left. \begin{array}{l} T_2 = 200^\circ\text{C} \\ x_2 = 0.009171 \end{array} \right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46 \text{ kJ/kg}$$

Then the heat transfer during this process is determined from the energy balance by substitution to be

$$Q = (129.7 \text{ kg})(852.26 \text{ kJ/kg}) + (129.7 \text{ kg})(866.46 \text{ kJ/kg}) - (259.4 \text{ kg})(850.46 \text{ kJ/kg})$$

$$= \mathbf{2308 \text{ kJ}}$$

6-135 A rigid tank initially contains saturated liquid-vapor mixture of refrigerant-134a. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank at constant pressure until no liquid remains inside. The amount of heat transfer is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. 2 Kinetic and potential energies are negligible. 3 There are no work interactions involved. 4 The direction of heat transfer is to the tank (will be verified).

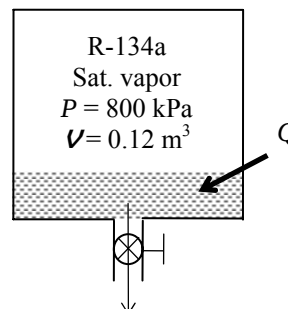
Properties The properties of R-134a are (Tables A-11 through A-13)

$$P_1 = 800 \text{ kPa} \rightarrow \nu_f = 0.0008458 \text{ m}^3/\text{kg}, \nu_g = 0.025621 \text{ m}^3/\text{kg}$$

$$u_f = 94.79 \text{ kJ/kg}, u_g = 246.79 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} \nu_2 = \nu_{g@800 \text{ kPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array} \right\} h_e = h_{f@800 \text{ kPa}} = 95.47 \text{ kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{\nu_f}{\nu_f} + \frac{\nu_g}{\nu_g} = \frac{0.12 \times 0.25 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.12 \times 0.75 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.47 + 3.513 = 38.98 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (35.47)(94.79) + (3.513)(246.79) = 4229.2 \text{ kJ}$$

$$m_2 = \frac{\nu}{\nu_2} = \frac{0.12 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 4.684 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.98 - 4.684 = 34.30 \text{ kg}$$

$$Q_{\text{in}} = (34.30 \text{ kg})(95.47 \text{ kJ/kg}) + (4.684 \text{ kg})(246.79 \text{ kJ/kg}) - 4229 \text{ kJ} = \mathbf{201.2 \text{ kJ}}$$

6-136E A rigid tank initially contains saturated liquid-vapor mixture of R-134a. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until all the liquid in the tank disappears. The amount of heat transfer is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

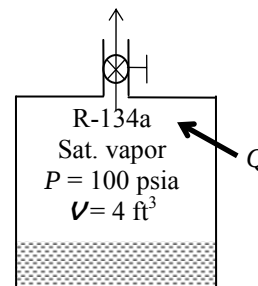
Properties The properties of R-134a are (Tables A-11E through A-13E)

$$P_1 = 100 \text{ psia} \rightarrow \nu_f = 0.01332 \text{ ft}^3/\text{lbm}, \nu_g = 0.4776 \text{ ft}^3/\text{lbm}$$

$$u_f = 37.623 \text{ Btu/lbm}, u_g = 104.99 \text{ Btu/lbm}$$

$$P_2 = 100 \text{ psia} \left\{ \begin{array}{l} \nu_2 = \nu_{g@100 \text{ psia}} = 0.4776 \text{ ft}^3/\text{lbm} \\ u_2 = u_{g@100 \text{ psia}} = 104.99 \text{ Btu/lbm} \end{array} \right.$$

$$P_e = 100 \text{ psia} \left\{ \begin{array}{l} h_e = h_{g@100 \text{ psia}} = 113.83 \text{ Btu/lbm} \end{array} \right.$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{\nu_f}{\nu_f} + \frac{\nu_g}{\nu_g} = \frac{4 \times 0.2 \text{ ft}^3}{0.01332 \text{ ft}^3/\text{lbm}} + \frac{4 \times 0.8 \text{ ft}^3}{0.4776 \text{ ft}^3/\text{lbm}} = 60.04 + 6.70 = 66.74 \text{ lbm}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (60.04)(37.623) + (6.70)(104.99) = 2962 \text{ Btu}$$

$$m_2 = \frac{\nu}{\nu_2} = \frac{4 \text{ ft}^3}{0.4776 \text{ ft}^3/\text{lbm}} = 8.375 \text{ lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 66.74 - 8.375 = 58.37 \text{ lbm}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1$$

$$= (58.37 \text{ lbm})(113.83 \text{ Btu/lbm}) + (8.375 \text{ lbm})(104.99 \text{ Btu/lbm}) - 2962 \text{ Btu}$$

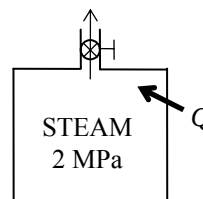
$$= \mathbf{4561 \text{ Btu}}$$

6-137 A rigid tank initially contains superheated steam. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until the temperature rises to 500°C. The amount of heat transfer is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the steam leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\begin{aligned} P_1 = 2 \text{ MPa} \quad & \left\{ \begin{array}{l} \nu_1 = 0.12551 \text{ m}^3/\text{kg} \\ T_1 = 300^\circ\text{C} \end{array} \right. \quad \left\{ \begin{array}{l} u_1 = 2773.2 \text{ kJ/kg}, \quad h_1 = 3024.2 \text{ kJ/kg} \end{array} \right. \\ P_2 = 2 \text{ MPa} \quad & \left\{ \begin{array}{l} \nu_2 = 0.17568 \text{ m}^3/\text{kg} \\ T_2 = 500^\circ\text{C} \end{array} \right. \quad \left\{ \begin{array}{l} u_2 = 3116.9 \text{ kJ/kg}, \quad h_2 = 3468.3 \text{ kJ/kg} \end{array} \right. \end{aligned}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The state and thus the enthalpy of the steam leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values. Thus,

$$h_e \cong \frac{h_1 + h_2}{2} = \frac{3024.2 + 3468.3 \text{ kJ/kg}}{2} = 3246.2 \text{ kJ/kg}$$

The initial and the final masses in the tank are

$$\begin{aligned} m_1 &= \frac{\nu_1}{\nu_1} = \frac{0.2 \text{ m}^3}{0.12551 \text{ m}^3/\text{kg}} = 1.594 \text{ kg} \\ m_2 &= \frac{\nu_2}{\nu_2} = \frac{0.2 \text{ m}^3}{0.17568 \text{ m}^3/\text{kg}} = 1.138 \text{ kg} \end{aligned}$$

Then from the mass and energy balance relations,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456 \text{ kg}$$

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456 \text{ kg})(3246.2 \text{ kJ/kg}) + (1.138 \text{ kg})(3116.9 \text{ kJ/kg}) - (1.594 \text{ kg})(2773.2 \text{ kJ/kg}) \\ &= \mathbf{606.8 \text{ kJ}} \end{aligned}$$

6-138 Steam is supplied from a line to a piston-cylinder device equipped with a spring. The final temperature (and quality if appropriate) of the steam in the cylinder and the total work produced as the device is filled are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

$$\text{Mass balance: } m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \longrightarrow m_i = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

Because of the spring, the relation between the pressure and volume is a linear relation. According to the data in the problem statement,

$$P - 300 = \frac{2700}{5} \nu$$

The final vapor volume is then

$$\nu_2 = \frac{5}{2700} (1500 - 300) = 2.222 \text{ m}^3$$

The work needed to compress the spring is

$$W_{b,\text{out}} = \int P d\nu = \int_0^{\nu_2} \left(\frac{2700}{5} \nu + 300 \right) d\nu = \frac{2700}{2 \times 5} \nu_2^2 + 300 \nu_2 = 270 \times 2.222^2 + 300 \times 2.222 = \mathbf{2000 \text{ kJ}}$$

The enthalpy of steam at the inlet is

$$\left. \begin{array}{l} P_1 = 1500 \text{ kPa} \\ T_1 = 200^\circ\text{C} \end{array} \right\} h_i = 2796.0 \text{ kJ/kg} \quad (\text{Table A - 6})$$

Substituting the information found into the energy balance equation gives

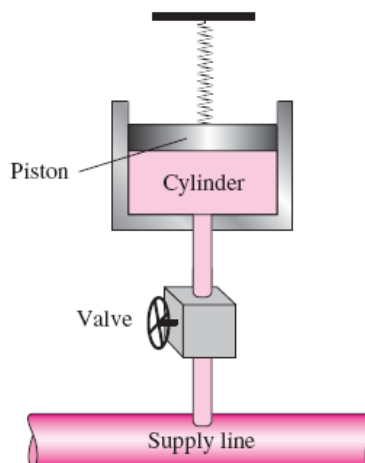
$$W_{b,\text{out}} = m_2 (h_i - u_2) \longrightarrow W_{b,\text{out}} = \frac{\nu_2}{\nu_2} (h_i - u_2) \longrightarrow 2000 = \frac{2.222}{\nu_2} (2796.0 - u_2)$$

Using an iterative solution of this equation with steam tables gives (or better yet using EES software)

$$T_2 = \mathbf{233.2^\circ\text{C}}$$

$$u_2 = 2664.8 \text{ kJ/kg}$$

$$\nu_2 = 0.1458 \text{ m}^3/\text{kg}$$



6-139 Air is supplied from a line to a piston-cylinder device equipped with a spring. The final temperature of the steam in the cylinder and the total work produced as the device is filled are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The specific heats of air at room temperature are $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ and $c_v = 0.718 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

$$\text{Mass balance: } m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \longrightarrow m_i = m_2$$

Energy balance:

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ m_i h_i - W_{b,\text{out}} &= m_2 u_2 \\ W_{b,\text{out}} &= m_i h_i - m_2 u_2 \end{aligned}$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

Because of the spring, the relation between the pressure and volume is a linear relation. According to the data in the problem statement,

$$P - 300 = \frac{2700}{5} \nu$$

The final air volume is then

$$\nu_2 = \frac{5}{2700} (2000 - 300) = 3.148 \text{ m}^3$$

The work needed to compress the spring is

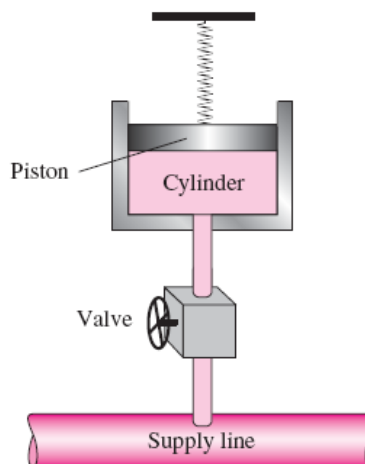
$$W_{b,\text{out}} = \int P d\nu = \int_0^{\nu_2} \left(\frac{2700}{5} \nu + 300 \right) d\nu = \frac{2700}{2 \times 5} \nu_2^2 + 300 \nu_2 = 270 \times 3.148^2 + 300 \times 3.148 = \mathbf{3620 \text{ kJ}}$$

Substituting the information found into the energy balance equation gives

$$\begin{aligned} W_{b,\text{out}} &= m_2 (h_i - u_2) \\ W_{b,\text{out}} &= \frac{P_2 \nu_2}{RT_2} (c_p T_i - c_v T_2) \\ 3620 &= \frac{2000 \times 3.148}{(0.287) T_2} (1.005 \times 523 - 0.718 \times T_2) \end{aligned}$$

The final temperature is then

$$T_2 = \mathbf{595.2 \text{ K} = 322.2^\circ\text{C}}$$



6-140 A hot-air balloon is considered. The final volume of the balloon and work produced by the air inside the balloon as it expands the balloon skin are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There is no heat transfer.

Properties The gas constant of air is $0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis The specific volume of the air at the entrance and exit, and in the balloon is

$$\nu = \frac{RT}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(35 + 273 \text{ K})}{100 \text{ kPa}} = 0.8840 \text{ m}^3/\text{kg}$$

The mass flow rate at the entrance is then

$$\dot{m}_i = \frac{A_i V_i}{\nu} = \frac{(1 \text{ m}^2)(2 \text{ m/s})}{0.8840 \text{ m}^3/\text{kg}} = 2.262 \text{ kg/s}$$

while that at the outlet is

$$\dot{m}_e = \frac{A_e V_e}{\nu} = \frac{(0.5 \text{ m}^2)(1 \text{ m/s})}{0.8840 \text{ m}^3/\text{kg}} = 0.5656 \text{ kg/s}$$

Applying a mass balance to the balloon,

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$

$$m_i - m_e = m_2 - m_1$$

$$m_2 - m_1 = (\dot{m}_i - \dot{m}_e) \Delta t = [(2.262 - 0.5656) \text{ kg/s}](2 \times 60 \text{ s}) = 203.6 \text{ kg}$$

The volume in the balloon then changes by the amount

$$\Delta \nu = (m_2 - m_1) \nu = (203.6 \text{ kg})(0.8840 \text{ m}^3/\text{kg}) = 180 \text{ m}^3$$

and the final volume of the balloon is

$$\nu_2 = \nu_1 + \Delta \nu = 75 + 180 = \mathbf{255 \text{ m}^3}$$

In order to push back the boundary of the balloon against the surrounding atmosphere, the amount of work that must be done is

$$W_{b,\text{out}} = P \Delta \nu = (100 \text{ kPa})(180 \text{ m}^3) \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{18,000 \text{ kJ}}$$

6-141 An insulated rigid tank initially contains helium gas at high pressure. A valve is opened, and half of the mass of helium is allowed to escape. The final temperature and pressure in the tank are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is $k=1.667$ (Table A-2).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

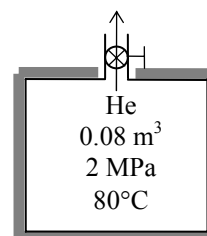
Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

$$m_2 = \frac{1}{2} m_1 \text{ (given)} \rightarrow m_e = m_2 = \frac{1}{2} m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong Q \cong ke \cong pe \cong 0)$$



Note that the state and thus the enthalpy of helium leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values.

Combining the mass and energy balances: $0 = \frac{1}{2} m_1 h_e + \frac{1}{2} m_1 u_2 - m_1 u_1$

Dividing by $m_1/2$ $0 = h_e + u_2 - 2u_1$ or $0 = c_p \frac{T_1 + T_2}{2} + c_v T_2 - 2c_v T_1$

Dividing by c_v : $0 = k(T_1 + T_2) + 2T_2 - 4T_1$ since $k = c_p / c_v$

Solving for T_2 : $T_2 = \frac{(4-k)}{(2+k)} T_1 = \frac{(4-1.667)}{(2+1.667)} (353 \text{ K}) = 225 \text{ K}$

The final pressure in the tank is

$$\frac{P_1 V}{P_2 V} = \frac{m_1 R T_1}{m_2 R T_2} \rightarrow P_2 = \frac{m_2 T_2}{m_1 T_1} P_1 = \frac{1}{2} \frac{225}{353} (2000 \text{ kPa}) = 637 \text{ kPa}$$

6-142E An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 30 psia. The amount of electrical work transferred is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.3704 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R}$ (Table A-1E). The properties of air are (Table A-21E)

$$\begin{aligned} T_i &= 580 \text{ R} & \longrightarrow & h_i = 138.66 \text{ Btu / lbm} \\ T_1 &= 580 \text{ R} & \longrightarrow & u_1 = 98.90 \text{ Btu / lbm} \\ T_2 &= 580 \text{ R} & \longrightarrow & u_2 = 98.90 \text{ Btu / lbm} \end{aligned}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

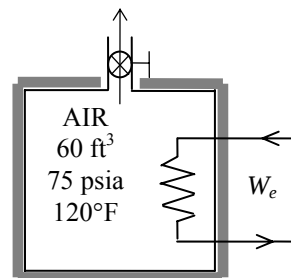
Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



The initial and the final masses of air in the tank are

$$\begin{aligned} m_1 &= \frac{P_1 V}{RT_1} = \frac{(75 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R})(580 \text{ R})} = 20.95 \text{ lbm} \\ m_2 &= \frac{P_2 V}{RT_2} = \frac{(30 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3 / \text{lbm} \cdot \text{R})(580 \text{ R})} = 8.38 \text{ lbm} \end{aligned}$$

Then from the mass and energy balances,

$$\begin{aligned} m_e &= m_1 - m_2 = 20.95 - 8.38 = 12.57 \text{ lbm} \\ W_{e,\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (12.57 \text{ lbm})(138.66 \text{ Btu/lbm}) + (8.38 \text{ lbm})(98.90 \text{ Btu/lbm}) - (20.95 \text{ lbm})(98.90 \text{ Btu/lbm}) \\ &= \mathbf{500 \text{ Btu}} \end{aligned}$$

6-143 A vertical cylinder initially contains air at room temperature. Now a valve is opened, and air is allowed to escape at constant pressure and temperature until the volume of the cylinder goes down by half. The amount air that left the cylinder and the amount of heat transfer are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. 2 Kinetic and potential energies are negligible. 3 There are no work interactions other than boundary work. 4 Air is an ideal gas with constant specific heats. 5 The direction of heat transfer is to the cylinder (will be verified).

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis (a) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

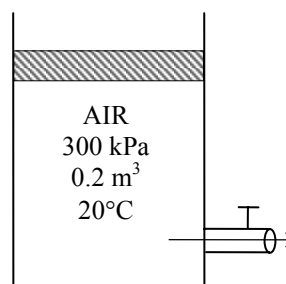
Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$



The initial and the final masses of air in the cylinder are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(300 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 0.714 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(300 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 0.357 \text{ kg} = \frac{1}{2} m_1$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 0.714 - 0.357 = \mathbf{0.357 \text{ kg}}$$

(b) This is a constant pressure process, and thus the W_b and the ΔU terms can be combined into ΔH to yield

$$Q = m_e h_e + m_2 h_2 - m_1 h_1$$

Noting that the temperature of the air remains constant during this process, we have

$$h_i = h_1 = h_2 = h.$$

Also,

$$m_e = m_2 = \frac{1}{2} m_1.$$

Thus,

$$Q = \left(\frac{1}{2} m_1 + \frac{1}{2} m_1 - m_1\right) h = \mathbf{0}$$

6-144 A balloon is initially filled with helium gas at atmospheric conditions. The tank is connected to a supply line, and helium is allowed to enter the balloon until the pressure rises from 100 to 150 kPa. The final temperature in the balloon is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Helium is an ideal gas with constant specific heats. **3** The expansion process is quasi-equilibrium. **4** Kinetic and potential energies are negligible. **5** There are no work interactions involved other than boundary work. **6** Heat transfer is negligible.

Properties The gas constant of helium is $R = 2.0769 \text{ kJ/kg}\cdot\text{K}$ (Table A-1). The specific heats of helium are $c_p = 5.1926$ and $c_v = 3.1156 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$m_i h_i = W_{b,\text{out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(65 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 10.61 \text{ kg}$$

$$\frac{P_1}{P_2} = \frac{V_1}{V_2} \rightarrow V_2 = \frac{P_2}{P_1} V_1 = \frac{150 \text{ kPa}}{100 \text{ kPa}} (65 \text{ m}^3) = 97.5 \text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(150 \text{ kPa})(97.5 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2 \text{ K})} = \frac{7041.74}{T_2} \text{ kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = \frac{7041.74}{T_2} - 10.61 \text{ kg}$$

Noting that P varies linearly with V , the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2} (V_2 - V_1) = \frac{(100 + 150) \text{ kPa}}{2} (97.5 - 65) \text{ m}^3 = 4062.5 \text{ kJ}$$

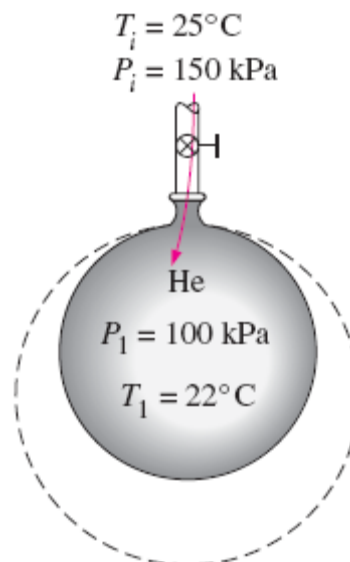
Using specific heats, the energy balance relation reduces to

$$W_{b,\text{out}} = m_i c_p T_i - m_2 c_v T_2 + m_1 c_v T_1$$

Substituting,

$$4062.5 = \left(\frac{7041.74}{T_2} - 10.61 \right) (5.1926)(298) - \frac{7041.74}{T_2} (3.1156)T_2 + (10.61)(3.1156)(295)$$

It yields $T_2 = \mathbf{333.6 \text{ K}}$



6-145 An insulated piston-cylinder device with a linear spring is applying force to the piston. A valve at the bottom of the cylinder is opened, and refrigerant is allowed to escape. The amount of refrigerant that escapes and the final temperature of the refrigerant are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process assuming that the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The initial properties of R-134a are (Tables A-11 through A-13)

$$\left. \begin{array}{l} P_1 = 1.2 \text{ MPa} \\ T_1 = 120^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.02423 \text{ m}^3/\text{kg} \\ u_1 = 325.03 \text{ kJ/kg} \\ h_1 = 354.11 \text{ kJ/kg} \end{array}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

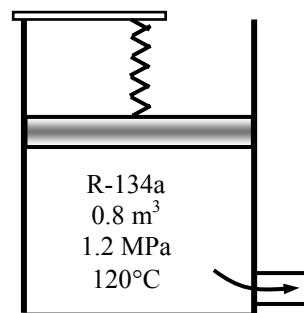
$$W_{b,\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial mass and the relations for the final and exiting masses are

$$m_1 = \frac{V_1}{v_1} = \frac{0.8 \text{ m}^3}{0.02423 \text{ m}^3/\text{kg}} = 33.02 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.5 \text{ m}^3}{v_2}$$

$$m_e = m_1 - m_2 = 33.02 - \frac{0.5 \text{ m}^3}{v_2}$$



Noting that the spring is linear, the boundary work can be determined from

$$W_{b,\text{in}} = \frac{P_1 + P_2}{2} (V_1 - V_2) = \frac{(1200 + 600) \text{ kPa}}{2} (0.8 - 0.5) \text{ m}^3 = 270 \text{ kJ}$$

Substituting the energy balance,

$$270 - \left(33.02 - \frac{0.5 \text{ m}^3}{v_2} \right) h_e = \left(\frac{0.5 \text{ m}^3}{v_2} \right) u_2 - (33.02 \text{ kg})(325.03 \text{ kJ/kg}) \quad (\text{Eq. 1})$$

where the enthalpy of exiting fluid is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder. That is,

$$h_e = \frac{h_1 + h_2}{2} = \frac{(354.11 \text{ kJ/kg}) + h_2}{2}$$

Final state properties of the refrigerant (h_2 , u_2 , and v_2) are all functions of final pressure (known) and temperature (unknown). The solution may be obtained by a trial-error approach by trying different final state temperatures until Eq. (1) is satisfied. Or solving the above equations simultaneously using an equation solver with built-in thermodynamic functions such as EES, we obtain

$$T_2 = 96.8^\circ\text{C}, \quad m_e = 22.47 \text{ kg}, \quad h_2 = 336.20 \text{ kJ/kg},$$

$$u_2 = 307.77 \text{ kJ/kg}, \quad v_2 = 0.04739 \text{ m}^3/\text{kg}, \quad m_2 = 10.55 \text{ kg}$$

6-146 Steam at a specified state is allowed to enter a piston-cylinder device in which steam undergoes a constant pressure expansion process. The amount of mass that enters and the amount of heat transfer are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The properties of steam at various states are (Tables A-4 through A-6)

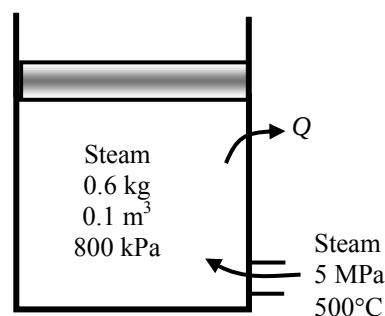
$$\nu_1 = \frac{\nu_1}{m_1} = \frac{0.1 \text{ m}^3}{0.6 \text{ kg}} = 0.16667 \text{ m}^3/\text{kg}$$

$$P_2 = P_1$$

$$\left. \begin{array}{l} P_1 = 800 \text{ kPa} \\ \nu_1 = 0.16667 \text{ m}^3/\text{kg} \end{array} \right\} u_1 = 2004.4 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 250^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_2 = 0.29321 \text{ m}^3/\text{kg} \\ u_2 = 2715.9 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} P_i = 5 \text{ MPa} \\ T_i = 500^\circ\text{C} \end{array} \right\} h_i = 3434.7 \text{ kJ/kg}$$



Analysis (a) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{\text{b,out}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

Noting that the pressure remains constant, the boundary work is determined from

$$W_{\text{b,out}} = P(\nu_2 - \nu_1) = (800 \text{ kPa})(2 \times 0.1 - 0.1) \text{ m}^3 = 80 \text{ kJ}$$

The final mass and the mass that has entered are

$$m_2 = \frac{\nu_2}{\nu_2} = \frac{0.2 \text{ m}^3}{0.29321 \text{ m}^3/\text{kg}} = 0.682 \text{ kg}$$

$$m_i = m_2 - m_1 = 0.682 - 0.6 = \mathbf{0.082 \text{ kg}}$$

(b) Finally, substituting into energy balance equation

$$Q_{\text{in}} - 80 \text{ kJ} + (0.082 \text{ kg})(3434.7 \text{ kJ/kg}) = (0.682 \text{ kg})(2715.9 \text{ kJ/kg}) - (0.6 \text{ kg})(2004.4 \text{ kJ/kg})$$

$$Q_{\text{in}} = \mathbf{447.9 \text{ kJ}}$$

Review Problems

6-147 A water tank open to the atmosphere is initially filled with water. The tank discharges to the atmosphere through a long pipe connected to a valve. The initial discharge velocity from the tank and the time required to empty the tank are to be determined.

Assumptions 1 The flow is incompressible. 2 The draining pipe is horizontal. 3 The tank is considered to be empty when the water level drops to the center of the valve.

Analysis (a) Substituting the known quantities, the discharge velocity can be expressed as

$$V = \sqrt{\frac{2gz}{1.5 + fL/D}} = \sqrt{\frac{2gz}{1.5 + 0.015(100\text{ m})/(0.10\text{ m})}} = \sqrt{0.1212gz}$$

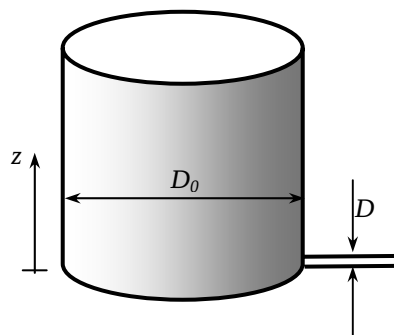
Then the initial discharge velocity becomes

$$V_1 = \sqrt{0.1212gz_1} = \sqrt{0.1212(9.81\text{ m/s}^2)(2\text{ m})} = \mathbf{1.54\text{ m/s}}$$

where z is the water height relative to the center of the orifice at that time.

(b) The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V} = A_{\text{pipe}} V_2 = \frac{\pi D^2}{4} \sqrt{0.1212gz}$$



Then the amount of water that flows through the pipe during a differential time interval dt is

$$dV = \dot{V}dt = \frac{\pi D^2}{4} \sqrt{0.1212gz} dt \quad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\text{tank}}(-dz) = -\frac{\pi D_0^2}{4} dz \quad (2)$$

where dz is the change in the water level in the tank during dt . (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4} \sqrt{0.1212gz} dt = -\frac{\pi D_0^2}{4} dz \rightarrow dt = -\frac{D_0^2}{D^2} \frac{dz}{\sqrt{0.1212gz}} = -\frac{D_0^2}{D^2 \sqrt{0.1212g}} z^{-\frac{1}{2}} dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2 \sqrt{0.1212g}} \int_{z=z_1}^0 z^{-1/2} dz \rightarrow t_f = -\frac{D_0^2}{D^2 \sqrt{0.1212g}} \left[\frac{z^{\frac{1}{2}}}{\frac{1}{2}} \right]_{z_1}^0 = \frac{2D_0^2}{D^2 \sqrt{0.1212g}} z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{2D_0^2}{D^2} \sqrt{\frac{z_1}{0.1212g}} = \frac{2(10\text{ m})^2}{(0.1\text{ m})^2} \sqrt{\frac{2\text{ m}}{0.1212(9.81\text{ m/s}^2)}} = 25,940\text{ s} = \mathbf{7.21\text{ h}}$$

Discussion The draining time can be shortened considerably by installing a pump in the pipe.

6-148 The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

Assumptions **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.

Analysis The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is

$$\dot{V}_e = A_e V_e = (\pi D^2/4) V_e = [\pi(0.05 \text{ m})^2/4](5 \text{ m/s}) = 0.00982 \text{ m}^3/\text{s}$$

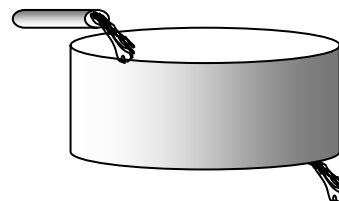
The rate of accumulation of water in the pool is equal to the cross-section of the pool times the rate at which the water level rises,

$$\frac{dV_{\text{pool}}}{dt} = A_{\text{cross-section}} V_{\text{level}} = (3 \text{ m} \times 4 \text{ m})(0.015 \text{ m/min}) = 0.18 \text{ m}^3/\text{min} = 0.00300 \text{ m}^3/\text{s}$$

Substituting, the rate at which water is supplied to the pool is determined to be

$$\dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e = 0.003 + 0.00982 = \mathbf{0.01282 \text{ m}^3/\text{s}}$$

Therefore, water is supplied at a rate of $0.01282 \text{ m}^3/\text{s} = 12.82 \text{ L/s}$.



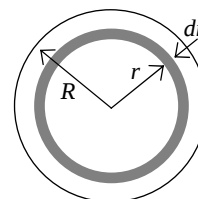
6-149 A fluid is flowing in a circular pipe. A relation is to be obtained for the average fluid velocity in terms of $V(r)$, R , and r .

Analysis Choosing a circular ring of area $dA = 2\pi r dr$ as our differential area, the mass flow rate through a cross-sectional area can be expressed as

$$\dot{m} = \int_A \rho V(r) dA = \int_0^R \rho V(r) 2\pi r dr$$

Solving for V_{avg} ,

$$V_{\text{avg}} = \frac{2}{R^2} \int_0^R V(r) r dr$$



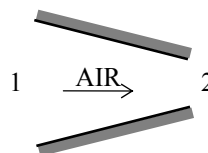
6-150 Air is accelerated in a nozzle. The density of air at the nozzle exit is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 4.18 kg/m^3 at the inlet.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\begin{aligned}\dot{m}_1 &= \dot{m}_2 \\ \rho_1 A_1 V_1 &= \rho_2 A_2 V_2 \\ \rho_2 &= \frac{A_1}{A_2} \frac{V_1}{V_2} \rho_1 = 2 \frac{120 \text{ m/s}}{380 \text{ m/s}} (4.18 \text{ kg/m}^3) = \mathbf{2.64 \text{ kg/m}^3}\end{aligned}$$



Discussion Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

6-151 An air compressor consumes 4.5 kW of power to compress a specified rate of air. The flow work required by the compressor is to be compared to the power used to increase the pressure of the air.

Assumptions 1 Flow through the compressor is steady. 2 Air is an ideal gas.

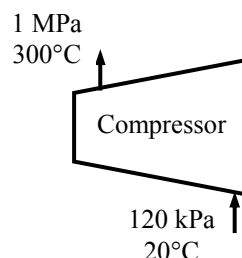
Properties The gas constant of air is $0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis The specific volume of the air at the inlet is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(20 + 273 \text{ K})}{120 \text{ kPa}} = 0.7008 \text{ m}^3/\text{kg}$$

The mass flow rate of the air is

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.010 \text{ m}^3/\text{s}}{0.7008 \text{ m}^3/\text{kg}} = 0.01427 \text{ kg/s}$$



Combining the flow work expression with the ideal gas equation of state gives the flow work as

$$w_{\text{flow}} = P_2 \nu_2 - P_1 \nu_1 = R(T_2 - T_1) = (0.287 \text{ kJ/kg} \cdot \text{K})(300 - 20) \text{ K} = 80.36 \text{ kJ/kg}$$

The flow power is

$$\dot{W}_{\text{flow}} = \dot{m} w_{\text{flow}} = (0.01427 \text{ kg/s})(80.36 \text{ kJ/kg}) = \mathbf{1.147 \text{ kW}}$$

The remainder of compressor power input is used to increase the pressure of the air:

$$\dot{W} = \dot{W}_{\text{total, in}} - \dot{W}_{\text{flow}} = 4.5 - 1.147 = \mathbf{3.353 \text{ kW}}$$

6-152 Steam expands in a turbine whose power production is 9000 kW. The rate of heat lost from the turbine is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the steam tables (Tables A-6 and A-4)

$$\left. \begin{array}{l} P_1 = 1.6 \text{ MPa} \\ T_1 = 350^\circ\text{C} \end{array} \right\} h_1 = 3146.0 \text{ kJ/kg}$$

$$\left. \begin{array}{l} T_2 = 30^\circ\text{C} \\ x_2 = 1 \end{array} \right\} h_2 = 2555.6 \text{ kJ/kg}$$

Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that there is one inlet and one exit the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\phi=0 \text{ (steady)}}{=} 0$$

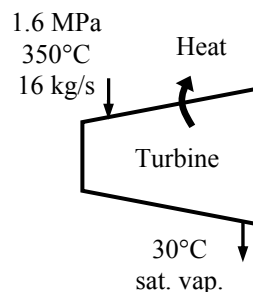
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{W}_{\text{out}} + \dot{Q}_{\text{out}}$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2) - \dot{W}_{\text{out}}$$

Substituting,

$$\dot{Q}_{\text{out}} = (16 \text{ kg/s})(3146.0 - 2555.6) \text{ kJ/kg} - 9000 \text{ kW} = \mathbf{446.4 \text{ kW}}$$



6-153E Nitrogen gas flows through a long, constant-diameter adiabatic pipe. The velocities at the inlet and exit are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Nitrogen is an ideal gas with constant specific heats. 3 Potential energy changes are negligible. 4 There are no work interactions. 5 There is no heat transfer from the nitrogen.

Properties The specific heat of nitrogen at the room temperature is $c_p = 0.248 \text{ Btu/lbm}\cdot\text{R}$ (Table A-2Ea).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the pipe as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

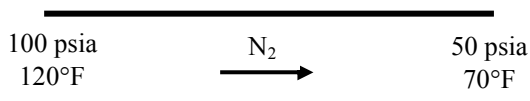
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{No (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$\frac{V_1^2 - V_2^2}{2} = c_p(T_2 - T_1)$$



Combining the mass balance and ideal gas equation of state yields

$$\begin{aligned} \dot{m}_1 &= \dot{m}_2 \\ \frac{A_1 V_1}{v_1} &= \frac{A_2 V_2}{v_2} \\ V_2 &= \frac{A_1}{A_2} \frac{v_2}{v_1} V_1 = \frac{v_2}{v_1} V_1 = \frac{T_2}{T_1} \frac{P_1}{P_2} V_1 \end{aligned}$$

Substituting this expression for V_2 into the energy balance equation gives

$$V_1 = \left[\frac{2c_p(T_2 - T_1)}{1 - \left(\frac{T_2}{T_1} \frac{P_1}{P_2} \right)^2} \right]^{0.5} = \left[\frac{2(0.248)(70 - 120) \left(\frac{25,037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right)}{1 - \left(\frac{530}{580} \frac{100}{50} \right)^2} \right]^{0.5} = \mathbf{515 \text{ ft/s}}$$

The velocity at the exit is

$$V_2 = \frac{T_2}{T_1} \frac{P_1}{P_2} V_1 = \frac{530}{580} \frac{100}{50} 515 = \mathbf{941 \text{ ft/s}}$$

6-154 Water at a specified rate is heated by an electrical heater. The current is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The heat losses from the water is negligible.

Properties The specific heat and the density of water are taken to be $c_p = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ and $\rho = 1 \text{ kg/L}$ (Table A-3).

Analysis We take the pipe in which water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

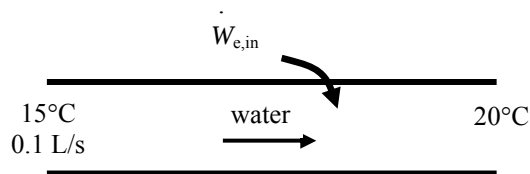
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{e,in}} = \dot{m}h_2$$

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1)$$

$$\mathbf{VI} = \dot{m}c_p(T_2 - T_1)$$



The mass flow rate of the water is

$$\dot{m} = \rho \dot{V} = (1 \text{ kg/L})(0.1 \text{ L/s}) = 0.1 \text{ kg/s}$$

Substituting into the energy balance equation and solving for the current gives

$$I = \frac{\dot{m}c_p(T_2 - T_1)}{\mathbf{V}} = \frac{(0.1 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot\text{K})(20 - 15)\text{K}}{110 \text{ V}} \left(\frac{1000 \text{ VI}}{1 \text{ kJ/s}} \right) = \mathbf{19.0 \text{ A}}$$

6-155 Steam flows in an insulated pipe. The mass flow rate of the steam and the speed of the steam at the pipe outlet are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions.

Analysis We take the pipe in which steam flows as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\begin{array}{ccc} \dot{E}_{\text{in}} = \dot{E}_{\text{out}} & & \\ \dot{m}h_1 = \dot{m}h_2 & \begin{array}{ccc} \text{1400 kPa, 350°C} & \text{Water} & \text{1000 kPa} \\ D=0.15 \text{ m, 10 m/s} & \longrightarrow & D=0.1 \text{ m} \end{array} & \\ h_1 = h_2 & & \end{array}$$

The properties of the steam at the inlet and exit are (Table A-6)

$$\left. \begin{array}{l} P = 1400 \text{ kPa} \\ T = 350^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.20029 \text{ m}^3/\text{kg} \\ h_1 = 3150.1 \text{ kJ/kg} \end{array}$$

$$\left. \begin{array}{l} P_2 = 1000 \text{ kPa} \\ h_2 = h_1 = 3150.1 \text{ kJ/kg} \end{array} \right\} v_2 = 0.28064 \text{ m}^3/\text{kg}$$

The mass flow rate is

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{\pi D_1^2}{4} \frac{V_1}{v_1} = \frac{\pi (0.15 \text{ m})^2}{4} \frac{10 \text{ m/s}}{0.20029 \text{ m}^3/\text{kg}} = \mathbf{0.8823 \text{ kg/s}}$$

The outlet velocity will then be

$$V_2 = \frac{\dot{m} v_2}{A_2} = \frac{4 \dot{m} v_2}{\pi D_2^2} = \frac{4(0.8823 \text{ kg/s})(0.28064 \text{ m}^3/\text{kg})}{\pi (0.10 \text{ m})^2} = \mathbf{31.53 \text{ m/s}}$$

6-156 The mass flow rate of a compressed air line is divided into two equal streams by a T-fitting in the line. The velocity of the air at the outlets and the rate of change of flow energy (flow power) across the T-fitting are to be determined.

Assumptions 1 Air is an ideal gas with constant specific heats. 2 The flow is steady. 3 Since the outlets are identical, it is presumed that the flow divides evenly between the two.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1).

Analysis The specific volumes of air at the inlet and outlets are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(40 + 273 \text{ K})}{1600 \text{ kPa}} = 0.05614 \text{ m}^3/\text{kg}$$

$$\nu_2 = \nu_3 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(36 + 273 \text{ K})}{1400 \text{ kPa}} = 0.06335 \text{ m}^3/\text{kg}$$

Assuming an even division of the inlet flow rate, the mass balance can be written as

$$\frac{A_1 V_1}{\nu_1} = 2 \frac{A_2 V_2}{\nu_2} \longrightarrow V_2 = V_3 = \frac{A_1}{A_2} \frac{\nu_2}{\nu_1} \frac{V_1}{2} = \frac{0.06335}{0.05614} \frac{50}{2} = \mathbf{28.21 \text{ m/s}}$$

The mass flow rate at the inlet is

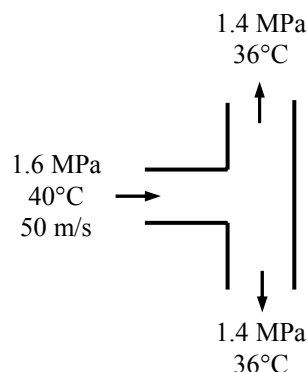
$$\dot{m}_1 = \frac{A_1 V_1}{\nu_1} = \frac{\pi D_1^2}{4} \frac{V_1}{\nu_1} = \frac{\pi (0.025 \text{ m})^2}{4} \frac{50 \text{ m/s}}{0.05614 \text{ m}^3/\text{kg}} = 0.4372 \text{ kg/s}$$

while that at the outlets is

$$\dot{m}_2 = \dot{m}_3 = \frac{\dot{m}_1}{2} = \frac{0.4372 \text{ kg/s}}{2} = 0.2186 \text{ kg/s}$$

Substituting the above results into the flow power expression produces

$$\begin{aligned} \dot{W}_{\text{flow}} &= 2\dot{m}_2 P_2 \nu_2 - \dot{m}_1 P_1 \nu_1 \\ &= 2(0.2186 \text{ kg/s})(1400 \text{ kPa})(0.06335 \text{ m}^3/\text{kg}) - (0.4372 \text{ kg/s})(1600 \text{ kPa})(0.05614 \text{ m}^3/\text{kg}) \\ &= \mathbf{-0.496 \text{ kW}} \end{aligned}$$

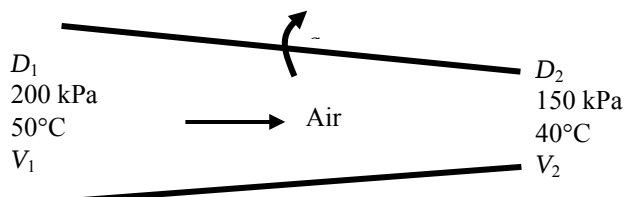


6-157 Air flows through a non-constant cross-section pipe. The inlet and exit velocities of the air are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Potential energy change is negligible. 3 There are no work interactions. 4 The device is adiabatic and thus heat transfer is negligible. 5 Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$. Also, $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ for air at room temperature (Table A-2)

Analysis We take the pipe as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as



Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \longrightarrow \frac{P_1}{RT_1} \frac{\pi D_1^2}{4} V_1 = \frac{P_2}{RT_2} \frac{\pi D_2^2}{4} V_2 \longrightarrow \frac{P_1}{T_1} D_1^2 V_1 = \frac{P_2}{T_2} D_2^2 V_2$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0 \quad \text{since } \dot{W} \cong \Delta p e \cong 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

or

$$c_p T_1 + \frac{V_1^2}{2} = c_p T_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting given values into mass and energy balance equations

$$\left(\frac{200 \text{ kPa}}{323 \text{ K}} \right) (1.8 \text{ m})^2 V_1 = \left(\frac{150 \text{ kPa}}{313 \text{ K}} \right) (1.0 \text{ m})^2 V_2 \quad (1)$$

$$(1.005 \text{ kJ/kg}\cdot\text{K})(323 \text{ K}) + \frac{V_1^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = (1.005 \text{ kJ/kg}\cdot\text{K})(313 \text{ K}) + \frac{V_2^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) + 3.3 \text{ kJ/kg} \quad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

$$V_1 = \mathbf{28.6 \text{ m/s}}$$

$$V_2 = \mathbf{120 \text{ m/s}}$$

6-158 Cold water enters a steam generator at 20°C, and leaves as saturated vapor at $T_{\text{sat}} = 150^\circ\text{C}$. The fraction of heat used to preheat the liquid water from 20°C to saturation temperature of 150°C is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the steam generator are negligible. 3 The specific heat of water is constant at the average temperature.

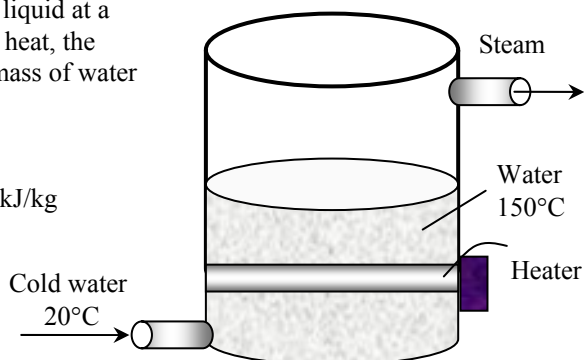
Properties The heat of vaporization of water at 150°C is $h_{\text{fg}} = 2113.8 \text{ kJ/kg}$ (Table A-4), and the specific heat of liquid water is $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3).

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature. Using the average specific heat, the amount of heat transfer needed to preheat a unit mass of water from 20°C to 150°C is

$$q_{\text{preheating}} = c\Delta T \\ = (4.18 \text{ kJ/kg}\cdot^\circ\text{C})(150 - 20)^\circ\text{C} = 543.4 \text{ kJ/kg}$$

and

$$q_{\text{total}} = q_{\text{boiling}} + q_{\text{preheating}} \\ = 2113.8 + 543.4 = 2657.2 \text{ kJ/kg}$$



Therefore, the fraction of heat used to preheat the water is

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{543.4}{2657.2} = \mathbf{0.205} \text{ (or } \mathbf{20.5\%})$$

6-159 Cold water enters a steam generator at 20°C and is boiled, and leaves as saturated vapor at boiler pressure. The boiler pressure at which the amount of heat needed to preheat the water to saturation temperature that is equal to the heat of vaporization is to be determined.

Assumptions Heat losses from the steam generator are negligible.

Properties The enthalpy of liquid water at 20°C is 83.91 kJ/kg. Other properties needed to solve this problem are the heat of vaporization h_{fg} and the enthalpy of saturated liquid at the specified temperatures, and they can be obtained from Table A-4.

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature, and Δh represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

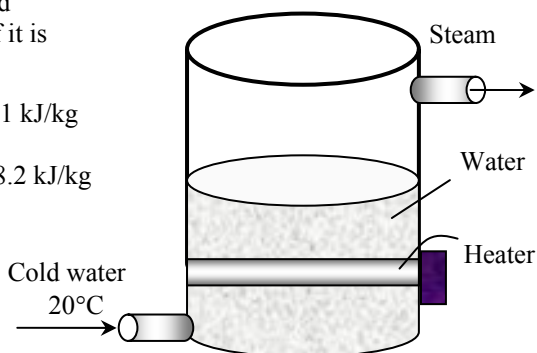
$$\begin{aligned} q_{\text{preheating}} &= q_{\text{boiling}} \\ (h_{f@T_{\text{sat}}} - h_{f@20^\circ\text{C}}) &= h_{fg@T_{\text{sat}}} \\ h_{f@T_{\text{sat}}} - 83.91 \text{ kJ/kg} &= h_{fg@T_{\text{sat}}} \rightarrow h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 83.91 \text{ kJ/kg} \end{aligned}$$

The solution of this problem requires choosing a boiling temperature, reading h_f and h_{fg} at that temperature, and substituting the values into the relation above to see if it is satisfied. By trial and error, (Table A-4)

$$\text{At } 310^\circ\text{C: } h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1402.0 - 1325.9 = 76.1 \text{ kJ/kg}$$

$$\text{At } 315^\circ\text{C: } h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1431.6 - 1283.4 = 148.2 \text{ kJ/kg}$$

The temperature that satisfies this condition is determined from the two values above by interpolation to be 310.6°C. The saturation pressure corresponding to this temperature is **9.94 MPa**.



6-160E A refrigeration system is to cool eggs by chilled air at a rate of 10,000 eggs per hour. The rate of heat removal from the eggs, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The eggs are at uniform temperatures before and after cooling. 3 The cooling section is well-insulated. 4 The properties of eggs are constant. 5 The local atmospheric pressure is 1 atm.

Properties The properties of the eggs are given to $\rho = 67.4 \text{ lbm/ft}^3$ and $c_p = 0.80 \text{ Btu/lbm}\cdot^\circ\text{F}$. The specific heat of air at room temperature $c_p = 0.24 \text{ Btu/lbm}\cdot^\circ\text{F}$ (Table A-2E). The gas constant of air is $R = 0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$ (Table A-1E).

Analysis (a) Noting that eggs are cooled at a rate of 10,000 eggs per hour, eggs can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{egg}} = (10,000 \text{ eggs/h})(0.14 \text{ lbm/egg}) = 1400 \text{ lbm/h} = 0.3889 \text{ lbm/s}$$

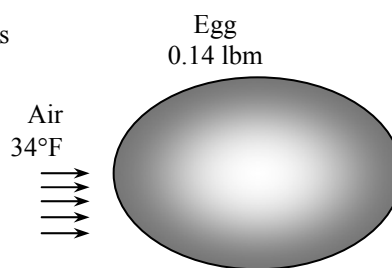
Taking the egg flow stream in the cooler as the system, the energy balance for steadily flowing eggs can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{egg}} = \dot{m}_{\text{egg}}c_p(T_1 - T_2)$$



Then the rate of heat removal from the eggs as they are cooled from 90°F to 50°F at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m}c_p\Delta T)_{\text{egg}} = (1400 \text{ lbm/h})(0.80 \text{ Btu/lbm}\cdot^\circ\text{F})(90 - 50)^\circ\text{F} = \mathbf{44,800 \text{ Btu/h}}$$

(b) All the heat released by the eggs is absorbed by the refrigerated air since heat transfer through the walls of cooler is negligible, and the temperature rise of air is not to exceed 10°F. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p\Delta T)_{\text{air}}} = \frac{44,800 \text{ Btu/h}}{(0.24 \text{ Btu/lbm}\cdot^\circ\text{F})(10^\circ\text{F})} = 18,667 \text{ lbm/h}$$

$$\rho_{\text{air}} = \frac{P}{RT} = \frac{14.7 \text{ psia}}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(34 + 460)\text{R}} = 0.0803 \text{ lbm/ft}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{18,667 \text{ lbm/h}}{0.0803 \text{ lbm/ft}^3} = \mathbf{232,500 \text{ ft}^3/\text{h}}$$

6-161 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The entire water body is maintained at a uniform temperature of 55°C. 3 Heat losses from the outer surfaces of the bath are negligible. 4 Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is $c_p = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. Also, the specific heat of glass is $0.80 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis (a) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

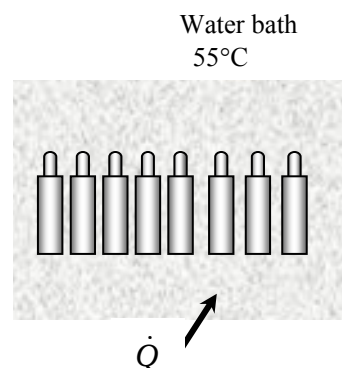
Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$



Then the rate of heat removal by the bottles as they are heated from 20 to 55°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg} \cdot ^\circ\text{C})(55 - 20)^\circ\text{C} = 56,000 \text{ W}$$

The amount of water removed by the bottles is

$$\begin{aligned} \dot{m}_{\text{water, out}} &= (\text{Flow rate of bottles})(\text{Water removed per bottle}) \\ &= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = 2.67 \times 10^{-3} \text{ kg/s} \end{aligned}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C})(55 - 15)^\circ\text{C} = 446 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 56,000 + 446 = 56,446 \text{ W}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

6-162 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The entire water body is maintained at a uniform temperature of 50°C. 3 Heat losses from the outer surfaces of the bath are negligible. 4 Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is $c_p = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. Also, the specific heat of glass is $0.80 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis (a) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

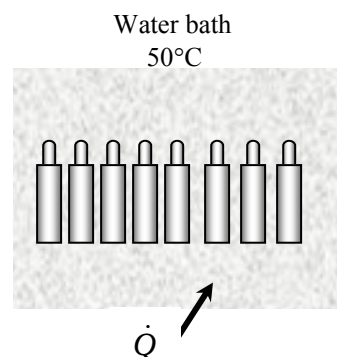
Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}}c_p(T_2 - T_1)$$



Then the rate of heat removal by the bottles as they are heated from 20 to 50°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}}c_p\Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg} \cdot ^\circ\text{C})(50 - 20)^\circ\text{C} = 48,000 \text{ W}$$

The amount of water removed by the bottles is

$$\begin{aligned} \dot{m}_{\text{water, out}} &= (\text{Flow rate of bottles})(\text{Water removed per bottle}) \\ &= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = 2.67 \times 10^{-3} \text{ kg/s} \end{aligned}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}}c_p\Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C})(50 - 15)^\circ\text{C} = 391 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 48,000 + 391 = \mathbf{48,391 \text{ W}}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

6-163 Long aluminum wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The thermal properties of the wire are constant.

Properties The properties of aluminum are given to be $\rho = 2702 \text{ kg/m}^3$ and $c_p = 0.896 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{V} = \rho(\pi r_0^2) \mathcal{V} = (2702 \text{ kg/m}^3) \pi (0.0015 \text{ m})^2 (10 \text{ m/min}) = 0.191 \text{ kg/min}$$

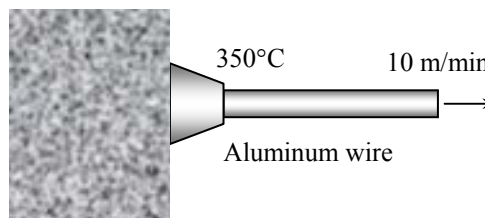
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} h_1 = \dot{Q}_{\text{out}} + \dot{m} h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{wire}} = \dot{m}_{\text{wire}} c_p (T_1 - T_2)$$



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m} c_p [T(t) - T_\infty] = (0.191 \text{ kg/min})(0.896 \text{ kJ/kg} \cdot ^\circ\text{C})(350 - 50)^\circ\text{C} = 51.3 \text{ kJ/min} = \mathbf{0.856 \text{ kW}}$$

6-164 Long copper wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The thermal properties of the wire are constant.

Properties The properties of copper are given to be $\rho = 8950 \text{ kg/m}^3$ and $c_p = 0.383 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{V} = \rho(\pi r_0^2) \mathcal{V} = (8950 \text{ kg/m}^3) \pi (0.0015 \text{ m})^2 (10 \text{ m/min}) = 0.633 \text{ kg/min}$$

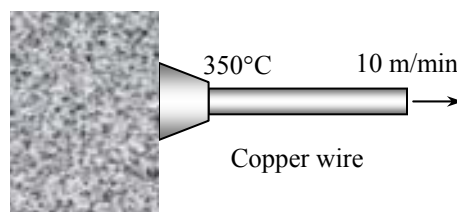
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} h_1 = \dot{Q}_{\text{out}} + \dot{m} h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{wire}} = \dot{m}_{\text{wire}} c_p (T_1 - T_2)$$



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m} c_p [T(t) - T_\infty] = (0.633 \text{ kg/min})(0.383 \text{ kJ/kg} \cdot ^\circ\text{C})(350 - 50)^\circ\text{C} = 72.7 \text{ kJ/min} = \mathbf{1.21 \text{ kW}}$$

6-165 Steam at a saturation temperature of $T_{\text{sat}} = 40^\circ\text{C}$ condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C . The rate of condensation of steam is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Water is an incompressible substance with constant properties at room temperature. 3 The changes in kinetic and potential energies are negligible.

Properties The properties of water at room temperature are $\rho = 997 \text{ kg/m}^3$ and $c_p = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3). The enthalpy of vaporization of water at 40°C is $h_{fg} = 2406.0 \text{ kJ/kg}$ (Table A-4).

Analysis The mass flow rate of water through the tube is

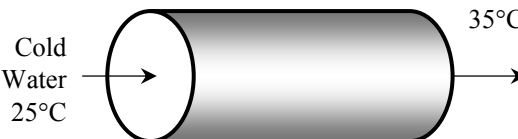
$$\dot{m}_{\text{water}} = \rho V A_c = (997 \text{ kg/m}^3)(2 \text{ m/s})[\pi(0.03 \text{ m})^2 / 4] = 1.409 \text{ kg/s}$$

Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{steady}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{water}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$


Then the rate of heat transfer to the water and the rate of condensation become

$$\dot{Q} = \dot{m} c_p (T_{\text{out}} - T_{\text{in}}) = (1.409 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(35 - 25)^\circ\text{C} = 58.9 \text{ kW}$$

$$\dot{Q} = \dot{m}_{\text{cond}} h_{fg} \rightarrow \dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9 \text{ kJ/s}}{2406.0 \text{ kJ/kg}} = \mathbf{0.0245 \text{ kg/s}}$$

6-166 A fan is powered by a 0.5 hp motor, and delivers air at a rate of 85 m³/min. The highest possible air velocity at the fan exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity and the change in potential energy are negligible, $V_1 \cong 0$ and $\Delta pe \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** The efficiencies of the motor and the fan are 100% since best possible operation is assumed. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be $\rho = 1.18 \text{ kg/m}^3$. The constant pressure specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2).

Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The velocity of air leaving the fan will be highest when all of the entire electrical energy drawn by the motor is converted to kinetic energy, and the friction between the air layers is zero. In this best possible case, no energy will be converted to thermal energy, and thus the temperature change of air will be zero, $T_2 = T_1$. Then the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } V_1 \cong 0 \text{ and } \Delta pe \cong 0)$$

Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}V_2^2/2$$

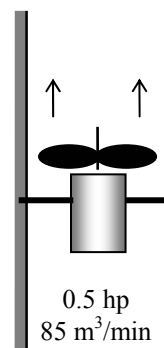
where

$$\dot{m} = \rho \dot{V} = (1.18 \text{ kg/m}^3)(85 \text{ m}^3/\text{min}) = 100.3 \text{ kg/min} = 1.67 \text{ kg/s}$$

Solving for V_2 and substituting gives

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5 \text{ hp}) \left(\frac{745.7 \text{ W}}{1 \text{ hp}} \right) \left(\frac{1 \text{ m}^2/\text{s}^2}{1 \text{ W}} \right)}{1.67 \text{ kg/s}}} = \mathbf{21.1 \text{ m/s}}$$

Discussion In reality, the velocity will be less because of the inefficiencies of the motor and the fan.



6-167 A tank initially contains saturated mixture of R-134a. A valve is opened and R-134a vapor only is allowed to escape slowly such that temperature remains constant. The heat transfer necessary with the surroundings to maintain the temperature and pressure of the R-134a constant is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the exit remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

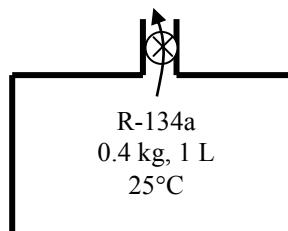
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$\begin{aligned} m_{\text{in}} - m_{\text{out}} &= \Delta m_{\text{system}} \\ -m_e &= m_2 - m_1 \\ m_e &= m_1 - m_2 \end{aligned}$$

Energy balance:

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ Q_{\text{in}} - m_e h_e &= m_2 u_2 - m_1 u_1 \\ Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + m_e h_e \end{aligned}$$



Combining the two balances:

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The specific volume at the initial state is

$$v_1 = \frac{V}{m_1} = \frac{0.001 \text{ m}^3}{0.4 \text{ kg}} = 0.0025 \text{ m}^3/\text{kg}$$

The initial state properties of R-134a in the tank are

$$\left. \begin{aligned} T_1 &= 26^\circ\text{C} \\ v_1 &= 0.0025 \text{ m}^3/\text{kg} \end{aligned} \right\} \begin{aligned} x_1 &= \frac{v_1 - v_f}{v_{fg}} = \frac{0.0025 - 0.0008313}{0.029976 - 0.0008313} = 0.05726 \\ u_1 &= u_f + x_1 u_{fg} = 87.26 + (0.05726)(156.87) = 96.24 \text{ kJ/kg} \end{aligned} \quad (\text{Table A-11})$$

The enthalpy of saturated vapor refrigerant leaving the bottle is

$$h_e = h_g @ 26^\circ\text{C} = 264.68 \text{ kJ/kg}$$

The specific volume at the final state is

$$v_2 = \frac{V}{m_2} = \frac{0.001 \text{ m}^3}{0.1 \text{ kg}} = 0.01 \text{ m}^3/\text{kg}$$

The internal energy at the final state is

$$\left. \begin{aligned} T_2 &= 26^\circ\text{C} \\ v_2 &= 0.01 \text{ m}^3/\text{kg} \end{aligned} \right\} \begin{aligned} x_2 &= \frac{v_2 - v_f}{v_{fg}} = \frac{0.01 - 0.0008313}{0.029976 - 0.0008313} = 0.3146 \\ u_2 &= u_f + x_2 u_{fg} = 87.26 + (0.3146)(156.87) = 136.61 \text{ kJ/kg} \end{aligned} \quad (\text{Table A-11})$$

Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e \\ &= (0.1 \text{ kg})(136.61 \text{ kJ/kg}) - (0.4 \text{ kg})(96.24 \text{ kJ/kg}) + (0.4 - 0.1 \text{ kg})(264.68 \text{ kJ/kg}) \\ &= \mathbf{54.6 \text{ kJ}} \end{aligned}$$

6-168 CD EES Steam expands in a turbine steadily. The mass flow rate of the steam, the exit velocity, and the power output are to be determined.

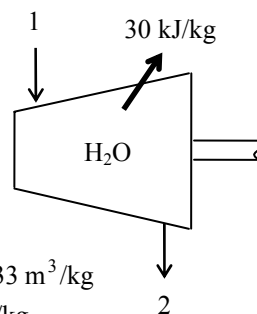
Assumptions 1 This is a steady-flow process since there is no change with time. 2 Potential energy changes are negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left. \begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 550^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_1 = 0.035655 \text{ m}^3/\text{kg} \\ h_1 = 3502.0 \text{ kJ/kg} \end{array}$$

and

$$\left. \begin{array}{l} P_2 = 25 \text{ kPa} \\ x_2 = 0.95 \end{array} \right\} \begin{array}{l} \nu_2 = \nu_f + x_2 \nu_{fg} = 0.00102 + (0.95)(6.2034 - 0.00102) = 5.8933 \text{ m}^3/\text{kg} \\ h_2 = h_f + x_2 h_{fg} = 271.96 + (0.95)(2345.5) = 2500.2 \text{ kJ/kg} \end{array}$$



Analysis (a) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\nu_1} V_1 A_1 = \frac{1}{0.035655 \text{ m}^3/\text{kg}} (60 \text{ m/s})(0.015 \text{ m}^2) = \mathbf{25.24 \text{ kg/s}}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity is determined from

$$\dot{m} = \frac{1}{\nu_2} V_2 A_2 \longrightarrow V_2 = \frac{\dot{m} \nu_2}{A_2} = \frac{(25.24 \text{ kg/s})(5.8933 \text{ m}^3/\text{kg})}{0.14 \text{ m}^2} = \mathbf{1063 \text{ m/s}}$$

(c) We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \Delta p_e \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{Q}_{\text{out}} - \dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Then the power output of the turbine is determined by substituting to be

$$\begin{aligned} \dot{W}_{\text{out}} &= -(25.24 \times 30) \text{ kJ/s} - (25.24 \text{ kg/s}) \left(2500.2 - 3502.0 + \frac{(1063 \text{ m/s})^2 - (60 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right) \\ &= \mathbf{10,330 \text{ kW}} \end{aligned}$$

6-169 EES Problem 6-168 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm² to 3000 cm² is to be investigated. The exit velocity and the power output are to be plotted against the exit pressure for the exit areas of 1000, 2000, and 3000 cm².

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

Fluid\$='Steam_IAPWS'

```
A[1]=150 [cm^2]
T[1]=550 [C]
P[1]=10000 [kPa]
Vel[1]= 60 [m/s]
A[2]=1400 [cm^2]
P[2]=25 [kPa]
q_out = 30 [kJ/kg]
m_dot = A[1]*Vel[1]/v[1]*convert(cm^2,m^2)
v[1]=volume(Fluid$, T=T[1], P=P[1]) "specific volume of steam at state 1"
Vel[2]=m_dot*v[2]/(A[2]*convert(cm^2,m^2))
v[2]=volume(Fluid$, x=0.95, P=P[2]) "specific volume of steam at state 2"
T[2]=temperature(Fluid$, P=P[2], v=v[2]) "[C]" "not required, but good to know"
```

"[conservation of Energy for steady-flow:]

"Ein_dot - Eout_dot = DeltaE_dot" "For steady-flow, DeltaE_dot = 0"

DELTAE_dot=0 "[kW]"

"For the turbine as the control volume, neglecting the PE of each flow steam:"

E_dot_in=E_dot_out

h[1]=enthalpy(Fluid\$,T=T[1], P=P[1])

E_dot_in=m_dot*(h[1]+ Vel[1]^2/2*Convert(m^2/s^2, kJ/kg))

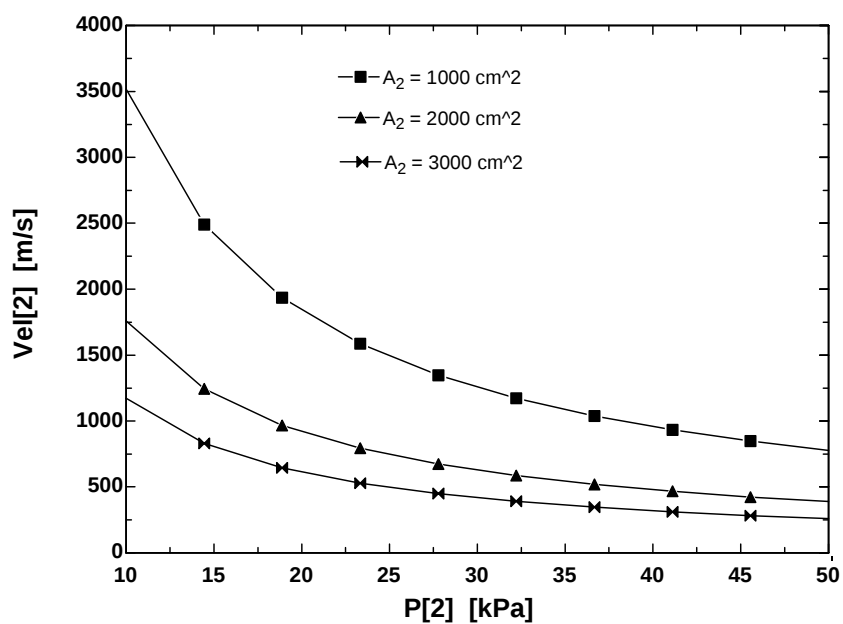
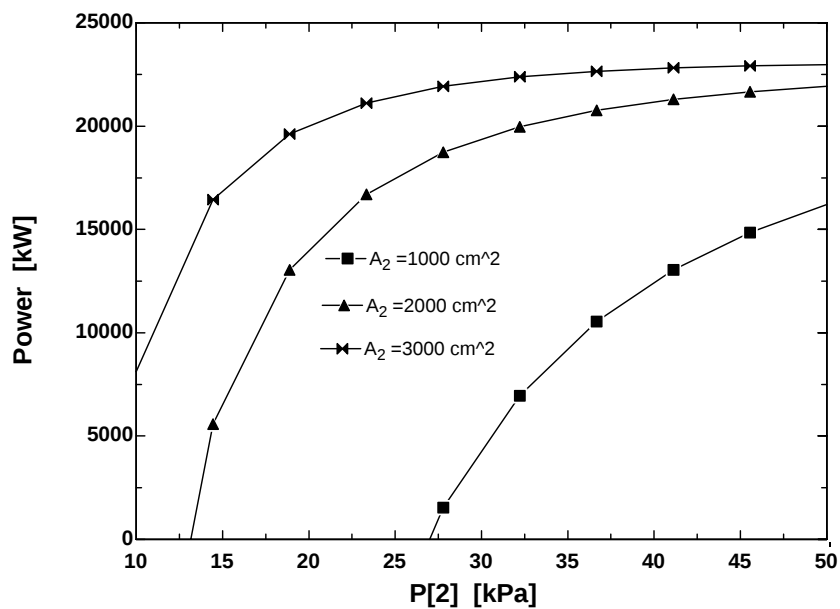
h[2]=enthalpy(Fluid\$,x=0.95, P=P[2])

E_dot_out=m_dot*(h[2]+ Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+ m_dot *q_out+ W_dot_out

Power=W_dot_out

Q_dot_out=m_dot*q_out

Power [kW]	P ₂ [kPa]	Vel ₂ [m/s]
-54208	10	2513
-14781	14.44	1778
750.2	18.89	1382
8428	23.33	1134
12770	27.78	962.6
15452	32.22	837.6
17217	36.67	742.1
18432	41.11	666.7
19299	45.56	605.6
19935	50	555



6-170E Refrigerant-134a is compressed steadily by a compressor. The mass flow rate of the refrigerant and the exit temperature are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Tables A-11E through A-13E)

$$\left. \begin{array}{l} P_1 = 15 \text{ psia} \\ T_1 = 20^\circ\text{F} \end{array} \right\} \begin{array}{l} v_1 = 3.2551 \text{ ft}^3/\text{lbm} \\ h_1 = 107.52 \text{ Btu/lbm} \end{array}$$

Analysis (a) The mass flow rate of refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ ft}^3/\text{s}}{3.2551 \text{ ft}^3/\text{lbm}} = \mathbf{3.072 \text{ lbm/s}}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

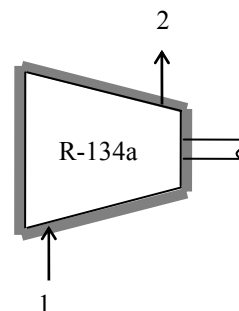
Substituting,

$$(45 \text{ hp}) \left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}} \right) = (3.072 \text{ lbm/s})(h_2 - 107.52) \text{ Btu/lbm}$$

$$h_2 = 117.87 \text{ Btu/lbm}$$

Then the exit temperature becomes

$$\left. \begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 117.87 \text{ Btu/lbm} \end{array} \right\} T_2 = \mathbf{95.7^\circ\text{F}}$$



6-171 Air is preheated by the exhaust gases of a gas turbine in a regenerator. For a specified heat transfer rate, the exit temperature of air and the mass flow rate of exhaust gases are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 There are no work interactions. 4 Heat loss from the regenerator to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 5 Exhaust gases can be treated as air. 6 Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The enthalpies of air are (Table A-21)

$$T_1 = 550 \text{ K} \rightarrow h_1 = 555.74 \text{ kJ/kg}$$

$$T_3 = 800 \text{ K} \rightarrow h_3 = 821.95 \text{ kJ/kg}$$

$$T_4 = 600 \text{ K} \rightarrow h_4 = 607.02 \text{ kJ/kg}$$

Analysis (a) We take the *air side* of the heat exchanger as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \quad (\text{since } \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_2 - h_1)$$

Substituting,

$$3200 \text{ kJ/s} = (800/60 \text{ kg/s})(h_2 - 554.71 \text{ kJ/kg}) \rightarrow h_2 = 794.71 \text{ kJ/kg}$$

Then from Table A-21 we read

$$T_2 = 775.1 \text{ K}$$

(b) Treating the exhaust gases as an ideal gas, the mass flow rate of the exhaust gases is determined from the steady-flow energy relation applied only to the exhaust gases,

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

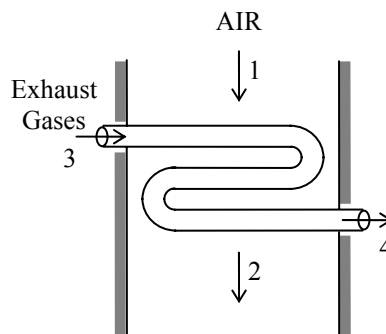
$$\dot{m}_{\text{exhaust}} h_3 = \dot{Q}_{\text{out}} + \dot{m}_{\text{exhaust}} h_4 \quad (\text{since } \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}_{\text{exhaust}} (h_3 - h_4)$$

$$3200 \text{ kJ/s} = \dot{m}_{\text{exhaust}} (821.95 - 607.02) \text{ kJ/kg}$$

It yields

$$\dot{m}_{\text{exhaust}} = 14.9 \text{ kg/s}$$



6-172 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

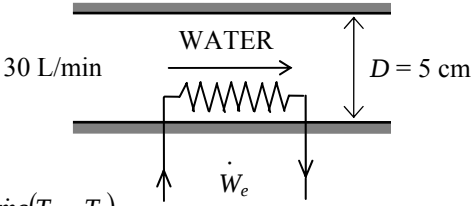
Properties The density and specific heat of water at room temperature are $\rho = 1000 \text{ kg/m}^3$ and $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-3).

Analysis (a) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\phi^0} (\text{steady})}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{e,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{\text{out}} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\phi^0}] = \dot{m}c(T_2 - T_1)$$


The mass flow rate of water through the pipe is

$$\dot{m} = \rho \dot{V}_1 = (1000 \text{ kg/m}^3)(0.030 \text{ m}^3/\text{min}) = 30 \text{ kg/min}$$

Therefore,

$$\dot{W}_{e,\text{in}} = \dot{m}c(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55 - 20)^\circ\text{C} = \mathbf{73.2 \text{ kW}}$$

(b) The average velocity of water through the pipe is determined from

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3 \text{ m/min}}$$

6-173 CD EES An insulated cylinder equipped with an external spring initially contains air. The tank is connected to a supply line, and air is allowed to enter the cylinder until its volume doubles. The mass of the air that entered and the final temperature in the cylinder are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **4** The spring is a linear spring. **5** The device is insulated and thus heat transfer is negligible. **6** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-1). The specific heats of air at room temperature are $c_v = 0.718$ and $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a). Also, $u = c_v T$ and $h = c_p T$.

Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{b,\text{out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

$$\text{Combining the two relations,} \quad (m_2 - m_1) h_i = W_{b,\text{out}} + m_2 u_2 - m_1 u_1$$

$$\text{or,} \quad (m_2 - m_1) c_p T_i = W_{b,\text{out}} + m_2 c_v T_2 - m_1 c_v T_1$$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(200 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 0.472 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(600 \text{ kPa})(0.4 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{836.2}{T_2}$$

$$\text{Then from the mass balance becomes} \quad m_i = m_2 - m_1 = \frac{836.2}{T_2} - 0.472$$

The spring is a linear spring, and thus the boundary work for this process can be determined from

$$W_b = \text{Area} = \frac{P_1 + P_2}{2} (V_2 - V_1) = \frac{(200 + 600) \text{ kPa}}{2} (0.4 - 0.2) \text{ m}^3 = 80 \text{ kJ}$$

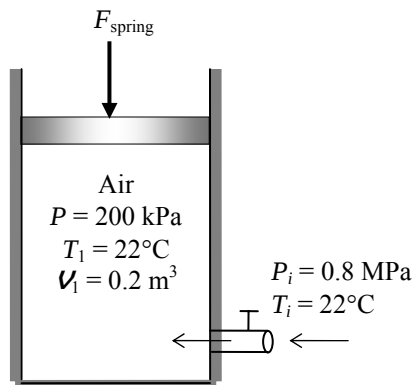
Substituting into the energy balance, the final temperature of air T_2 is determined to be

$$-80 = -\left(\frac{836.2}{T_2} - 0.472\right)(1.005)(295) + \left(\frac{836.2}{T_2}\right)(0.718)(T_2) - (0.472)(0.718)(295)$$

$$\text{It yields} \quad T_2 = \mathbf{344.1 \text{ K}}$$

$$\text{Thus,} \quad m_2 = \frac{836.2}{T_2} = \frac{836.2}{344.1} = 2.430 \text{ kg}$$

$$\text{and} \quad m_i = m_2 - m_1 = 2.430 - 0.472 = \mathbf{1.96 \text{ kg}}$$



6-174E ... 6-178 Design and Essay Problems

Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 7
**THE SECOND LAW OF
THERMODYNAMICS**

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Second Law of Thermodynamics and Thermal Energy Reservoirs

7-1C Water is not a fuel; thus the claim is false.

7-2C Transferring 5 kWh of heat to an electric resistance wire and producing 5 kWh of electricity.

7-3C An electric resistance heater which consumes 5 kWh of electricity and supplies 6 kWh of heat to a room.

7-4C Transferring 5 kWh of heat to an electric resistance wire and producing 6 kWh of electricity.

7-5C No. Heat cannot flow from a low-temperature medium to a higher temperature medium.

7-6C A thermal-energy reservoir is a body that can supply or absorb finite quantities of heat isothermally. Some examples are the oceans, the lakes, and the atmosphere.

7-7C Yes. Because the temperature of the oven remains constant no matter how much heat is transferred to the potatoes.

7-8C The surrounding air in the room that houses the TV set.

Heat Engines and Thermal Efficiency

7-9C No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

7-10C Heat engines are cyclic devices that receive heat from a source, convert some of it to work, and reject the rest to a sink.

7-11C Method (b). With the heating element in the water, heat losses to the surrounding air are minimized, and thus the desired heating can be achieved with less electrical energy input.

7-12C No. Because 100% of the work can be converted to heat.

7-13C It is expressed as "No heat engine can exchange heat with a single reservoir, and produce an equivalent amount of work".

7-14C (a) No, (b) Yes. According to the second law, no heat engine can have an efficiency of 100%.

7-15C No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

7-16C No. The Kelvin-Planck limitation applies only to heat engines; engines that receive heat and convert some of it to work.

7-17 The power output and thermal efficiency of a power plant are given. The rate of heat rejection is to be determined, and the result is to be compared to the actual case in practice.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.

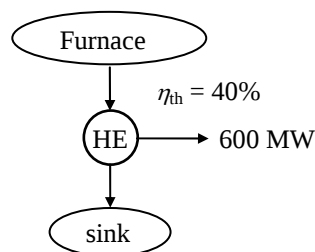
Analysis The rate of heat supply to the power plant is determined from the thermal efficiency relation,

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{600 \text{ MW}}{0.4} = 1500 \text{ MW}$$

The rate of heat transfer to the river water is determined from the first law relation for a heat engine,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,out}} = 1500 - 600 = \mathbf{900 \text{ MW}}$$

In reality the amount of heat rejected to the river will be **lower** since part of the heat will be lost to the surrounding air from the working fluid as it passes through the pipes and other components.



7-18 The heat input and thermal efficiency of a heat engine are given. The work output of the heat engine is to be determined.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.

Analysis Applying the definition of the thermal efficiency to the heat engine,

$$W_{\text{net}} = \eta_{\text{th}} Q_H = (0.35)(1.3 \text{ kJ}) = \mathbf{0.455 \text{ kJ}}$$

7-19E The work output and heat rejection of a heat engine that propels a ship are given. The thermal efficiency of the engine is to be determined.

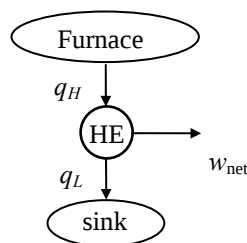
Assumptions 1 The plant operates steadily. 2 Heat losses from the working fluid at the pipes and other components are negligible.

Analysis Applying the first law to the heat engine gives

$$q_H = w_{\text{net}} + q_L = 500 \text{ Btu/lbm} + 300 \text{ Btu/lbm} = 800 \text{ Btu/lbm}$$

Substituting this result into the definition of the thermal efficiency,

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_H} = \frac{500 \text{ Btu/lbm}}{800 \text{ Btu/lbm}} = \mathbf{0.625}$$

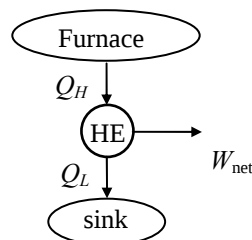


7-20 The work output and heat input of a heat engine are given. The heat rejection is to be determined.

Assumptions 1 The plant operates steadily. 2 Heat losses from the working fluid at the pipes and other components are negligible.

Analysis Applying the first law to the heat engine gives

$$Q_L = Q_H - W_{\text{net}} = 500 \text{ kJ} - 200 \text{ kJ} = \mathbf{300 \text{ kJ}}$$



7-21 The heat rejection and thermal efficiency of a heat engine are given. The heat input to the engine is to be determined.

Assumptions 1 The plant operates steadily. 2 Heat losses from the working fluid at the pipes and other components are negligible.

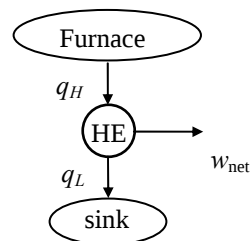
Analysis According to the definition of the thermal efficiency as applied to the heat engine,

$$w_{\text{net}} = \eta_{\text{th}} q_H$$

$$q_H - q_L = \eta_{\text{th}} q_H$$

which when rearranged gives

$$q_H = \frac{q_L}{1 - \eta_{\text{th}}} = \frac{1000 \text{ kJ/kg}}{1 - 0.4} = \mathbf{1667 \text{ kJ/kg}}$$



7-22 The power output and fuel consumption rate of a power plant are given. The thermal efficiency is to be determined.

Assumptions The plant operates steadily.

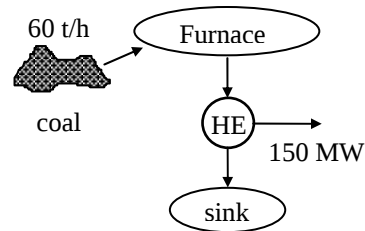
Properties The heating value of coal is given to be 30,000 kJ/kg.

Analysis The rate of heat supply to this power plant is

$$\begin{aligned}\dot{Q}_H &= \dot{m}_{\text{coal}} q_{\text{HV,coal}} \\ &= (60,000 \text{ kg/h})(30,000 \text{ kJ/kg}) = 1.8 \times 10^9 \text{ kJ/h} \\ &= 500 \text{ MW}\end{aligned}$$

Then the thermal efficiency of the plant becomes

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{150 \text{ MW}}{500 \text{ MW}} = 0.300 = \mathbf{30.0\%}$$



7-23 The power output and fuel consumption rate of a car engine are given. The thermal efficiency of the engine is to be determined.

Assumptions The car operates steadily.

Properties The heating value of the fuel is given to be 44,000 kJ/kg.

Analysis The mass consumption rate of the fuel is

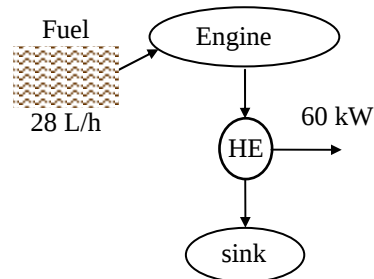
$$\dot{m}_{\text{fuel}} = (\rho \dot{V})_{\text{fuel}} = (0.8 \text{ kg/L})(28 \text{ L/h}) = 22.4 \text{ kg/h}$$

The rate of heat supply to the car is

$$\begin{aligned}\dot{Q}_H &= \dot{m}_{\text{fuel}} q_{\text{HV,fuel}} \\ &= (22.4 \text{ kg/h})(44,000 \text{ kJ/kg}) \\ &= 985,600 \text{ kJ/h} = 273.78 \text{ kW}\end{aligned}$$

Then the thermal efficiency of the car becomes

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{60 \text{ kW}}{273.78 \text{ kW}} = 0.219 = \mathbf{21.9\%}$$

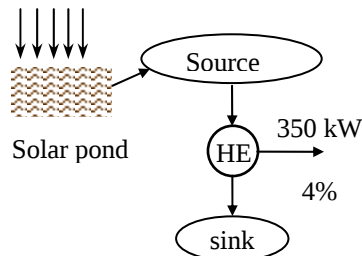


7-24E The power output and thermal efficiency of a solar pond power plant are given. The rate of solar energy collection is to be determined.

Assumptions The plant operates steadily.

Analysis The rate of solar energy collection or the rate of heat supply to the power plant is determined from the thermal efficiency relation to be

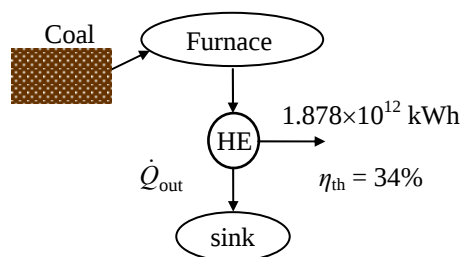
$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{350 \text{ kW}}{0.04} \left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}} \right) \left(\frac{3600 \text{ s}}{1 \text{ h}} \right) = \mathbf{2.986 \times 10^7 \text{ Btu/h}}$$



7-25 The United States produces about 51 percent of its electricity from coal at a conversion efficiency of about 34 percent. The amount of heat rejected by the coal-fired power plants per year is to be determined.

Analysis Noting that the conversion efficiency is 34%, the amount of heat rejected by the coal plants per year is

$$\begin{aligned} \eta_{\text{th}} &= \frac{W_{\text{coal}}}{Q_{\text{in}}} = \frac{W_{\text{coal}}}{Q_{\text{out}} + W_{\text{coal}}} \\ Q_{\text{out}} &= \frac{W_{\text{coal}}}{\eta_{\text{th}}} - W_{\text{coal}} \\ &= \frac{1.878 \times 10^{12} \text{ kWh}}{0.34} - 1.878 \times 10^{12} \text{ kWh} \\ &= \mathbf{3.646 \times 10^{12} \text{ kWh}} \end{aligned}$$

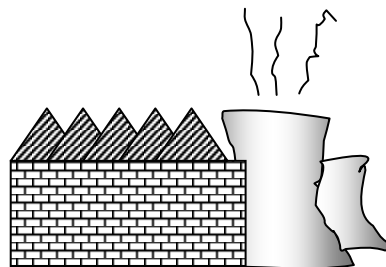


7-26 The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 5 years is to be determined.

Assumptions 1 Power is generated continuously by either plant at full capacity. 2 The time value of money (interest, inflation, etc.) is not considered.

Properties The heating value of the coal is given to be 28×10^6 kJ/ton.

Analysis For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are



$$\text{Construction cost}_{\text{coal}} = (150,000,000 \text{ kW})(\$1300/\text{kW}) = \$195 \times 10^9$$

$$\text{Construction cost}_{\text{IGCC}} = (150,000,000 \text{ kW})(\$1500/\text{kW}) = \$225 \times 10^9$$

$$\text{Construction cost difference} = \$225 \times 10^9 - \$195 \times 10^9 = \$30 \times 10^9$$

The amount of electricity produced by either plant in 5 years is

$$W_e = \dot{W} \Delta t = (150,000,000 \text{ kW})(5 \times 365 \times 24 \text{ h}) = 6.570 \times 10^{12} \text{ kWh}$$

The amount of fuel needed to generate a specified amount of power can be determined from

$$\eta = \frac{W_e}{Q_{\text{in}}} \rightarrow Q_{\text{in}} = \frac{W_e}{\eta} \quad \text{or} \quad m_{\text{fuel}} = \frac{Q_{\text{in}}}{\text{Heating value}} = \frac{W_e}{\eta(\text{Heating value})}$$

Then the amount of coal needed to generate this much electricity by each plant and their difference are

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570 \times 10^{12} \text{ kWh}}{(0.34)(28 \times 10^6 \text{ kJ/ton})} \left(\frac{3600 \text{ kJ}}{1 \text{ kWh}} \right) = 2.484 \times 10^9 \text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570 \times 10^{12} \text{ kWh}}{(0.45)(28 \times 10^6 \text{ kJ/ton})} \left(\frac{3600 \text{ kJ}}{1 \text{ kWh}} \right) = 1.877 \times 10^9 \text{ tons}$$

$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 2.484 \times 10^9 - 1.877 \times 10^9 = 0.607 \times 10^9 \text{ tons}$$

For Δm_{coal} to pay for the construction cost difference of \$30 billion, the price of coal should be

$$\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta m_{\text{coal}}} = \frac{\$30 \times 10^9}{0.607 \times 10^9 \text{ tons}} = \mathbf{\$49.4/\text{ton}}$$

Therefore, the IGCC plant becomes attractive when the price of coal is above \$49.4 per ton.

7-27 EES Problem 7-26 is reconsidered. The price of coal is to be investigated for varying simple payback periods, plant construction costs, and operating efficiency.

Analysis The problem is solved using EES, and the solution is given below.

"Knowns:"

HeatingValue = 28E+6 [kJ/ton]

W_dot = 150E+6 [kW]

{PayBackPeriod = 5 [years]

eta_coal = 0.34

eta_IGCC = 0.45

CostPerkW_Coal = 1300 [\$/kW]

CostPerkW_IGCC=1500 [\$/kW]}

"Analysis:"

"For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are"

ConstructionCost_coal = W_dot *CostPerkW_Coal

ConstructionCost_IGCC= W_dot *CostPerkW_IGCC

ConstructionCost_diff = ConstructionCost_IGCC - ConstructionCost_coal

"The amount of electricity produced by either plant in 5 years is "

W_ele = W_dot*PayBackPeriod*convert(year,h)

"The amount of fuel needed to generate a specified amount of power can be determined from the plant efficiency and the heating value of coal."

"Then the amount of coal needed to generate this much electricity by each plant and their difference are"

"Coal Plant:"

eta_coal = W_ele/Q_in_coal

Q_in_coal =

m_fuel_CoalPlant*HeatingValue*convert(kJ,kWh)

"IGCC Plant:"

eta_IGCC = W_ele/Q_in_IGCC

Q_in_IGCC =

m_fuel_IGCCPlant*HeatingValue*convert(kJ,kWh)

DELTAm_coal = m_fuel_CoalPlant - m_fuel_IGCCPlant

"For to pay for the construction cost difference of \$30 billion, the price of coal should be"

UnitCost_coal = ConstructionCost_diff /DELTAm_coal

"Therefore, the IGCC plant becomes attractive when the price of coal is above \$49.4 per ton. "

SOLUTION

ConstructionCost_coal=1.950E+11 [dollars] ConstructionCost_diff=3.000E+10 [dollars]

ConstructionCost_IGCC=2.250E+11 [dollars] CostPerkW_Coal=1300 [dollars/kW]

CostPerkW_IGCC=1500 [dollars/kW] DELTAm_coal=6.073E+08 [tons]

eta_coal=0.34

eta_IGCC=0.45

HeatingValue=2.800E+07 [kJ/ton]

m_fuel_CoalPlant=2.484E+09 [tons]

m_fuel_IGCCPlant=1.877E+09 [tons]

PayBackPeriod=5 [years]

Q_in_coal=1.932E+13 [kWh]

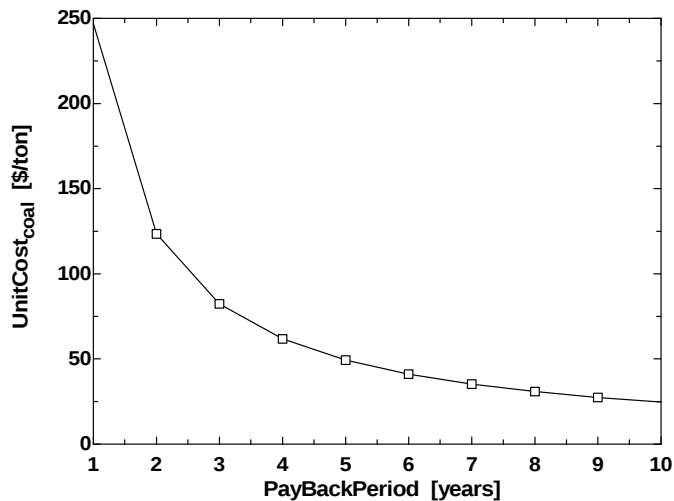
Q_in_IGCC=1.460E+13 [kWh]

UnitCost_coal=49.4 [dollars/ton]

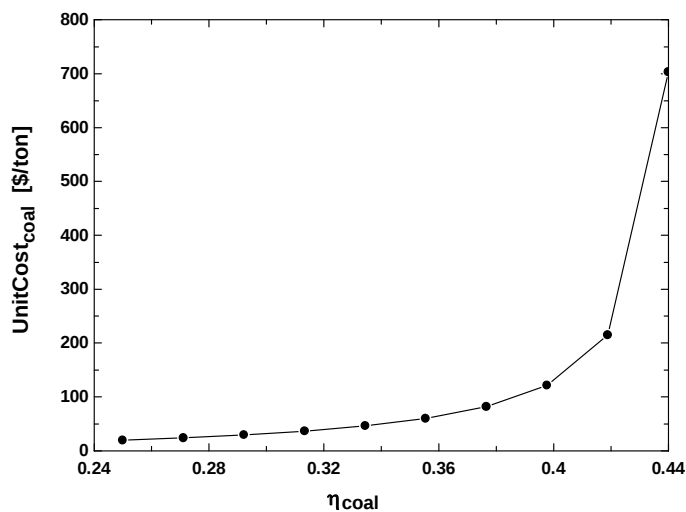
W_dot=1.500E+08 [kW]

W_ele=6.570E+12 [kWh]

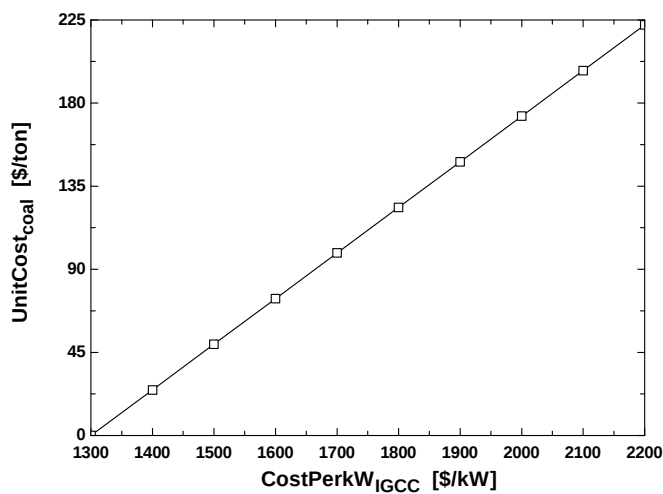
PaybackPeriod [years]	UnitCost _{coal} [\$/ton]
1	247
2	123.5
3	82.33
4	61.75
5	49.4
6	41.17
7	35.28
8	30.87
9	27.44
10	24.7



η_{coal}	UnitCost _{coal} [\$/ton]
0.25	19.98
0.2711	24.22
0.2922	29.6
0.3133	36.64
0.3344	46.25
0.3556	60.17
0.3767	82.09
0.3978	121.7
0.4189	215.2
0.44	703.2



CostPerkW _{IGCC} [\$/kW]	UnitCost _{coal} [\$/ton]
1300	0
1400	24.7
1500	49.4
1600	74.1
1700	98.8
1800	123.5
1900	148.2
2000	172.9
2100	197.6
2200	222.3

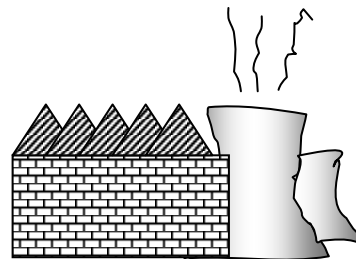


7-28 The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 3 years is to be determined.

Assumptions 1 Power is generated continuously by either plant at full capacity. 2 The time value of money (interest, inflation, etc.) is not considered.

Properties The heating value of the coal is given to be 28×10^6 kJ/ton.

Analysis For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are



$$\text{Construction cost}_{\text{coal}} = (150,000,000 \text{ kW})(\$1300/\text{kW}) = \$195 \times 10^9$$

$$\text{Construction cost}_{\text{IGCC}} = (150,000,000 \text{ kW})(\$1500/\text{kW}) = \$225 \times 10^9$$

$$\text{Construction cost difference} = \$225 \times 10^9 - \$195 \times 10^9 = \$30 \times 10^9$$

The amount of electricity produced by either plant in 3 years is

$$W_e = \dot{W} \Delta t = (150,000,000 \text{ kW})(3 \times 365 \times 24 \text{ h}) = 3.942 \times 10^{12} \text{ kWh}$$

The amount of fuel needed to generate a specified amount of power can be determined from

$$\eta = \frac{W_e}{Q_{\text{in}}} \rightarrow Q_{\text{in}} = \frac{W_e}{\eta} \quad \text{or} \quad m_{\text{fuel}} = \frac{Q_{\text{in}}}{\text{Heating value}} = \frac{W_e}{\eta(\text{Heating value})}$$

Then the amount of coal needed to generate this much electricity by each plant and their difference are

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942 \times 10^{12} \text{ kWh}}{(0.34)(28 \times 10^6 \text{ kJ/ton}) \left(\frac{3600 \text{ kJ}}{1 \text{ kWh}} \right)} = 1.491 \times 10^9 \text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942 \times 10^{12} \text{ kWh}}{(0.45)(28 \times 10^6 \text{ kJ/ton}) \left(\frac{3600 \text{ kJ}}{1 \text{ kWh}} \right)} = 1.126 \times 10^9 \text{ tons}$$

$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 1.491 \times 10^9 - 1.126 \times 10^9 = 0.365 \times 10^9 \text{ tons}$$

For Δm_{coal} to pay for the construction cost difference of \$30 billion, the price of coal should be

$$\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta m_{\text{coal}}} = \frac{\$30 \times 10^9}{0.365 \times 10^9 \text{ tons}} = \mathbf{\$82.2/\text{ton}}$$

Therefore, the IGCC plant becomes attractive when the price of coal is above \$82.2 per ton.

7-29 A coal-burning power plant produces 300 MW of power. The amount of coal consumed during a one-day period and the rate of air flowing through the furnace are to be determined.

Assumptions 1 The power plant operates steadily. 2 The kinetic and potential energy changes are zero.

Properties The heating value of the coal is given to be 28,000 kJ/kg.

Analysis (a) The rate and the amount of heat inputs to the power plant are

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{300 \text{ MW}}{0.32} = 937.5 \text{ MW}$$

$$Q_{\text{in}} = \dot{Q}_{\text{in}} \Delta t = (937.5 \text{ MJ/s})(24 \times 3600 \text{ s}) = 8.1 \times 10^7 \text{ MJ}$$

The amount and rate of coal consumed during this period are

$$m_{\text{coal}} = \frac{Q_{\text{in}}}{q_{\text{HV}}} = \frac{8.1 \times 10^7 \text{ MJ}}{28 \text{ MJ/kg}} = \mathbf{2.893 \times 10^6 \text{ kg}}$$

$$\dot{m}_{\text{coal}} = \frac{m_{\text{coal}}}{\Delta t} = \frac{2.893 \times 10^6 \text{ kg}}{24 \times 3600 \text{ s}} = 33.48 \text{ kg/s}$$

(b) Noting that the air-fuel ratio is 12, the rate of air flowing through the furnace is

$$\dot{m}_{\text{air}} = (\text{AF})\dot{m}_{\text{coal}} = (12 \text{ kg air/kg fuel})(33.48 \text{ kg/s}) = \mathbf{401.8 \text{ kg/s}}$$

Refrigerators and Heat Pumps

7-30C The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a cold medium whereas the purpose of a heat pump is to supply heat to a warm medium.

7-31C The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a refrigerated space whereas the purpose of an air-conditioner is remove heat from a living space.

7-32C No. Because the refrigerator consumes work to accomplish this task.

7-33C No. Because the heat pump consumes work to accomplish this task.

7-34C The coefficient of performance of a refrigerator represents the amount of heat removed from the refrigerated space for each unit of work supplied. It can be greater than unity.

7-35C The coefficient of performance of a heat pump represents the amount of heat supplied to the heated space for each unit of work supplied. It can be greater than unity.

7-36C No. The heat pump captures energy from a cold medium and carries it to a warm medium. It does not create it.

7-37C No. The refrigerator captures energy from a cold medium and carries it to a warm medium. It does not create it.

7-38C No device can transfer heat from a cold medium to a warm medium without requiring a heat or work input from the surroundings.

7-39C The violation of one statement leads to the violation of the other one, as shown in Sec. 7-4, and thus we conclude that the two statements are equivalent.

7-40 The COP and the refrigeration rate of a refrigerator are given. The power consumption and the rate of heat rejection are to be determined.

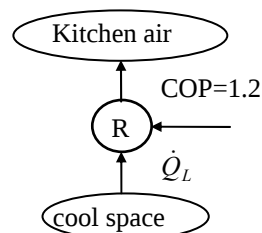
Assumptions The refrigerator operates steadily.

Analysis (a) Using the definition of the coefficient of performance, the power input to the refrigerator is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{60 \text{ kJ/min}}{1.2} = 50 \text{ kJ/min} = \mathbf{0.83 \text{ kW}}$$

(b) The heat transfer rate to the kitchen air is determined from the energy balance,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = 60 + 50 = \mathbf{110 \text{ kJ/min}}$$

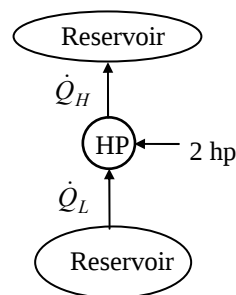


7-41E The heat absorption, the heat rejection, and the power input of a commercial heat pump are given. The COP of the heat pump is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the definition of the heat pump coefficient of performance to this heat pump gives

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{net,in}}} = \frac{15,090 \text{ Btu/h}}{2 \text{ hp}} \left(\frac{1 \text{ hp}}{2544.5 \text{ Btu/h}} \right) = \mathbf{2.97}$$



7-42 The COP and the power input of a residential heat pump are given. The rate of heating effect is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the definition of the heat pump coefficient of performance to this heat pump gives

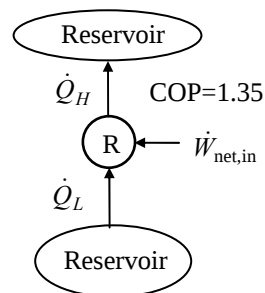
$$\dot{Q}_H = \text{COP}_{\text{HP}} \dot{W}_{\text{net,in}} = (1.6)(2 \text{ kW}) = 3.2 \text{ kW} = \mathbf{3.2 \text{ kJ/s}}$$

7-43 The cooling effect and the COP of a refrigerator are given. The power input to the refrigerator is to be determined.

Assumptions The refrigerator operates steadily.

Analysis Rearranging the definition of the refrigerator coefficient of performance and applying the result to this refrigerator gives

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{10,000 \text{ kJ/h}}{1.35} \left(\frac{1 \text{ h}}{3600 \text{ s}} \right) = \mathbf{2.06 \text{ kW}}$$

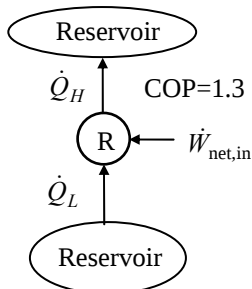


7-44 The cooling effect and the COP of a refrigerator are given. The power input to the refrigerator is to be determined.

Assumptions The refrigerator operates steadily.

Analysis Rearranging the definition of the refrigerator coefficient of performance and applying the result to this refrigerator gives

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{5 \text{ kW}}{1.3} = \mathbf{3.85 \text{ kW}}$$



7-45 The COP and the work input of a heat pump are given. The heat transferred to and from this heat pump are to be determined.

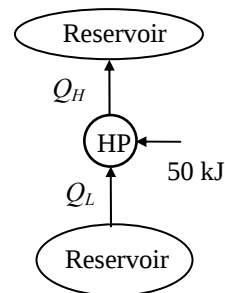
Assumptions The heat pump operates steadily.

Analysis Applying the definition of the heat pump coefficient of performance,

$$Q_H = \text{COP}_{\text{HP}} W_{\text{net,in}} = (1.7)(50 \text{ kJ}) = \mathbf{85 \text{ kJ}}$$

Adapting the first law to this heat pump produces

$$Q_L = Q_H - W_{\text{net,in}} = 85 \text{ kJ} - 50 \text{ kJ} = \mathbf{35 \text{ kJ}}$$



7-46E The COP and the refrigeration rate of an ice machine are given. The power consumption is to be determined.

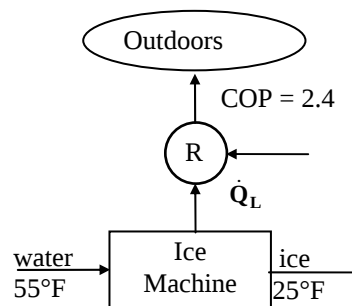
Assumptions The ice machine operates steadily.

Analysis The cooling load of this ice machine is

$$\dot{Q}_L = \dot{m}q_L = (28 \text{ lbm/h})(169 \text{ Btu/lbm}) = 4732 \text{ Btu/h}$$

Using the definition of the coefficient of performance, the power input to the ice machine system is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{4732 \text{ Btu/h}}{2.4} \left(\frac{1 \text{ hp}}{2545 \text{ Btu/h}} \right) = \mathbf{0.775 \text{ hp}}$$



7-47 The COP and the power consumption of a refrigerator are given. The time it will take to cool 5 watermelons is to be determined.

Assumptions 1 The refrigerator operates steadily. 2 The heat gain of the refrigerator through its walls, door, etc. is negligible. 3 The watermelons are the only items in the refrigerator to be cooled.

Properties The specific heat of watermelons is given to be $c = 4.2 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The total amount of heat that needs to be removed from the watermelons is

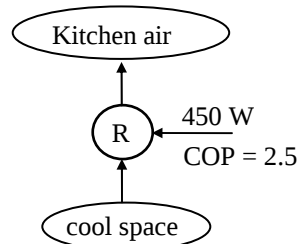
$$Q_L = (mc\Delta T)_{\text{watermelons}} = 5 \times (10 \text{ kg})(4.2 \text{ kJ/kg} \cdot ^\circ\text{C})(20 - 8)^\circ\text{C} = 2520 \text{ kJ}$$

The rate at which this refrigerator removes heat is

$$\dot{Q}_L = (\text{COP}_R)(\dot{W}_{\text{net,in}}) = (2.5)(0.45 \text{ kW}) = 1.125 \text{ kW}$$

That is, this refrigerator can remove 1.125 kJ of heat per second. Thus the time required to remove 2520 kJ of heat is

$$\Delta t = \frac{Q_L}{\dot{Q}_L} = \frac{2520 \text{ kJ}}{1.125 \text{ kJ/s}} = 2240 \text{ s} = \mathbf{37.3 \text{ min}}$$



This answer is optimistic since the refrigerated space will gain some heat during this process from the surrounding air, which will increase the work load. Thus, in reality, it will take longer to cool the watermelons.

7-48 CD EES An air conditioner with a known COP cools a house to desired temperature in 15 min. The power consumption of the air conditioner is to be determined.

Assumptions 1 The air conditioner operates steadily. 2 The house is well-sealed so that no air leaks in or out during cooling. 3 Air is an ideal gas with constant specific heats at room temperature.

Properties The constant volume specific heat of air is given to be $c_v = 0.72 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis Since the house is well-sealed (constant volume), the total amount of heat that needs to be removed from the house is

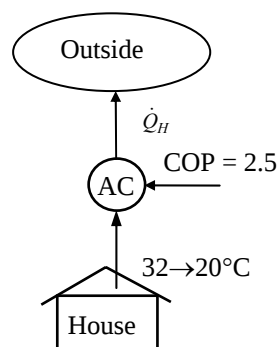
$$Q_L = (mc_v \Delta T)_{\text{House}} = (800 \text{ kg})(0.72 \text{ kJ/kg} \cdot ^\circ\text{C})(32 - 20)^\circ\text{C} = 6912 \text{ kJ}$$

This heat is removed in 15 minutes. Thus the average rate of heat removal from the house is

$$\dot{Q}_L = \frac{Q_L}{\Delta t} = \frac{6912 \text{ kJ}}{15 \times 60 \text{ s}} = 7.68 \text{ kW}$$

Using the definition of the coefficient of performance, the power input to the air-conditioner is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{7.68 \text{ kW}}{2.5} = \mathbf{3.07 \text{ kW}}$$



7-49 EES Problem 7-48 is reconsidered. The rate of power drawn by the air conditioner required to cool the house as a function for air conditioner EER ratings in the range 9 to 16 is to be investigated. Representative costs of air conditioning units in the EER rating range are to be included.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Input Data"

T_1=32 [C]
 T_2=20 [C]
 C_v = 0.72 [kJ/kg-C]
 m_house=800 [kg]
 DELTAtime=20 [min]
 COP=EER/3.412

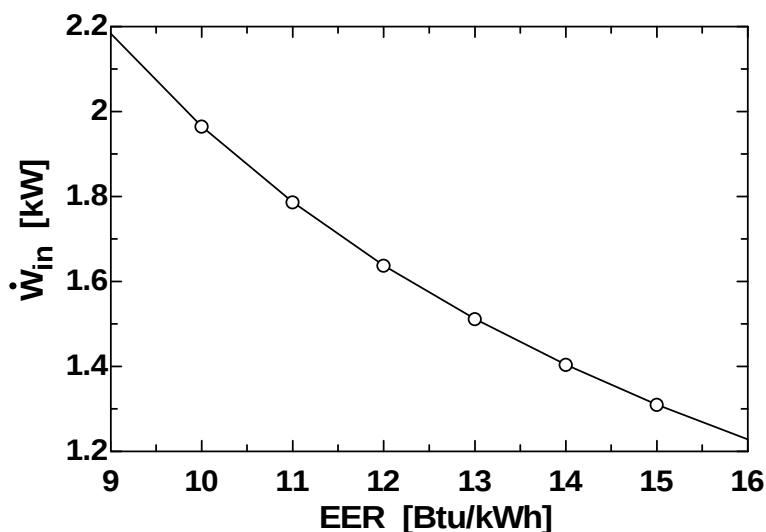
"Assuming no work done on the house and no heat energy added to the house in the time period with no change in KE and PE, the first law applied to the house is:"

E_dot_in - E_dot_out = DELTAE_dot
 E_dot_in = 0
 E_dot_out = Q_dot_L
 DELTAE_dot = m_house*DELTAu_house/DELTAtime
 DELTAu_house = C_v*(T_2-T_1)

"Using the definition of the coefficient of performance of the A/C:"

W_dot_in = Q_dot_L/COP "kJ/min"*convert('kJ/min','kW') "kW"
 Q_dot_H= W_dot_in*convert('KW','kJ/min') + Q_dot_L "kJ/min"

EER [Btu/kWh]	W _{in} [kW]
9	2.184
10	1.965
11	1.787
12	1.638
13	1.512
14	1.404
15	1.31
16	1.228



7-50 A house is heated by resistance heaters, and the amount of electricity consumed during a winter month is given. The amount of money that would be saved if this house were heated by a heat pump with a known COP is to be determined.

Assumptions The heat pump operates steadily.

Analysis The amount of heat the resistance heaters supply to the house is equal to the amount of electricity they consume. Therefore, to achieve the same heating effect, the house must be supplied with 1200 kWh of energy. A heat pump that supplied this much heat will consume electrical power in the amount of

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{1200 \text{ kWh}}{2.4} = 500 \text{ kWh}$$

which represent a savings of $1200 - 500 = 700 \text{ kWh}$. Thus the homeowner would have saved

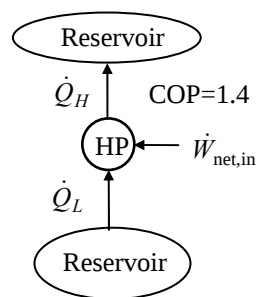
$$(700 \text{ kWh})(0.085 \text{ \$/kWh}) = \textbf{\$59.50}$$

7-51E The COP and the heating effect of a heat pump are given. The power input to the heat pump is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the definition of the coefficient of performance,

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{100,000 \text{ Btu/h}}{1.4} \left(\frac{1 \text{ hp}}{2544.5 \text{ Btu/h}} \right) = \textbf{28.1 hp}$$



7-52 The cooling effect and the rate of heat rejection of a refrigerator are given. The COP of the refrigerator is to be determined.

Assumptions The refrigerator operates steadily.

Analysis Applying the first law to the refrigerator gives

$$\dot{W}_{\text{net,in}} = \dot{Q}_H - \dot{Q}_L = 22,000 - 15,000 = 7000 \text{ kJ/h}$$

Applying the definition of the coefficient of performance,

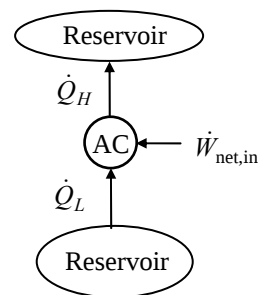
$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{15,000 \text{ kJ/h}}{7000} = \textbf{2.14}$$

7-53 The cooling effect and the power consumption of an air conditioner are given. The rate of heat rejection from this air conditioner is to be determined.

Assumptions The air conditioner operates steadily.

Analysis Applying the first law to the air conditioner gives

$$\dot{Q}_H = \dot{W}_{\text{net,in}} + \dot{Q}_L = 0.75 + 1 = \mathbf{1.75 \text{ kW}}$$



7-54 The rate of heat loss, the rate of internal heat gain, and the COP of a heat pump are given. The power input to the heat pump is to be determined.

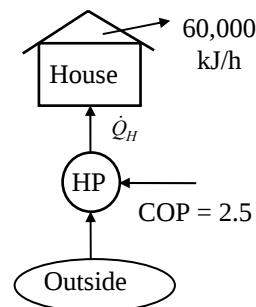
Assumptions The heat pump operates steadily.

Analysis The heating load of this heat pump system is the difference between the heat lost to the outdoors and the heat generated in the house from the people, lights, and appliances,

$$\dot{Q}_H = 60,000 - 4,000 = 56,000 \text{ kJ/h}$$

Using the definition of COP, the power input to the heat pump is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{56,000 \text{ kJ/h}}{2.5} \left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}} \right) = \mathbf{6.22 \text{ kW}}$$



7-55E An office that is being cooled adequately by a 12,000 Btu/h window air-conditioner is converted to a computer room. The number of additional air-conditioners that need to be installed is to be determined.

Assumptions 1 The computers are operated by 4 adult men. **2** The computers consume 40 percent of their rated power at any given time.

Properties The average rate of heat generation from a person seated in a room/office is 100 W (given).

Analysis The amount of heat dissipated by the computers is equal to the amount of electrical energy they consume. Therefore,

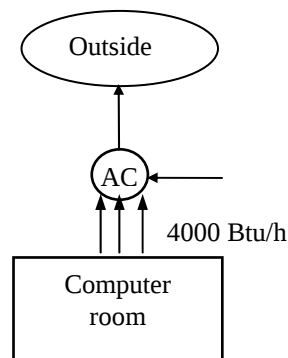
$$\dot{Q}_{\text{computers}} = (\text{Rated power}) \times (\text{Usage factor}) = (3.5 \text{ kW})(0.4) = 1.4 \text{ kW}$$

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 4 \times (100 \text{ W}) = 400 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{computers}} + \dot{Q}_{\text{people}} = 1400 + 400 = 1800 \text{ W} = 6142 \text{ Btu/h}$$

since 1 W = 3.412 Btu/h. Then noting that each available air conditioner provides 4,000 Btu/h cooling, the number of air-conditioners needed becomes

$$\begin{aligned} \text{No. of air conditioners} &= \frac{\text{Cooling load}}{\text{Cooling capacity of A/C}} = \frac{6142 \text{ Btu/h}}{4000 \text{ Btu/h}} \\ &= 1.5 \approx \mathbf{2 \text{ Air conditioners}} \end{aligned}$$



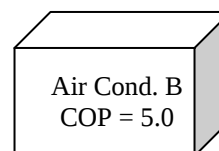
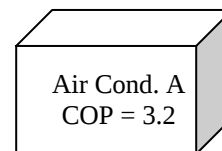
7-56 A decision is to be made between a cheaper but inefficient air-conditioner and an expensive but efficient air-conditioner for a building. The better buy is to be determined.

Assumptions The two air conditioners are comparable in all aspects other than the initial cost and the efficiency.

Analysis The unit that will cost less during its lifetime is a better buy. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period. The energy and cost savings of the more efficient air conditioner in this case is

$$\begin{aligned} \text{Energy savings} &= (\text{Annual energy usage of A}) - (\text{Annual energy usage of B}) \\ &= (\text{Annual cooling load})(1/\text{COP}_A - 1/\text{COP}_B) \\ &= (120,000 \text{ kWh/year})(1/3.2 - 1/5.0) \\ &= 13,500 \text{ kWh/year} \end{aligned}$$

$$\begin{aligned} \text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy}) \\ &= (13,500 \text{ kWh/year})(\$0.10/\text{kWh}) = \mathbf{\$1350/\text{year}} \end{aligned}$$



The installation cost difference between the two air-conditioners is

$$\text{Cost difference} = \text{Cost of B} - \text{cost of A} = 7000 - 5500 = \$1500$$

Therefore, the more efficient air-conditioner B will pay for the \$1500 cost differential in this case in about 1 year.

Discussion A cost conscious consumer will have no difficulty in deciding that the more expensive but more efficient air-conditioner B is clearly the better buy in this case since air conditioners last at least 15 years. But the decision would not be so easy if the unit cost of electricity at that location was much less than \$0.10/kWh, or if the annual air-conditioning load of the house was much less than 120,000 kWh.

7-57 Refrigerant-134a flows through the condenser of a residential heat pump unit. For a given compressor power consumption the COP of the heat pump and the rate of heat absorbed from the outside air are to be determined.

Assumptions 1 The heat pump operates steadily. 2 The kinetic and potential energy changes are zero.

Properties The enthalpies of R-134a at the condenser inlet and exit are

$$\left. \begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 35^\circ\text{C} \end{array} \right\} h_1 = 271.22 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ x_2 = 0 \end{array} \right\} h_2 = 95.47 \text{ kJ/kg}$$

Analysis (a) An energy balance on the condenser gives the heat rejected in the condenser

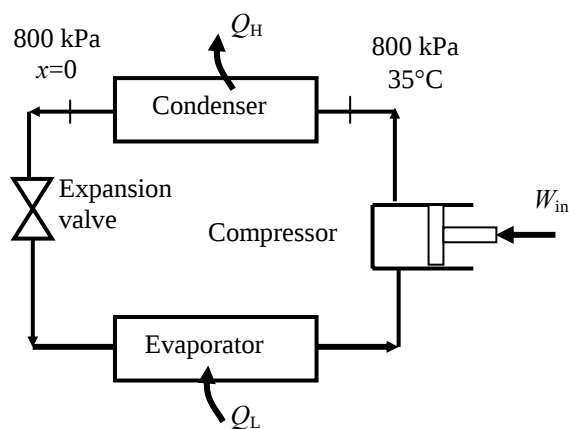
$$\dot{Q}_H = \dot{m}(h_1 - h_2) = (0.018 \text{ kg/s})(271.22 - 95.47) \text{ kJ/kg} = 3.164 \text{ kW}$$

The COP of the heat pump is

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.164 \text{ kW}}{1.2 \text{ kW}} = \mathbf{2.64}$$

(b) The rate of heat absorbed from the outside air

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 3.164 - 1.2 = \mathbf{1.96 \text{ kW}}$$



7-58 A commercial refrigerator with R-134a as the working fluid is considered. The evaporator inlet and exit states are specified. The mass flow rate of the refrigerant and the rate of heat rejected are to be determined.

Assumptions 1 The refrigerator operates steadily. 2 The kinetic and potential energy changes are zero.

Properties The properties of R-134a at the evaporator inlet and exit states are (Tables A-11 through A-13)

$$\left. \begin{array}{l} P_1 = 120 \text{ kPa} \\ x_1 = 0.2 \end{array} \right\} h_1 = 65.38 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 120 \text{ kPa} \\ T_2 = -20^\circ\text{C} \end{array} \right\} h_2 = 238.84 \text{ kJ/kg}$$

Analysis (a) The refrigeration load is

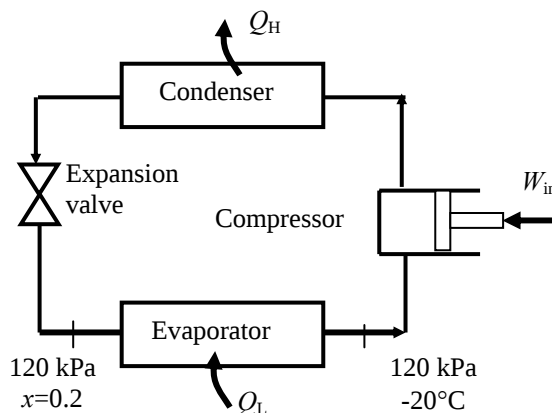
$$\dot{Q}_L = (\text{COP})\dot{W}_{\text{in}} = (1.2)(0.45 \text{ kW}) = 0.54 \text{ kW}$$

The mass flow rate of the refrigerant is determined from

$$\dot{m}_R = \frac{\dot{Q}_L}{h_2 - h_1} = \frac{0.54 \text{ kW}}{(238.84 - 65.38) \text{ kJ/kg}} = \mathbf{0.0031 \text{ kg/s}}$$

(b) The rate of heat rejected from the refrigerator is

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 0.54 + 0.45 = \mathbf{0.99 \text{ kW}}$$



Perpetual-Motion Machines

7-59C This device creates energy, and thus it is a PMM1.

7-60C This device creates energy, and thus it is a PMM1.

Reversible and Irreversible Processes

7-61C This process is irreversible. As the block slides down the plane, two things happen, (a) the potential energy of the block decreases, and (b) the block and plane warm up because of the friction between them. The potential energy that has been released can be stored in some form in the surroundings (e.g., perhaps in a spring). When we restore the system to its original condition, we must (a) restore the potential energy by lifting the block back to its original elevation, and (b) cool the block and plane back to their original temperatures.

The potential energy may be restored by returning the energy that was stored during the original process as the block decreased its elevation and released potential energy. The portion of the surroundings in which this energy had been stored would then return to its original condition as the elevation of the block is restored to its original condition.

In order to cool the block and plane to their original temperatures, we have to remove heat from the block and plane. When this heat is transferred to the surroundings, something in the surroundings has to change its state (e.g., perhaps we warm up some water in the surroundings). This change in the surroundings is permanent and cannot be undone. Hence, the original process is irreversible.

7-62C Adiabatic stirring processes are irreversible because the energy stored within the system can not be spontaneously released in a manner to cause the mass of the system to turn the **paddle wheel** in the opposite direction to do work on the surroundings.

7-63C The chemical reactions of combustion processes of a natural gas and air mixture will generate carbon dioxide, water, and other compounds and will release heat energy to a lower temperature surroundings. It is unlikely that the surroundings will return this energy to the reacting system and the products of combustion react spontaneously to reproduce the natural gas and air mixture.

7-64C No. Because it involves heat transfer through a finite temperature difference.

7-65C Because reversible processes can be approached in reality, and they form the limiting cases. Work producing devices that operate on reversible processes deliver the most work, and work consuming devices that operate on reversible processes consume the least work.

7-66C When the compression process is non-quasi equilibrium, the molecules before the piston face cannot escape fast enough, forming a high pressure region in front of the piston. It takes more work to move the piston against this high pressure region.

7-67C When an expansion process is non-quasiequilibrium, the molecules before the piston face cannot follow the piston fast enough, forming a low pressure region behind the piston. The lower pressure that pushes the piston produces less work.

7-68C The irreversibilities that occur within the system boundaries are **internal** irreversibilities; those which occur outside the system boundaries are **external** irreversibilities.

7-69C A reversible expansion or compression process cannot involve unrestrained expansion or sudden compression, and thus it is quasi-equilibrium. A quasi-equilibrium expansion or compression process, on the other hand, may involve external irreversibilities (such as heat transfer through a finite temperature difference), and thus is not necessarily reversible.

The Carnot Cycle and Carnot's Principle

7-70C The four processes that make up the Carnot cycle are isothermal expansion, reversible adiabatic expansion, isothermal compression, and reversible adiabatic compression.

7-71C They are (1) the thermal efficiency of an irreversible heat engine is lower than the efficiency of a reversible heat engine operating between the same two reservoirs, and (2) the thermal efficiency of all the reversible heat engines operating between the same two reservoirs are equal.

7-72C False. The second Carnot principle states that no heat engine cycle can have a higher thermal efficiency than the Carnot cycle operating between the same temperature limits.

7-73C Yes. The second Carnot principle states that all reversible heat engine cycles operating between the same temperature limits have the same thermal efficiency.

7-74C (a) No, (b) No. They would violate the Carnot principle.

Carnot Heat Engines

7-75C No.

7-76C The one that has a source temperature of 600°C. This is true because the higher the temperature at which heat is supplied to the working fluid of a heat engine, the higher the thermal efficiency.

7-77 Two pairs of thermal energy reservoirs are to be compared from a work-production perspective.

Assumptions The heat engine operates steadily.

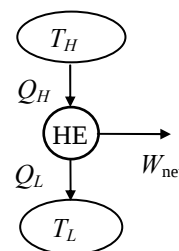
Analysis For the maximum production of work, a heat engine operating between the energy reservoirs would have to be completely reversible. Then, for the first pair of reservoirs

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{325 \text{ K}}{675 \text{ K}} = \mathbf{0.519}$$

For the second pair of reservoirs,

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{275 \text{ K}}{625 \text{ K}} = \mathbf{0.560}$$

The second pair is then capable of producing more work for each unit of heat extracted from the hot reservoir.

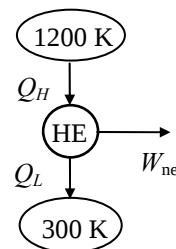


7-78 The source and sink temperatures of a power plant are given. The maximum efficiency of the plant is to be determined.

Assumptions The plant operates steadily.

Analysis The maximum efficiency this plant can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{1200 \text{ K}} = \mathbf{0.75}$$

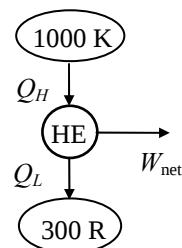


7-79 The source and sink temperatures of a power plant are given. The maximum efficiency of the plant is to be determined.

Assumptions The plant operates steadily.

Analysis The maximum efficiency this plant can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{1000 \text{ K}} = \mathbf{0.70}$$



7-80 CD EES The source and sink temperatures of a heat engine and the rate of heat supply are given. The maximum possible power output of this engine is to be determined.

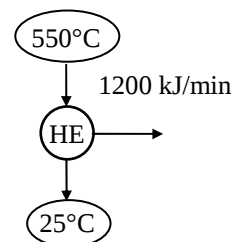
Assumptions The heat engine operates steadily.

Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{298 \text{ K}}{823 \text{ K}} = 0.638 \text{ or } 63.8\%$$

Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.638)(1200 \text{ kJ/min}) = 765.6 \text{ kJ/min} = \mathbf{12.8 \text{ kW}}$$



7-81 EES Problem 7-80 is reconsidered. The effects of the temperatures of the heat source and the heat sink on the power produced and the cycle thermal efficiency as the source temperature varies from 300°C to 1000°C and the sink temperature varies from 0°C to 50°C are to be studied. The power produced and the cycle efficiency against the source temperature for sink temperatures of 0°C, 25°C, and 50°C are to be plotted.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Input Data from the Diagram Window"

{T_H = 550 [C]

T_L = 25 [C]}

{Q_dot_H = 1200 [kJ/min]}

"First Law applied to the heat engine"

Q_dot_H - Q_dot_L - W_dot_net = 0

W_dot_net_KW = W_dot_net * convert(kJ/min, kW)

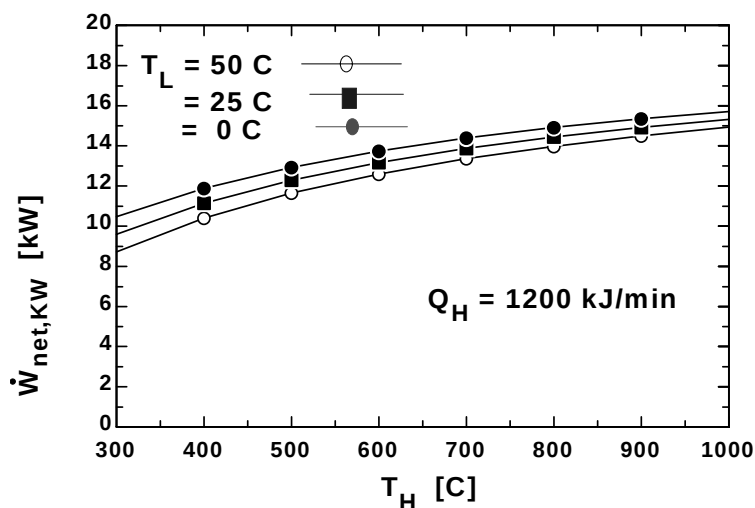
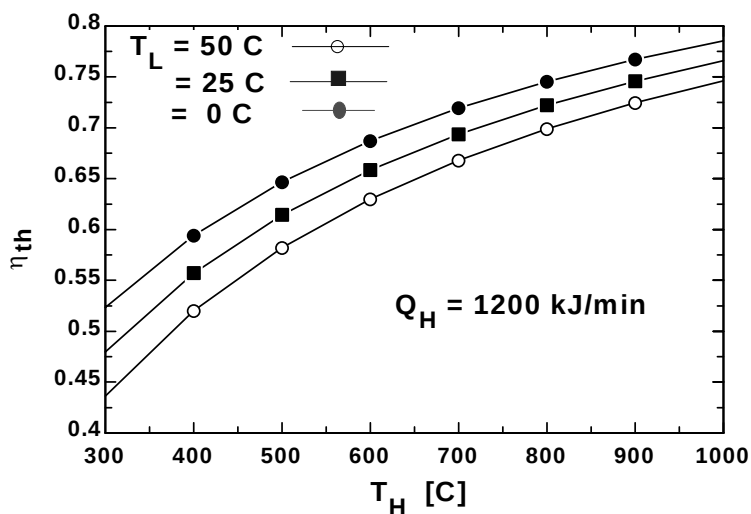
"Cycle Thermal Efficiency - Temperatures must be absolute"

eta_th = 1 - (T_L + 273)/(T_H + 273)

"Definition of cycle efficiency"

eta_th = W_dot_net / Q_dot_H

η_{th}	T_H [C]	$W_{net, KW}$ [kW]
0.52	300	10.47
0.59	400	11.89
0.65	500	12.94
0.69	600	13.75
0.72	700	14.39
0.75	800	14.91
0.77	900	15.35
0.79	1000	15.71



7-82E The sink temperature of a Carnot heat engine, the rate of heat rejection, and the thermal efficiency are given. The power output of the engine and the source temperature are to be determined.

Assumptions The Carnot heat engine operates steadily.

Analysis (a) The rate of heat input to this heat engine is determined from the definition of thermal efficiency,

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} \longrightarrow 0.75 = 1 - \frac{800 \text{ Btu/min}}{\dot{Q}_H} \longrightarrow \dot{Q}_H = 3200 \text{ Btu/min}$$

Then the power output of this heat engine can be determined from

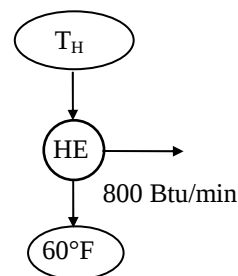
$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.75)(3200 \text{ Btu/min}) = 2400 \text{ Btu/min} = \mathbf{56.6 \text{ hp}}$$

(b) For reversible cyclic devices we have

$$\left(\frac{\dot{Q}_H}{\dot{Q}_L} \right)_{\text{rev}} = \left(\frac{T_H}{T_L} \right)$$

Thus the temperature of the source T_H must be

$$T_H = \left(\frac{\dot{Q}_H}{\dot{Q}_L} \right)_{\text{rev}} T_L = \left(\frac{3200 \text{ Btu/min}}{800 \text{ Btu/min}} \right) (520 \text{ R}) = \mathbf{2080 \text{ R}}$$



7-83E The source and sink temperatures and the power output of a reversible heat engine are given. The rate of heat input to the engine is to be determined.

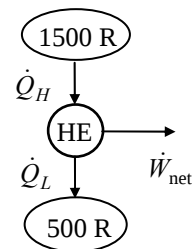
Assumptions The heat engine operates steadily.

Analysis The thermal efficiency of this reversible heat engine is determined from

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{500 \text{ R}}{1500 \text{ R}} = 0.6667$$

A rearrangement of the definition of the thermal efficiency produces

$$\eta_{\text{th,max}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_H} \longrightarrow \dot{Q}_H = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th,max}}} = \frac{5 \text{ hp}}{0.6667} \left(\frac{2544.5 \text{ Btu/h}}{1 \text{ hp}} \right) = \mathbf{19,080 \text{ Btu/h}}$$



7-84E The claim of an inventor about the operation of a heat engine is to be evaluated.

Assumptions The heat engine operates steadily.

Analysis If this engine were completely reversible, the thermal efficiency would be

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{550 \text{ R}}{1000 \text{ R}} = 0.45$$

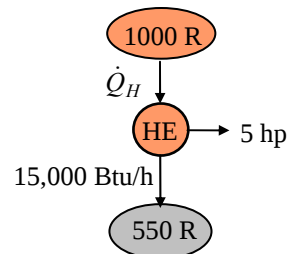
When the first law is applied to the engine above,

$$\dot{Q}_H = \dot{W}_{\text{net}} + \dot{Q}_L = (5 \text{ hp}) \left(\frac{2544.5 \text{ Btu/h}}{1 \text{ hp}} \right) + 15,000 \text{ Btu/h} = 27,720 \text{ Btu/h}$$

The actual thermal efficiency of the proposed heat engine is then

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_H} = \frac{5 \text{ hp}}{27,720 \text{ Btu/h}} \left(\frac{2544.5 \text{ Btu/h}}{1 \text{ hp}} \right) = 0.459$$

Since the thermal efficiency of the proposed heat engine is greater than that of a completely reversible heat engine which uses the same isothermal energy reservoirs, **the inventor's claim is invalid.**



7-85 The claim that the efficiency of a completely reversible heat engine can be doubled by doubling the temperature of the energy source is to be evaluated.

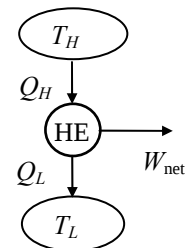
Assumptions The heat engine operates steadily.

Analysis The upper limit for the thermal efficiency of any heat engine occurs when a completely reversible engine operates between the same energy reservoirs. The thermal efficiency of this completely reversible engine is given by

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} = \frac{T_H - T_L}{T_H}$$

If we were to double the absolute temperature of the high temperature energy reservoir, the new thermal efficiency would be

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{2T_H} = \frac{2T_H - T_L}{2T_H} < 2 \frac{T_H - T_L}{T_H}$$



The thermal efficiency is then **not doubled** as the temperature of the high temperature reservoir is doubled.

7-86 An inventor claims to have developed a heat engine. The inventor reports temperature, heat transfer, and work output measurements. The claim is to be evaluated.

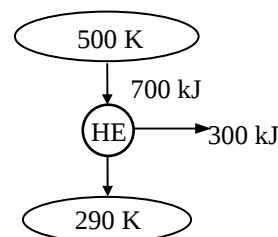
Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{290 \text{ K}}{500 \text{ K}} = 0.42 \text{ or } 42\%$$

The actual thermal efficiency of the heat engine in question is

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_H} = \frac{300 \text{ kJ}}{700 \text{ kJ}} = 0.429 \text{ or } 42.9\%$$

which is greater than the maximum possible thermal efficiency. Therefore, this heat engine is a PMM2 and the claim is **false**.



7-87E An inventor claims to have developed a heat engine. The inventor reports temperature, heat transfer, and work output measurements. The claim is to be evaluated.

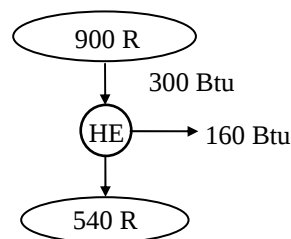
Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540 \text{ R}}{900 \text{ R}} = 0.40 \text{ or } 40\%$$

The actual thermal efficiency of the heat engine in question is

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_H} = \frac{160 \text{ Btu}}{300 \text{ Btu}} = 0.533 \text{ or } 53.3\%$$

which is greater than the maximum possible thermal efficiency. Therefore, this heat engine is a PMM2 and the claim is **false**.



7-88 A geothermal power plant uses geothermal liquid water at 160°C at a specified rate as the heat source. The actual and maximum possible thermal efficiencies and the rate of heat rejected from this power plant are to be determined.

Assumptions 1 The power plant operates steadily. 2 The kinetic and potential energy changes are zero. 3 Steam properties are used for geothermal water.

Properties Using saturated liquid properties, the source and the sink state enthalpies of geothermal water are (Table A-4)

$$\left. \begin{array}{l} T_{\text{source}} = 160^{\circ}\text{C} \\ x_{\text{source}} = 0 \end{array} \right\} h_{\text{source}} = 675.47 \text{ kJ/kg}$$

$$\left. \begin{array}{l} T_{\text{sink}} = 25^{\circ}\text{C} \\ x_{\text{sink}} = 0 \end{array} \right\} h_{\text{sink}} = 104.83 \text{ kJ/kg}$$

Analysis (a) The rate of heat input to the plant may be taken as the enthalpy difference between the source and the sink for the power plant

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{geo}} (h_{\text{source}} - h_{\text{sink}}) = (440 \text{ kg/s})(675.47 - 104.83) \text{ kJ/kg} = 251,083 \text{ kW}$$

The actual thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_{\text{in}}} = \frac{22 \text{ MW}}{251.083 \text{ MW}} = \mathbf{0.0876 = 8.8\%}$$

(b) The maximum thermal efficiency is the thermal efficiency of a reversible heat engine operating between the source and sink temperatures

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(25 + 273) \text{ K}}{(160 + 273) \text{ K}} = \mathbf{0.312 = 31.2\%}$$

(c) Finally, the rate of heat rejection is

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net,out}} = 251.1 - 22 = \mathbf{229.1 \text{ MW}}$$

Carnot Refrigerators and Heat Pumps

7-89C By increasing T_L or by decreasing T_H .

7-90C It is the COP that a Carnot refrigerator would have, $\text{COP}_R = \frac{1}{T_H/T_L - 1}$.

7-91C No. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the refrigerator. In reality, the work consumed by the refrigerator will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

7-92C No. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the refrigerator. In reality, the work consumed by the refrigerator will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

7-93C Bad idea. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the heat pump. In reality, the work consumed by the heat pump will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

7-94 The minimum work per unit of heat transfer from the low-temperature source for a heat pump is to be determined.

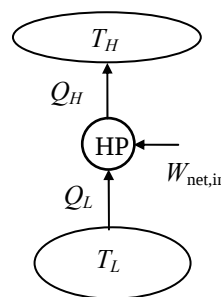
Assumptions The heat pump operates steadily.

Analysis Application of the first law gives

$$\frac{W_{\text{net,in}}}{Q_L} = \frac{Q_H - Q_L}{Q_L} = \frac{Q_H}{Q_L} - 1$$

For the minimum work input, this heat pump would be completely reversible and the thermodynamic definition of temperature would reduce the preceding expression to

$$\frac{W_{\text{net,in}}}{Q_L} = \frac{T_H}{T_L} - 1 = \frac{535 \text{ K}}{460 \text{ K}} - 1 = \mathbf{0.163}$$



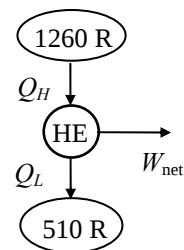
7-95E The claim of a thermodynamicist regarding to the thermal efficiency of a heat engine is to be evaluated.

Assumptions The plant operates steadily.

Analysis The maximum thermal efficiency would be achieved when this engine is completely reversible. When this is the case,

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{510 \text{ R}}{1260 \text{ R}} = 0.595$$

Since the thermal efficiency of the actual engine is less than this, this engine is **possible**.



7-96 An expression for the COP of a completely reversible refrigerator in terms of the thermal-energy reservoir temperatures, T_L and T_H is to be derived.

Assumptions The refrigerator operates steadily.

Analysis Application of the first law to the completely reversible refrigerator yields

$$W_{\text{net,in}} = Q_H - Q_L$$

This result may be used to reduce the coefficient of performance,

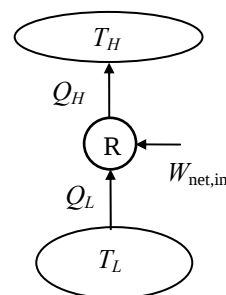
$$\text{COP}_{\text{R,rev}} = \frac{Q_L}{W_{\text{net,in}}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{Q_H / Q_L - 1}$$

Since this refrigerator is completely reversible, the thermodynamic definition of temperature tells us that,

$$\frac{Q_H}{Q_L} = \frac{T_H}{T_L}$$

When this is substituted into the COP expression, the result is

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H / T_L - 1} = \frac{T_L}{T_H - T_L}$$



7-97 The rate of cooling provided by a reversible refrigerator with specified reservoir temperatures is to be determined.

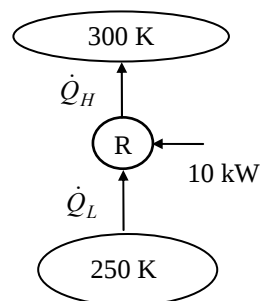
Assumptions The refrigerator operates steadily.

Analysis The COP of this reversible refrigerator is

$$\text{COP}_{\text{R,max}} = \frac{T_L}{T_H - T_L} = \frac{250 \text{ K}}{300 \text{ K} - 250 \text{ K}} = 5$$

Rearranging the definition of the refrigerator coefficient of performance gives

$$\dot{Q}_L = \text{COP}_{\text{R,max}} \dot{W}_{\text{net,in}} = (5)(10 \text{ kW}) = \mathbf{50 \text{ kW}}$$



7-98 The refrigerated space and the environment temperatures for a refrigerator and the rate of heat removal from the refrigerated space are given. The minimum power input required is to be determined.

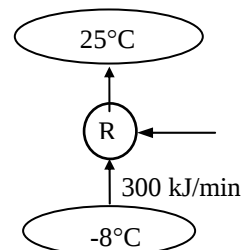
Assumptions The refrigerator operates steadily.

Analysis The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(25 + 273 \text{ K})/(-8 + 273 \text{ K}) - 1} = 8.03$$

The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{300 \text{ kJ/min}}{8.03} = 37.36 \text{ kJ/min} = \mathbf{0.623 \text{ kW}}$$



7-99 The refrigerated space temperature, the COP, and the power input of a Carnot refrigerator are given. The rate of heat removal from the refrigerated space and its temperature are to be determined.

Assumptions The refrigerator operates steadily.

Analysis (a) The rate of heat removal from the refrigerated space is determined from the definition of the COP of a refrigerator,

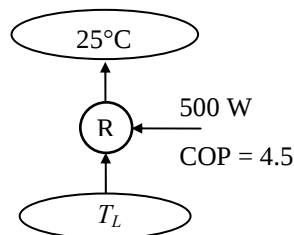
$$\dot{Q}_L = \text{COP}_R \dot{W}_{\text{net,in}} = (4.5)(0.5 \text{ kW}) = 2.25 \text{ kW} = \mathbf{135 \text{ kJ/min}}$$

(b) The temperature of the refrigerated space T_L is determined from the coefficient of performance relation for a Carnot refrigerator,

$$\text{COP}_{R,\text{rev}} = \frac{1}{(T_H/T_L) - 1} \longrightarrow 4.5 = \frac{1}{(25 + 273 \text{ K})/T_L - 1}$$

It yields

$$T_L = 243.8 \text{ K} = \mathbf{-29.2^\circ\text{C}}$$

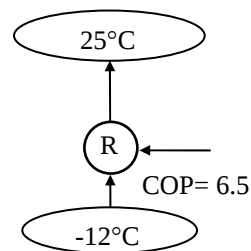


7-100 An inventor claims to have developed a refrigerator. The inventor reports temperature and COP measurements. The claim is to be evaluated.

Analysis The highest coefficient of performance a refrigerator can have when removing heat from a cool medium at -12°C to a warmer medium at 25°C is

$$\text{COP}_{R,\text{max}} = \text{COP}_{R,\text{rev}} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(25 + 273 \text{ K})/(-12 + 273 \text{ K}) - 1} = 7.1$$

The COP claimed by the inventor is 6.5, which is below this maximum value, thus the claim is **reasonable**. However, it is not probable.



7-101E An air-conditioning system maintains a house at a specified temperature. The rate of heat gain of the house and the rate of internal heat generation are given. The maximum power input required is to be determined.

Assumptions The air-conditioner operates steadily.

Analysis The power input to an air-conditioning system will be a minimum when the air-conditioner operates in a reversible manner. The coefficient of performance of a reversible air-conditioner (or refrigerator) depends on the temperature limits in the cycle only, and is determined from

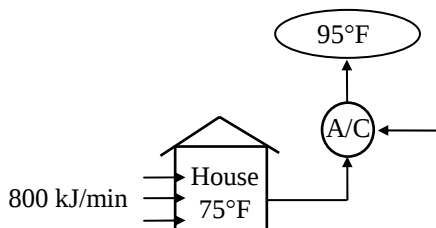
$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(95 + 460 \text{ R})/(75 + 460 \text{ R}) - 1} = 26.75$$

The cooling load of this air-conditioning system is the sum of the heat gain from the outside and the heat generated within the house,

$$\dot{Q}_L = 800 + 100 = 900 \text{ Btu/min}$$

The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{900 \text{ Btu/min}}{26.75} = 33.6 \text{ Btu/min} = \mathbf{0.79 \text{ hp}}$$



7-102 A heat pump maintains a house at a specified temperature. The rate of heat loss of the house is given. The minimum power input required is to be determined.

Assumptions The heat pump operates steadily.

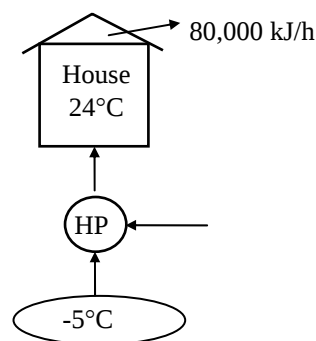
Analysis The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L/T_H)} = \frac{1}{1 - (-5 + 273 \text{ K})/(24 + 273 \text{ K})} = 10.2$$

The required power input to this reversible heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{80,000 \text{ kJ/h}}{10.2} \left(\frac{1 \text{ h}}{3600 \text{ s}} \right) = \mathbf{2.18 \text{ kW}}$$

which is the *minimum* power input required.



7-103 A heat pump maintains a house at a specified temperature. The rate of heat loss of the house and the power consumption of the heat pump are given. It is to be determined if this heat pump can do the job.

Assumptions The heat pump operates steadily.

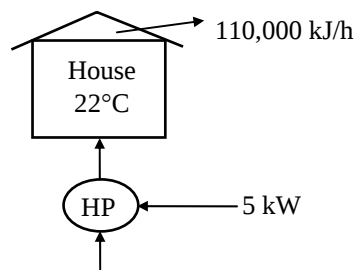
Analysis The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (2 + 273 \text{ K}) / (22 + 273 \text{ K})} = 14.75$$

The required power input to this reversible heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{110,000 \text{ kJ/h}}{14.75} \left(\frac{1 \text{ h}}{3600 \text{ s}} \right) = \mathbf{2.07 \text{ kW}}$$

This heat pump is **powerful enough** since $5 \text{ kW} > 2.07 \text{ kW}$.



7-104E The power required by a reversible refrigerator with specified reservoir temperatures is to be determined.

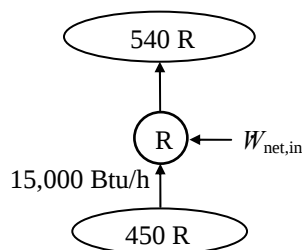
Assumptions The refrigerator operates steadily.

Analysis The COP of this reversible refrigerator is

$$\text{COP}_{\text{R,max}} = \frac{T_L}{T_H - T_L} = \frac{450 \text{ R}}{540 \text{ R} - 450 \text{ R}} = 5$$

Using this result in the coefficient of performance expression yields

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{15,000 \text{ Btu/h}}{5} \left(\frac{1 \text{ kW}}{3412.14 \text{ Btu/h}} \right) = \mathbf{0.879 \text{ kW}}$$



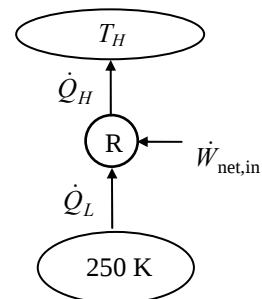
7-105 The COP of a completely reversible refrigerator as a function of the temperature of the sink is to be calculated and plotted.

Assumptions The refrigerator operates steadily.

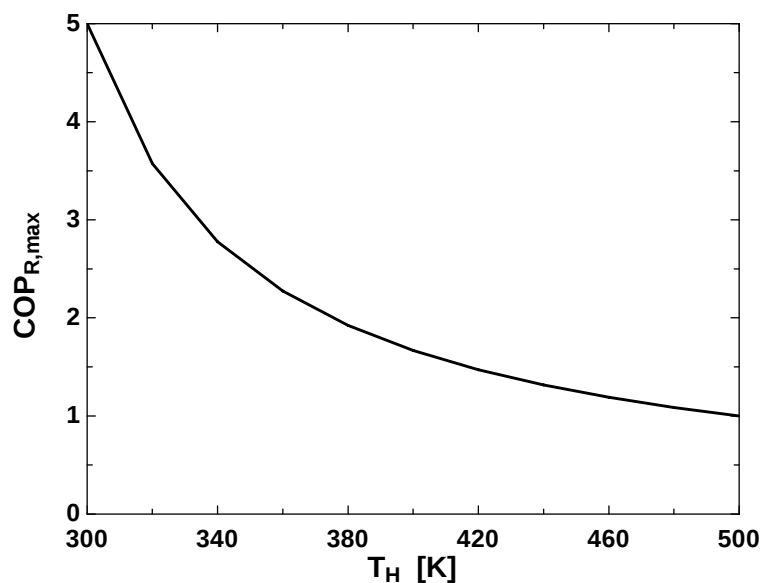
Analysis The coefficient of performance for this completely reversible refrigerator is given by

$$\text{COP}_{R,\max} = \frac{T_L}{T_H - T_L} = \frac{250 \text{ K}}{T_H - 250 \text{ K}} = 5$$

Using EES, we tabulate and plot the variation of COP with the sink temperature as follows:



T_H [K]	$\text{COP}_{R,\max}$
300	5
320	3.571
340	2.778
360	2.273
380	1.923
400	1.667
420	1.471
440	1.316
460	1.19
480	1.087
500	1



7-106 A reversible heat pump is considered. The temperature of the source and the rate of heat transfer to the sink are to be determined.

Assumptions The heat pump operates steadily.

Analysis Combining the first law, the expression for the coefficient of performance, and the thermodynamic temperature scale gives

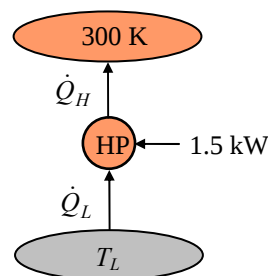
$$\text{COP}_{\text{HP,max}} = \frac{T_H}{T_H - T_L}$$

which upon rearrangement becomes

$$T_L = T_H \left(1 - \frac{1}{\text{COP}_{\text{HP,max}}} \right) = (300 \text{ K}) \left(1 - \frac{1}{1.6} \right) = \mathbf{112.5 \text{ K}}$$

Based upon the definition of the heat pump coefficient of performance,

$$\dot{Q}_H = \text{COP}_{\text{HP,max}} \dot{W}_{\text{net,in}} = (1.6)(1.5 \text{ kW}) = \mathbf{2.4 \text{ kW}}$$



7-107 A heat pump that consumes 5-kW of power when operating maintains a house at a specified temperature. The house is losing heat in proportion to the temperature difference between the indoors and the outdoors. The lowest outdoor temperature for which this heat pump can do the job is to be determined.

Assumptions The heat pump operates steadily.

Analysis Denoting the outdoor temperature by T_L , the heating load of this house can be expressed as

$$\dot{Q}_H = (5400 \text{ kJ/h} \cdot \text{K})(294 - T_L) = (1.5 \text{ kW/K})(294 - T_L) \text{ K}$$

The coefficient of performance of a Carnot heat pump depends on the temperature limits in the cycle only, and can be expressed as

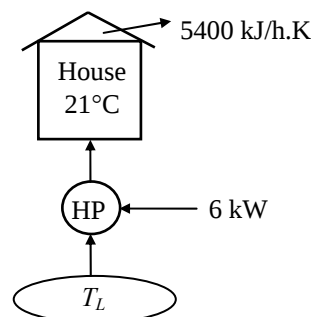
$$\text{COP}_{\text{HP}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - T_L / (294 \text{ K})}$$

or, as

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{net,in}}} = \frac{(1.5 \text{ kW/K})(294 - T_L) \text{ K}}{6 \text{ kW}}$$

Equating the two relations above and solving for T_L , we obtain

$$T_L = 259.7 \text{ K} = \mathbf{-13.3^\circ \text{C}}$$



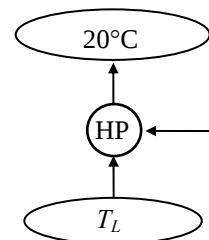
7-108 A heat pump maintains a house at a specified temperature in winter. The maximum COPs of the heat pump for different outdoor temperatures are to be determined.

Analysis The coefficient of performance of a heat pump will be a maximum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined for all three cases above to be

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (10 + 273\text{K}) / (20 + 273\text{K})} = \mathbf{29.3}$$

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (-5 + 273\text{K}) / (20 + 273\text{K})} = \mathbf{11.7}$$

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (-30 + 273\text{K}) / (20 + 273\text{K})} = \mathbf{5.86}$$



7-109E A heat pump maintains a house at a specified temperature. The rate of heat loss of the house is given. The minimum power inputs required for different source temperatures are to be determined.

Assumptions The heat pump operates steadily.

Analysis (a) The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. If the outdoor air at 25°F is used as the heat source, the COP of the heat pump and the required power input are determined to be

$$\begin{aligned} \text{COP}_{\text{HP,max}} = \text{COP}_{\text{HP,rev}} &= \frac{1}{1 - (T_L / T_H)} \\ &= \frac{1}{1 - (25 + 460\text{ R}) / (78 + 460\text{ R})} = \mathbf{10.15} \end{aligned}$$

and

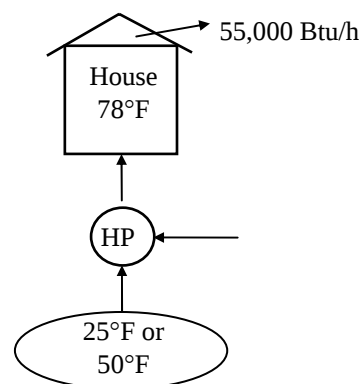
$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55,000\text{ Btu/h}}{10.15} \left(\frac{1\text{ hp}}{2545\text{ Btu/h}} \right) = \mathbf{2.13\text{ hp}}$$

(b) If the well-water at 50°F is used as the heat source, the COP of the heat pump and the required power input are determined to be

$$\text{COP}_{\text{HP,max}} = \text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (50 + 460\text{ R}) / (78 + 460\text{ R})} = \mathbf{19.2}$$

and

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55,000\text{ Btu/h}}{19.2} \left(\frac{1\text{ hp}}{2545\text{ Btu/h}} \right) = \mathbf{1.13\text{ hp}}$$



7-110 A Carnot heat pump consumes 8-kW of power when operating, and maintains a house at a specified temperature. The average rate of heat loss of the house in a particular day is given. The actual running time of the heat pump that day, the heating cost, and the cost if resistance heating is used instead are to be determined.

Analysis (a) The coefficient of performance of this Carnot heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (2 + 273 \text{ K}) / (20 + 273 \text{ K})} = 16.3$$

The amount of heat the house lost that day is

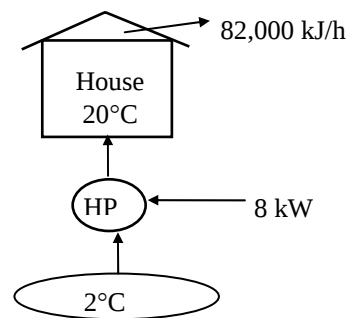
$$Q_H = \dot{Q}_H (1 \text{ day}) = (82,000 \text{ kJ/h})(24 \text{ h}) = 1,968,000 \text{ kJ}$$

Then the required work input to this Carnot heat pump is determined from the definition of the coefficient of performance to be

$$W_{\text{net,in}} = \frac{Q_H}{\text{COP}_{\text{HP}}} = \frac{1,968,000 \text{ kJ}}{16.3} = 120,736 \text{ kJ}$$

Thus the length of time the heat pump ran that day is

$$\Delta t = \frac{W_{\text{net,in}}}{\dot{W}_{\text{net,in}}} = \frac{120,736 \text{ kJ}}{8 \text{ kJ/s}} = 15,092 \text{ s} = \mathbf{4.19 \text{ h}}$$



(b) The total heating cost that day is

$$\text{Cost} = W \times \text{price} = (\dot{W}_{\text{net,in}} \times \Delta t)(\text{price}) = (8 \text{ kW})(4.19 \text{ h})(0.085 \text{ \$/kWh}) = \mathbf{\$2.85}$$

(c) If resistance heating were used, the entire heating load for that day would have to be met by electrical energy. Therefore, the heating system would consume 1,968,000 kJ of electricity that would cost

$$\text{New Cost} = Q_H \times \text{price} = (1,968,000 \text{ kJ}) \left(\frac{1 \text{ kWh}}{3600 \text{ kJ}} \right) (0.085 \text{ \$/kWh}) = \mathbf{\$46.47}$$

7-111 A Carnot heat engine is used to drive a Carnot refrigerator. The maximum rate of heat removal from the refrigerated space and the total rate of heat rejection to the ambient air are to be determined.

Assumptions The heat engine and the refrigerator operate steadily.

Analysis (a) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{1173 \text{ K}} = 0.744$$

Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.744)(800 \text{ kJ/min}) = 595.2 \text{ kJ/min}$$

which is also the power input to the refrigerator, $\dot{W}_{\text{net,in}}$.

The rate of heat removal from the refrigerated space will be a maximum if a Carnot refrigerator is used. The COP of the Carnot refrigerator is

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(27 + 273 \text{ K})/(-5 + 273 \text{ K}) - 1} = 8.37$$

Then the rate of heat removal from the refrigerated space becomes

$$\dot{Q}_{L,R} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.37)(595.2 \text{ kJ/min}) = \mathbf{4982 \text{ kJ/min}}$$

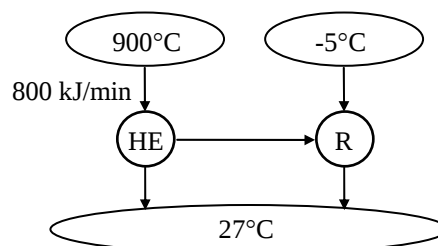
(b) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,R}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 800 - 595.2 = 204.8 \text{ kJ/min}$$

$$\dot{Q}_{H,R} = \dot{Q}_{L,R} + \dot{W}_{\text{net,in}} = 4982 + 595.2 = 5577.2 \text{ kJ/min}$$

and

$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,R} = 204.8 + 5577.2 = \mathbf{5782 \text{ kJ/min}}$$



7-112E A Carnot heat engine is used to drive a Carnot refrigerator. The maximum rate of heat removal from the refrigerated space and the total rate of heat rejection to the ambient air are to be determined.

Assumptions The heat engine and the refrigerator operate steadily.

Analysis (a) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540 \text{ R}}{2160 \text{ R}} = 0.75$$

Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.75)(700 \text{ Btu/min}) = 525 \text{ Btu/min}$$

which is also the power input to the refrigerator, $\dot{W}_{\text{net,in}}$.

The rate of heat removal from the refrigerated space will be a maximum if a Carnot refrigerator is used. The COP of the Carnot refrigerator is

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(80 + 460 \text{ R})/(20 + 460 \text{ R}) - 1} = 8.0$$

Then the rate of heat removal from the refrigerated space becomes

$$\dot{Q}_{L,R} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.0)(525 \text{ Btu/min}) = \mathbf{4200 \text{ Btu/min}}$$

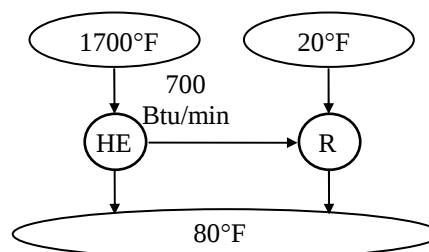
(b) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,R}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 700 - 525 = 175 \text{ Btu/min}$$

$$\dot{Q}_{H,R} = \dot{Q}_{L,R} + \dot{W}_{\text{net,in}} = 4200 + 525 = 4725 \text{ Btu/min}$$

and

$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,R} = 175 + 4725 = \mathbf{4900 \text{ Btu/min}}$$



7-113 An air-conditioner with R-134a as the working fluid is considered. The compressor inlet and exit states are specified. The actual and maximum COPs and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

Assumptions 1 The air-conditioner operates steadily. 2 The kinetic and potential energy changes are zero.

Properties The properties of R-134a at the compressor inlet and exit states are (Tables A-11 through A-13)

$$\begin{aligned} P_1 = 500 \text{ kPa} \quad \left. \begin{aligned} h_1 &= 259.30 \text{ kJ/kg} \\ x_1 = 1 \end{aligned} \right\} \quad v_1 = 0.04112 \text{ m}^3/\text{kg} \\ P_2 = 1.2 \text{ MPa} \quad \left. \begin{aligned} h_2 &= 278.27 \text{ kJ/kg} \\ T_2 = 50^\circ\text{C} \end{aligned} \right\} \end{aligned}$$

Analysis (a) The mass flow rate of the refrigerant and the power consumption of the compressor are

$$\dot{m}_R = \frac{\dot{V}_1}{v_1} = \frac{100 \text{ L/min} \left(\frac{1 \text{ m}^3}{1000 \text{ L}} \right) \left(\frac{1 \text{ min}}{60 \text{ s}} \right)}{0.04112 \text{ m}^3/\text{kg}} = 0.04053 \text{ kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}_R (h_2 - h_1) = (0.04053 \text{ kg/s})(278.27 - 259.30) \text{ kJ/kg} = 0.7686 \text{ kW}$$

The heat gains to the room must be rejected by the air-conditioner. That is,

$$\dot{Q}_L = \dot{Q}_{\text{heat}} + \dot{Q}_{\text{equipment}} = (250 \text{ kJ/min}) \left(\frac{1 \text{ min}}{60 \text{ s}} \right) + 0.9 \text{ kW} = 5.067 \text{ kW}$$

Then, the actual COP becomes

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.067 \text{ kW}}{0.7686 \text{ kW}} = \mathbf{6.59}$$

(b) The COP of a reversible refrigerator operating between the same temperature limits is

$$\text{COP}_{\text{max}} = \frac{1}{T_H / T_L - 1} = \frac{1}{(34 + 273) / (26 + 273) - 1} = \mathbf{37.4}$$

(c) The minimum power input to the compressor for the same refrigeration load would be

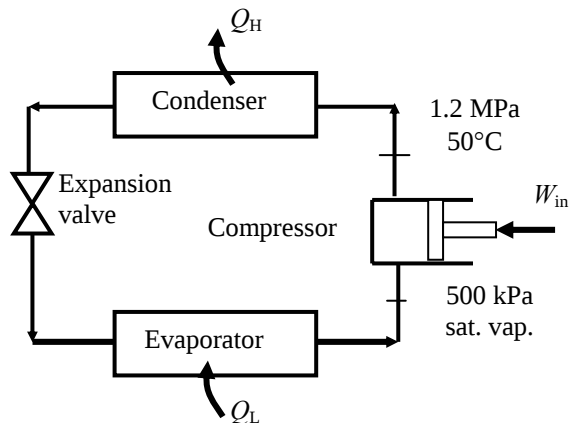
$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{max}}} = \frac{5.067 \text{ kW}}{37.38} = 0.1356 \text{ kW}$$

The minimum mass flow rate is

$$\dot{m}_{R,\text{min}} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.1356 \text{ kW}}{(278.27 - 259.30) \text{ kJ/kg}} = 0.007149 \text{ kg/s}$$

Finally, the minimum volume flow rate at the compressor inlet is

$$\dot{V}_{\text{min},1} = \dot{m}_{R,\text{min}} v_1 = (0.007149 \text{ kg/s})(0.04112 \text{ m}^3/\text{kg}) = 0.000294 \text{ m}^3/\text{s} = \mathbf{17.64 \text{ L/min}}$$



Review Problems

7-114 A Carnot heat engine cycle is executed in a steady-flow system with steam. The thermal efficiency and the mass flow rate of steam are given. The net power output of the engine is to be determined.

Assumptions All components operate steadily.

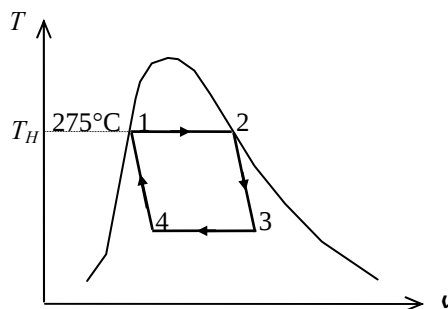
Properties The enthalpy of vaporization h_{fg} of water at 275°C is 1574.5 kJ/kg (Table A-4).

Analysis The enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that T or P . Therefore, the rate of heat transfer to the steam during heat addition process is

$$\dot{Q}_H = \dot{m} h_{fg@275^\circ\text{C}} = (3 \text{ kg/s})(1574.5 \text{ kJ/kg}) = 4723 \text{ kJ/s}$$

Then the power output of this heat engine becomes

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.30)(4723 \text{ kW}) = \mathbf{1417 \text{ kW}}$$



7-115 A heat pump with a specified COP is to heat a house. The rate of heat loss of the house and the power consumption of the heat pump are given. The time it will take for the interior temperature to rise from 3°C to 22°C is to be determined.

Assumptions 1 Air is an ideal gas with constant specific heats at room temperature. 2 The house is well-sealed so that no air leaks in or out. 3 The COP of the heat pump remains constant during operation.

Properties The constant volume specific heat of air at room temperature is $c_v = 0.718 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2)

Analysis The house is losing heat at a rate of

$$\dot{Q}_{\text{Loss}} = 40,000 \text{ kJ/h} = 11.11 \text{ kJ/s}$$

The rate at which this heat pump supplies heat is

$$\dot{Q}_H = \text{COP}_{\text{HP}} \dot{W}_{\text{net,in}} = (2.4)(8 \text{ kW}) = 19.2 \text{ kW}$$

That is, this heat pump can supply heat at a rate of 19.2 kJ/s. Taking the house as the system (a closed system), the energy balance can be written as

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ Q_{\text{in}} - Q_{\text{out}} &= \Delta U = m(u_2 - u_1) \\ Q_{\text{in}} - Q_{\text{out}} &= mc_v(T_2 - T_1) \\ (\dot{Q}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t &= mc_v(T_2 - T_1) \end{aligned}$$

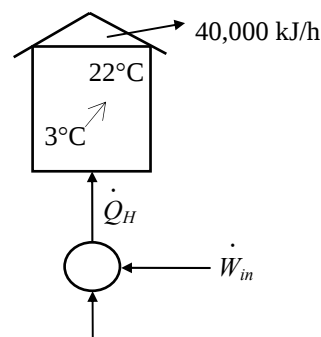
Substituting,

$$(19.2 - 11.11 \text{ kJ/s})\Delta t = (2000 \text{ kg})(0.718 \text{ kJ/kg} \cdot ^\circ\text{C})(22 - 3)^\circ\text{C}$$

Solving for Δt , it will take

$$\Delta t = 3373 \text{ s} = \mathbf{0.937 \text{ h}}$$

for the temperature in the house to rise to 22°C.



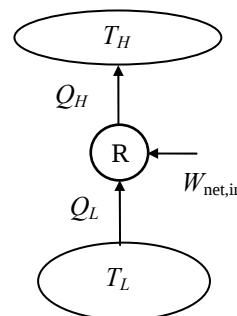
7-116 The claim of a manufacturer of ice cream freezers regarding the COP of these freezers is to be evaluated.

Assumptions The refrigerator operates steadily.

Analysis The maximum refrigerator coefficient of performance would occur if the refrigerator were completely reversible,

$$\text{COP}_{\text{R,max}} = \frac{T_L}{T_H - T_L} = \frac{250 \text{ K}}{300 \text{ K} - 250 \text{ K}} = 5$$

Since the claimed COP is less than this maximum, this refrigerator is **possible**.



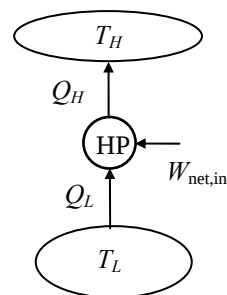
7-117 The claim of a heat pump designer regarding the COP of the heat pump is to be evaluated.

Assumptions The heat pump operates steadily.

Analysis The maximum heat pump coefficient of performance would occur if the heat pump were completely reversible,

$$\text{COP}_{\text{HP,max}} = \frac{T_H}{T_H - T_L} = \frac{300 \text{ K}}{300 \text{ K} - 260 \text{ K}} = 7.5$$

Since the claimed COP is less than this maximum, this heat pump is **possible**.



7-118E The operating conditions of a heat pump are given. The minimum temperature of the source that satisfies the second law of thermodynamics is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the first law to this heat pump gives

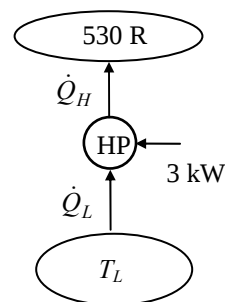
$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 100,000 \text{ Btu/h} - (3 \text{ kW}) \left(\frac{3412.14 \text{ Btu/h}}{1 \text{ kW}} \right) = 89,760 \text{ Btu/h}$$

In the reversible case we have

$$\frac{T_L}{T_H} = \frac{\dot{Q}_L}{\dot{Q}_H}$$

Then the minimum temperature may be determined to be

$$T_L = T_H \frac{\dot{Q}_L}{\dot{Q}_H} = (530 \text{ R}) \frac{89,760 \text{ Btu/h}}{100,000 \text{ Btu/h}} = \mathbf{476 \text{ R}}$$



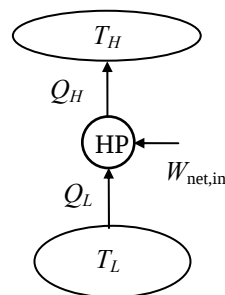
7-119 The claim of a thermodynamicist regarding the COP of a heat pump is to be evaluated.

Assumptions The heat pump operates steadily.

Analysis The maximum heat pump coefficient of performance would occur if the heat pump were completely reversible,

$$\text{COP}_{\text{HP,max}} = \frac{T_H}{T_H - T_L} = \frac{293 \text{ K}}{293 \text{ K} - 273 \text{ K}} = 14.7$$

Since the claimed COP is less than this maximum, the claim is **valid**.



7-120 A Carnot heat engine cycle is executed in a closed system with a fixed mass of R-134a. The thermal efficiency of the cycle is given. The net work output of the engine is to be determined.

Assumptions All components operate steadily.

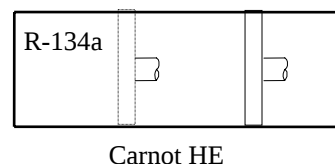
Properties The enthalpy of vaporization of R-134a at 50°C is $h_{fg} = 151.79$ kJ/kg (Table A-11).

Analysis The enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that T or P . Therefore, the amount of heat transfer to R-134a during the heat addition process of the cycle is

$$Q_H = mh_{fg@50^\circ\text{C}} = (0.01 \text{ kg})(151.79 \text{ kJ/kg}) = 1.518 \text{ kJ}$$

Then the work output of this heat engine becomes

$$W_{\text{net,out}} = \eta_{\text{th}} Q_H = (0.15)(1.518 \text{ kJ}) = \mathbf{0.228 \text{ kJ}}$$



7-121 A heat pump with a specified COP and power consumption is used to heat a house. The time it takes for this heat pump to raise the temperature of a cold house to the desired level is to be determined.

Assumptions 1 Air is an ideal gas with constant specific heats at room temperature. 2 The heat loss of the house during the warm-up period is negligible. 3 The house is well-sealed so that no air leaks in or out.

Properties The constant volume specific heat of air at room temperature is $c_v = 0.718$ kJ/kg·°C.

Analysis Since the house is well-sealed (constant volume), the total amount of heat that needs to be supplied to the house is

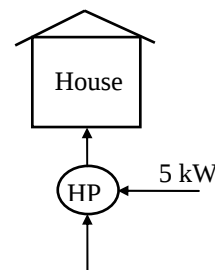
$$Q_H = (mc_v \Delta T)_{\text{house}} = (1500 \text{ kg})(0.718 \text{ kJ/kg} \cdot ^\circ\text{C})(22 - 7)^\circ\text{C} = 16,155 \text{ kJ}$$

The rate at which this heat pump supplies heat is

$$\dot{Q}_H = \text{COP}_{\text{HP}} \dot{W}_{\text{net,in}} = (2.8)(5 \text{ kW}) = 14 \text{ kW}$$

That is, this heat pump can supply 14 kJ of heat per second. Thus the time required to supply 16,155 kJ of heat is

$$\Delta t = \frac{Q_H}{\dot{Q}_H} = \frac{16,155 \text{ kJ}}{14 \text{ kJ/s}} = 1154 \text{ s} = \mathbf{19.2 \text{ min}}$$

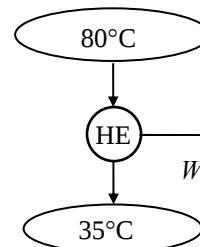


7-122 A solar pond power plant operates by absorbing heat from the hot region near the bottom, and rejecting waste heat to the cold region near the top. The maximum thermal efficiency that the power plant can have is to be determined.

Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{th,max} = \eta_{th,C} = 1 - \frac{T_L}{T_H} = 1 - \frac{308 \text{ K}}{353 \text{ K}} = 0.127 \text{ or } \mathbf{12.7\%}$$

In reality, the temperature of the working fluid must be above 35°C in the condenser, and below 80°C in the boiler to allow for any effective heat transfer. Therefore, the maximum efficiency of the actual heat engine will be lower than the value calculated above.



7-123 A Carnot heat engine cycle is executed in a closed system with a fixed mass of steam. The net work output of the cycle and the ratio of sink and source temperatures are given. The low temperature in the cycle is to be determined.

Assumptions The engine is said to operate on the Carnot cycle, which is totally reversible.

Analysis The thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{1}{2} = 0.5$$

Also,

$$\eta_{th} = \frac{W}{Q_H} \longrightarrow Q_H = \frac{W}{\eta_{th}} = \frac{25 \text{ kJ}}{0.5} = 50 \text{ kJ}$$

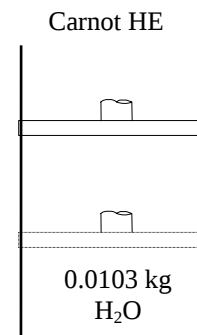
$$Q_L = Q_H - W = 50 - 25 = 25 \text{ kJ}$$

and

$$q_L = \frac{Q_L}{m} = \frac{25 \text{ kJ}}{0.0103 \text{ kg}} = 2427.2 \text{ kJ/kg} = h_{fg@T_L}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that T or P . Therefore, T_L is the temperature that corresponds to the h_{fg} value of 2427.2 kJ/kg, and is determined from the steam tables to be

$$T_L = \mathbf{31.3^\circ\text{C}}$$



7-124 EES Problem 7-123 is reconsidered. The effect of the net work output on the required temperature of the steam during the heat rejection process as the work output varies from 15 kJ to 25 kJ is to be investigated.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
m_Steam = 0.0103 [kg]
THtoTLRatio = 2 "T_H = 2*T_L"
{W_out =15 [kJ]} "Depending on the value of W_out, adjust the guess value of T_L."
eta= 1-1/ THtoTLRatio "eta = 1 - T_L/T_H"
Q_H= W_out/eta
```

"First law applied to the steam engine cycle yields:"

$$Q_H - Q_L = W_{out}$$

"Steady-flow analysis of the condenser yields

$$m_{Steam}h_4 = m_{Steam}h_1 + Q_L$$

$$Q_L = m_{Steam}(h_4 - h_1) \text{ and } h_{fg} = h_4 - h_1 \text{ also } T_L = T_1 = T_4$$

$$Q_L = m_{Steam}h_{fg}$$

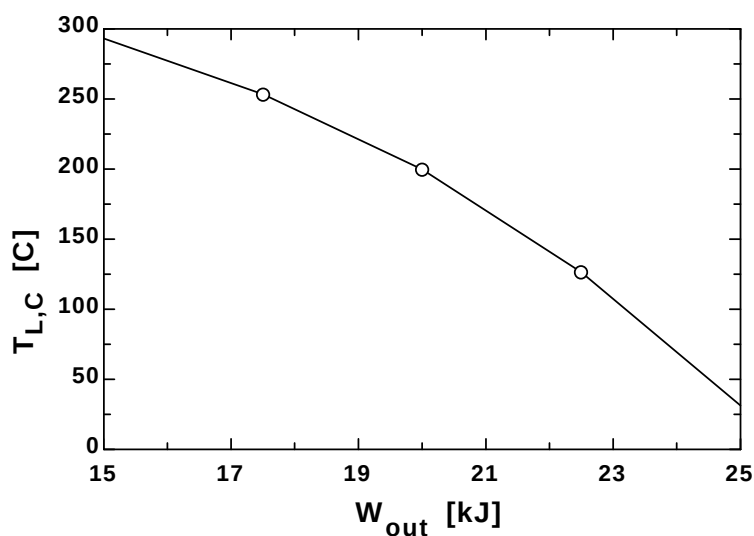
$$h_{fg} = \text{enthalpy}(\text{Steam_iapws}, T=T_L, x=1) - \text{enthalpy}(\text{Steam_iapws}, T=T_L, x=0)$$

$$T_H = \text{THtoTLRatio} * T_L$$

"The heat rejection temperature, in C is:"

$$T_{L,C} = T_L - 273$$

$T_{L,C}$ [C]	W_{out} [kJ]
293.1	15
253.3	17.5
199.6	20
126.4	22.5
31.3	25



7-125 A Carnot refrigeration cycle is executed in a closed system with a fixed mass of R-134a. The net work input and the ratio of maximum-to-minimum temperatures are given. The minimum pressure in the cycle is to be determined.

Assumptions The refrigerator is said to operate on the reversed Carnot cycle, which is totally reversible.

Analysis The coefficient of performance of the cycle is

$$\text{COP}_R = \frac{1}{T_H / T_L - 1} = \frac{1}{1.2 - 1} = 5$$

Also,

$$\text{COP}_R = \frac{Q_L}{W_{\text{in}}} \longrightarrow Q_L = \text{COP}_R \times W_{\text{in}} = (5)(22 \text{ kJ}) = 110 \text{ kJ}$$

$$Q_H = Q_L + W = 110 + 22 = 132 \text{ kJ}$$

and

$$q_H = \frac{Q_H}{m} = \frac{132 \text{ kJ}}{0.96 \text{ kg}} = 137.5 \text{ kJ/kg} = h_{fg@T_H}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that T or P . Therefore, T_H is the temperature that corresponds to the h_{fg} value of 137.5 kJ/kg, and is determined from the R-134a tables to be

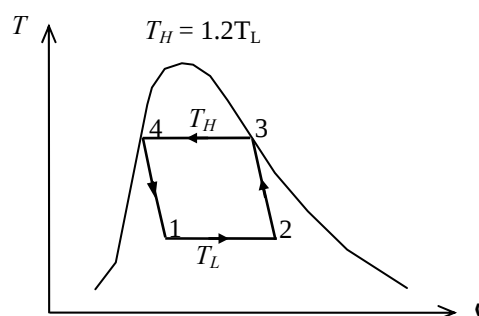
$$T_H \cong 61.3^\circ\text{C} = 334.3 \text{ K}$$

Then,

$$T_L = \frac{T_H}{1.2} = \frac{334.3 \text{ K}}{1.2} = 278.6 \text{ K} \cong 5.6^\circ\text{C}$$

Therefore,

$$P_{\text{min}} = P_{\text{sat}@5.6^\circ\text{C}} = \mathbf{355 \text{ kPa}}$$



7-126 EES Problem 7-125 is reconsidered. The effect of the net work input on the minimum pressure as the work input varies from 10 kJ to 30 kJ is to be investigated. The minimum pressure in the refrigeration cycle is to be plotted as a function of net work input.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

Analysis: The coefficient of performance of the cycle is given by"

$$m_{\text{R134a}} = 0.96 \text{ [kg]}$$

$$T_{\text{HtoTLRatio}} = 1.2 \quad "T_{\text{H}} = 1.2 T_{\text{L}}"$$

$$"W_{\text{in}} = 22 \text{ [kJ]} \quad " \text{Depending on the value of } W_{\text{in}}, \text{ adjust the guess value of } T_{\text{H}}."$$

$$\text{COP}_{\text{R}} = 1 / (T_{\text{HtoTLRatio}} - 1)$$

$$Q_{\text{L}} = W_{\text{in}} * \text{COP}_{\text{R}}$$

"First law applied to the refrigeration cycle yields:"

$$Q_{\text{L}} + W_{\text{in}} = Q_{\text{H}}$$

"Steady-flow analysis of the condenser yields

$$m_{\text{R134a}} * h_3 = m_{\text{R134a}} * h_4 + Q_{\text{H}}$$

$$Q_{\text{H}} = m_{\text{R134a}} * (h_3 - h_4) \quad \text{and} \quad h_{\text{fg}} = h_3 - h_4 \quad \text{also} \quad T_{\text{H}} = T_3 = T_4"$$

$$Q_{\text{H}} = m_{\text{R134a}} * h_{\text{fg}}$$

$$h_{\text{fg}} = \text{enthalpy}(\text{R134a}, T = T_{\text{H}}, x = 1) - \text{enthalpy}(\text{R134a}, T = T_{\text{H}}, x = 0)$$

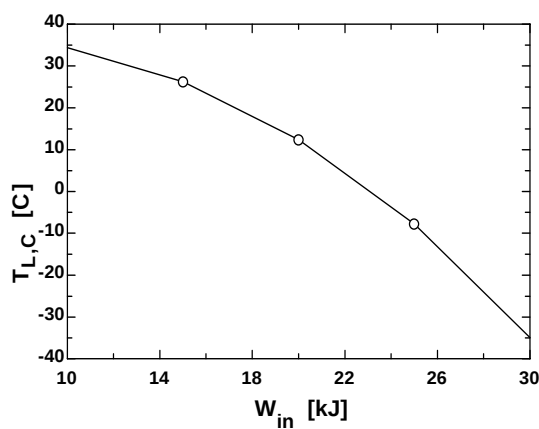
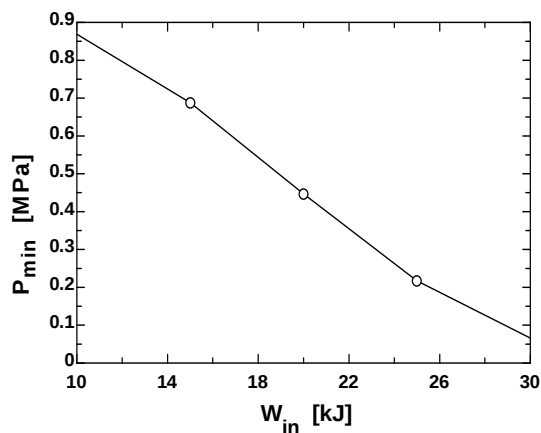
$$T_{\text{H}} = T_{\text{HtoTLRatio}} * T_{\text{L}}$$

"The minimum pressure is the saturation pressure corresponding to T_{L} ."

$$P_{\text{min}} = \text{pressure}(\text{R134a}, T = T_{\text{L}}, x = 0) * \text{convert}(\text{kPa}, \text{MPa})$$

$$T_{\text{L,C}} = T_{\text{L}} - 273$$

P_{min} [MPa]	T_{H} [K]	T_{L} [K]	W_{in} [kJ]	$T_{\text{L,C}}$ [C]
0.8673	368.8	307.3	10	34.32
0.6837	358.9	299	15	26.05
0.45	342.7	285.6	20	12.61
0.2251	319.3	266.1	25	-6.907
0.06978	287.1	239.2	30	-33.78



7-127 Two Carnot heat engines operate in series between specified temperature limits. If the thermal efficiencies of both engines are the same, the temperature of the intermediate medium between the two engines is to be determined.

Assumptions The engines are said to operate on the Carnot cycle, which is totally reversible.

Analysis The thermal efficiency of the two Carnot heat engines can be expressed as

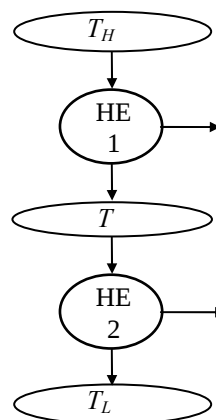
$$\eta_{\text{th,I}} = 1 - \frac{T}{T_H} \quad \text{and} \quad \eta_{\text{th,II}} = 1 - \frac{T_L}{T}$$

Equating,

$$1 - \frac{T}{T_H} = 1 - \frac{T_L}{T}$$

Solving for T ,

$$T = \sqrt{T_H T_L} = \sqrt{(1800 \text{ K})(300 \text{ K})} = 735 \text{ K}$$



7-128 It is to be proven that a refrigerator's COP cannot exceed that of a completely reversible refrigerator that shares the same thermal-energy reservoirs.

Assumptions The refrigerator operates steadily.

Analysis We begin by assuming that the COP of the general refrigerator B is greater than that of the completely reversible refrigerator A , $\text{COP}_B > \text{COP}_A$. When this is the case, a rearrangement of the coefficient of performance expression yields

$$W_B = \frac{Q_L}{\text{COP}_B} < \frac{Q_L}{\text{COP}_A} = W_A$$

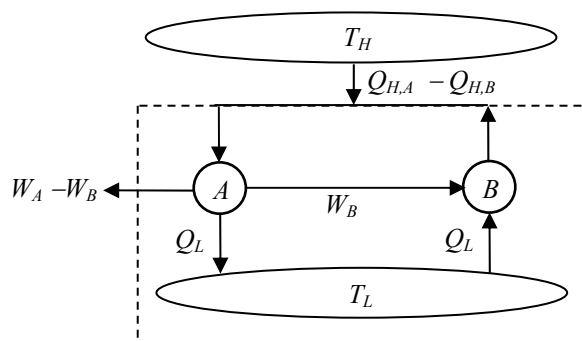
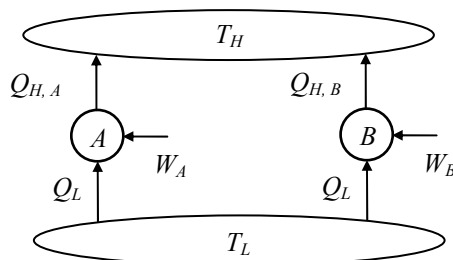
That is, the magnitude of the work required to drive refrigerator B is less than that needed to drive completely reversible refrigerator A . Applying the first law to both refrigerators yields

$$Q_{H,B} < Q_{H,A}$$

since the work supplied to refrigerator B is less than that supplied to refrigerator A , and both have the same cooling effect, Q_L .

Since A is a completely reversible refrigerator, we can reverse it without changing the magnitude of the heat and work transfers. This is illustrated in the figure below. The heat, Q_L , which is rejected by the reversed refrigerator A can now be routed directly to refrigerator B . The net effect when this is done is that no heat is exchanged with the T_L reservoir. The magnitude of the heat supplied to the reversed refrigerator A , $Q_{H,A}$ has been shown to be larger than that rejected by refrigerator B . There is then a net heat transfer from the T_H reservoir to the combined device in the dashed lines of the figure whose magnitude is given by $Q_{H,A} - Q_{H,B}$. Similarly, there is a net work production by the combined device whose magnitude is given by $W_A - W_B$.

The combined cyclic device then exchanges heat with a reservoir at a single temperature and produces work which is clearly a violation of the Kelvin-Planck statement of the second law. Our assumption the $\text{COP}_B > \text{COP}_A$ must then be wrong.



7-129E The thermal efficiency of a completely reversible heat engine as a function of the source temperature is to be calculated and plotted.

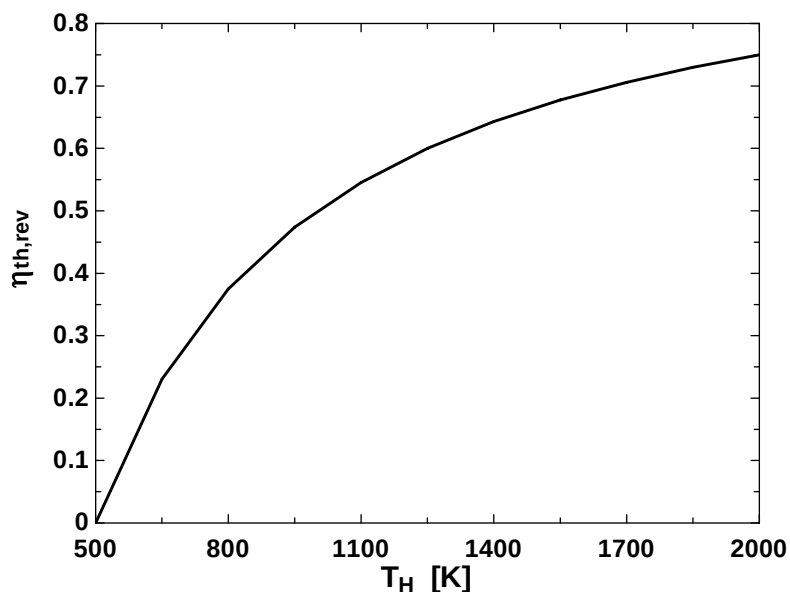
Assumptions The heat engine operates steadily.

Analysis With the specified sink temperature, the thermal efficiency of this completely reversible heat engine is

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} = 1 - \frac{500 \text{ R}}{T_H}$$

Using EES, we tabulate and plot the variation of thermal efficiency with the source temperature:

T_H [R]	$\eta_{\text{th,rev}}$
500	0
650	0.2308
800	0.375
950	0.4737
1100	0.5455
1250	0.6
1400	0.6429
1550	0.6774
1700	0.7059
1850	0.7297
2000	0.75



7-130 It is to be proven that the COP of all completely reversible refrigerators must be the same when the reservoir temperatures are the same.

Assumptions The refrigerators operate steadily.

Analysis We begin by assuming that $\text{COP}_A < \text{COP}_B$. When this is the case, a rearrangement of the coefficient of performance expression yields

$$W_A = \frac{Q_L}{\text{COP}_A} > \frac{Q_L}{\text{COP}_B} = W_B$$

That is, the magnitude of the work required to drive refrigerator *A* is greater than that needed to drive refrigerator *B*. Applying the first law to both refrigerators yields

$$Q_{H,A} > Q_{H,B}$$

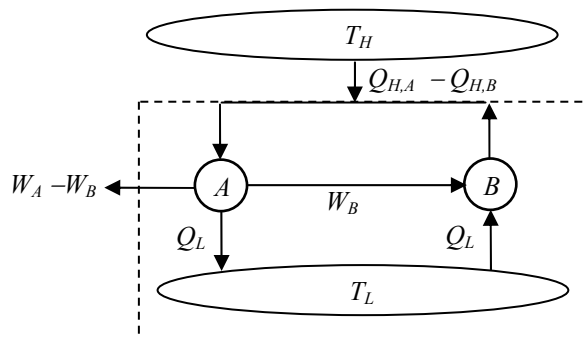
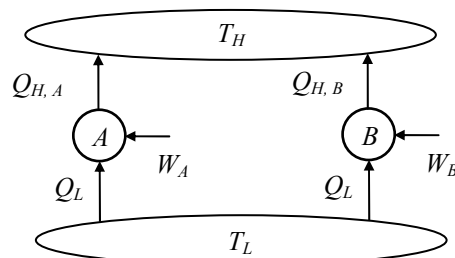
since the work supplied to refrigerator *A* is greater than that supplied to refrigerator *B*, and both have the same cooling effect, Q_L .

Since *A* is a completely reversible refrigerator, we can reverse it without changing the magnitude of the heat and work transfers. This is illustrated in the figure below. The heat, Q_L , which is rejected by the reversed refrigerator *A* can now be routed directly to refrigerator *B*. The net effect when this is done is that no heat is exchanged with the T_L reservoir. The magnitude of the heat supplied to the reversed refrigerator *A*, $Q_{H,A}$ has been shown to be larger than that rejected by refrigerator *B*. There is then a net heat transfer from the T_H reservoir to the combined device in the dashed lines of the figure whose magnitude is given by $Q_{H,A} - Q_{H,B}$. Similarly, there is a net work production by the combined device whose magnitude is given by $W_A - W_B$.

The combined cyclic device then exchanges heat with a reservoir at a single temperature and produces work which is clearly a violation of the Kelvin-Planck statement of the second law. Our assumption the $\text{COP}_A < \text{COP}_B$ must then be wrong.

If we interchange *A* and *B* in the previous argument, we would conclude that the COP_B cannot be less than COP_A . The only alternative left is that

$$\text{COP}_A = \text{COP}_B$$



7-131 An expression for the COP of a completely reversible heat pump in terms of the thermal-energy reservoir temperatures, T_L and T_H is to be derived.

Assumptions The heat pump operates steadily.

Analysis Application of the first law to the completely reversible heat pump yields

$$W_{\text{net,in}} = Q_H - Q_L$$

This result may be used to reduce the coefficient of performance,

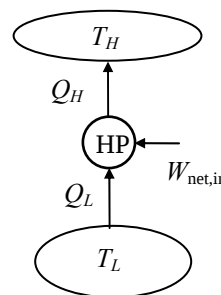
$$\text{COP}_{\text{HP,rev}} = \frac{Q_H}{W_{\text{net,in}}} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - Q_L / Q_H}$$

Since this heat pump is completely reversible, the thermodynamic definition of temperature tells us that,

$$\frac{Q_L}{Q_H} = \frac{T_L}{T_H}$$

When this is substituted into the COP expression, the result is

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{T_H}{T_H - T_L}$$



7-132 A Carnot heat engine drives a Carnot refrigerator that removes heat from a cold medium at a specified rate. The rate of heat supply to the heat engine and the total rate of heat rejection to the environment are to be determined.

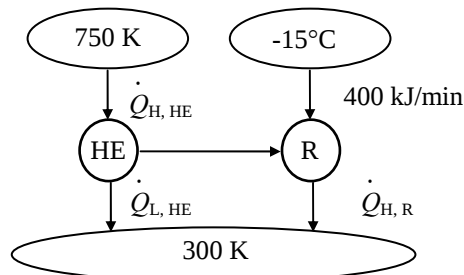
Analysis (a) The coefficient of performance of the Carnot refrigerator is

$$\text{COP}_{R,C} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(300 \text{ K})/(258 \text{ K}) - 1} = 6.14$$

Then power input to the refrigerator becomes

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{R,C}} = \frac{400 \text{ kJ/min}}{6.14} = 65.1 \text{ kJ/min}$$

which is equal to the power output of the heat engine, $\dot{W}_{\text{net,out}}$.



The thermal efficiency of the Carnot heat engine is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{750 \text{ K}} = 0.60$$

Then the rate of heat input to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,HE} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th,HE}}} = \frac{65.1 \text{ kJ/min}}{0.60} = \mathbf{108.5 \text{ kJ/min}}$$

(b) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,HE}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,R}$),

$$\dot{Q}_{L,HE} = \dot{Q}_{H,HE} - \dot{W}_{\text{net,out}} = 108.5 - 65.1 = 43.4 \text{ kJ/min}$$

$$\dot{Q}_{H,R} = \dot{Q}_{L,R} + \dot{W}_{\text{net,in}} = 400 + 65.1 = 465.1 \text{ kJ/min}$$

and

$$\dot{Q}_{\text{Ambient}} = \dot{Q}_{L,HE} + \dot{Q}_{H,R} = 43.4 + 465.1 = \mathbf{508.5 \text{ kJ/min}}$$

7-133 EES Problem 7-132 is reconsidered. The effects of the heat engine source temperature, the environment temperature, and the cooled space temperature on the required heat supply to the heat engine and the total rate of heat rejection to the environment as the source temperature varies from 500 K to 1000 K, the environment temperature varies from 275 K to 325 K, and the cooled space temperature varies from -20°C to 0°C are to be investigated. The required heat supply is to be plotted against the source temperature for the cooled space temperature of -15°C and environment temperatures of 275, 300, and 325 K.

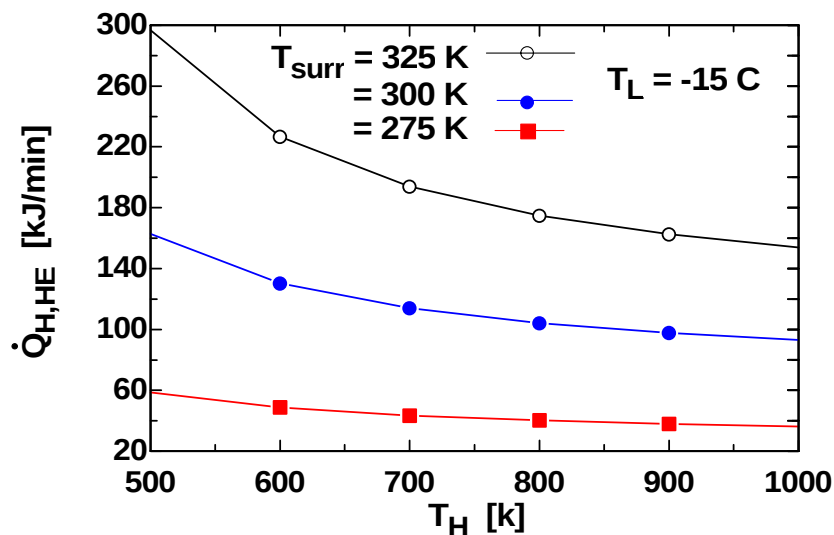
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

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Q_dot_L_R = 400 [kJ/min]
T_surr = 300 [K]
T_H = 750 [K]
T_L_C = -15 [C]
T_L = T_L_C + 273 "[K]"
"Coefficient of performance of the Carnot refrigerator:"
T_H_R = T_surr
COP_R = 1/(T_H_R/T_L - 1)
"Power input to the refrigerator:"
W_dot_in_R = Q_dot_L_R/COP_R
"Power output from heat engine must be:"
W_dot_out_HE = W_dot_in_R
"The efficiency of the heat engine is:"
T_L_HE = T_surr
eta_HE = 1 - T_L_HE/T_H
"The rate of heat input to the heat engine is:"
Q_dot_H_HE = W_dot_out_HE/eta_HE
"First law applied to the heat engine and refrigerator:"
Q_dot_L_HE = Q_dot_H_HE - W_dot_out_HE
Q_dot_H_R = Q_dot_L_R + W_dot_in_R
"Total heat transfer rate to the surroundings:"
Q_dot_surr = Q_dot_L_HE + Q_dot_H_R "[kJ/min]"

```

$\dot{Q}_{HHE}$ [kJ/min]	T_H [K]
162.8	500
130.2	600
114	700
104.2	800
97.67	900
93.02	1000



7-134 Half of the work output of a Carnot heat engine is used to drive a Carnot heat pump that is heating a house. The minimum rate of heat supply to the heat engine is to be determined.

Assumptions Steady operating conditions exist.

Analysis The coefficient of performance of the Carnot heat pump is

$$\text{COP}_{\text{HP,C}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (2 + 273 \text{ K}) / (22 + 273 \text{ K})} = 14.75$$

Then power input to the heat pump, which is supplying heat to the house at the same rate as the rate of heat loss, becomes

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,C}}} = \frac{62,000 \text{ kJ/h}}{14.75} = 4203 \text{ kJ/h}$$

which is half the power produced by the heat engine. Thus the power output of the heat engine is

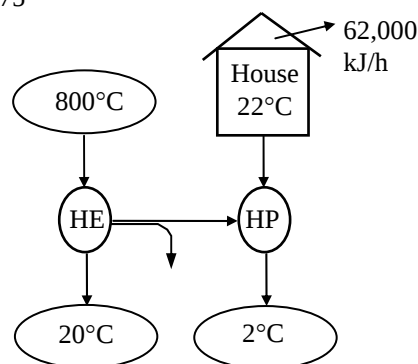
$$\dot{W}_{\text{net,out}} = 2\dot{W}_{\text{net,in}} = 2(4203 \text{ kJ/h}) = 8406 \text{ kJ/h}$$

To minimize the rate of heat supply, we must use a Carnot heat engine whose thermal efficiency is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{293 \text{ K}}{1073 \text{ K}} = 0.727$$

Then the rate of heat supply to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\text{HE}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th,HE}}} = \frac{8406 \text{ kJ/h}}{0.727} = \mathbf{11,560 \text{ kJ/h}}$$



7-135 A Carnot refrigeration cycle is executed in a closed system with a fixed mass of R-134a. The net work input and the maximum and minimum temperatures are given. The mass fraction of the refrigerant that vaporizes during the heat addition process, and the pressure at the end of the heat rejection process are to be determined.

Properties The enthalpy of vaporization of R-134a at -8°C is $h_{fg} = 204.52 \text{ kJ/kg}$ (Table A-12).

Analysis The coefficient of performance of the cycle is

$$\text{COP}_R = \frac{1}{T_H / T_L - 1} = \frac{1}{293 / 265 - 1} = 9.464$$

and

$$Q_L = \text{COP}_R \times W_{\text{in}} = (9.464)(15 \text{ kJ}) = 142 \text{ kJ}$$

Then the amount of refrigerant that vaporizes during heat absorption is

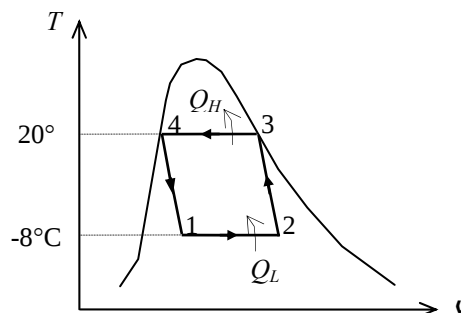
$$Q_L = m h_{fg@T_L=-8^{\circ}\text{C}} \longrightarrow m = \frac{142 \text{ kJ}}{204.52 \text{ kJ/kg}} = 0.695 \text{ kg}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that T or P . Therefore, the fraction of mass that vaporized during heat addition process is

$$\frac{0.695 \text{ kg}}{0.8 \text{ kg}} = 0.868 \text{ or } \mathbf{86.8\%}$$

The pressure at the end of the heat rejection process is

$$P_4 = P_{\text{sat}@20^{\circ}\text{C}} = \mathbf{572.1 \text{ kPa}}$$



7-136 A Carnot heat pump cycle is executed in a steady-flow system with R-134a flowing at a specified rate. The net power input and the ratio of the maximum-to-minimum temperatures are given. The ratio of the maximum to minimum pressures is to be determined.

Analysis The coefficient of performance of the cycle is

$$\text{COP}_{\text{HP}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - 1/1.25} = 5.0$$

and

$$\dot{Q}_H = \text{COP}_{\text{HP}} \times \dot{W}_{\text{in}} = (5.0)(7 \text{ kW}) = 35.0 \text{ kJ/s}$$

$$q_H = \frac{\dot{Q}_H}{\dot{m}} = \frac{35.0 \text{ kJ/s}}{0.264 \text{ kg/s}} = 132.58 \text{ kJ/kg} = h_{fg@T_H}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that T or P . Therefore, T_H is the temperature that corresponds to the h_{fg} value of 132.58 kJ/kg, and is determined from the R-134a tables to be

$$T_H \cong 64.6^\circ\text{C} = 337.8 \text{ K}$$

and

$$P_{\text{max}} = P_{\text{sat}@64.6^\circ\text{C}} = 1875 \text{ kPa}$$

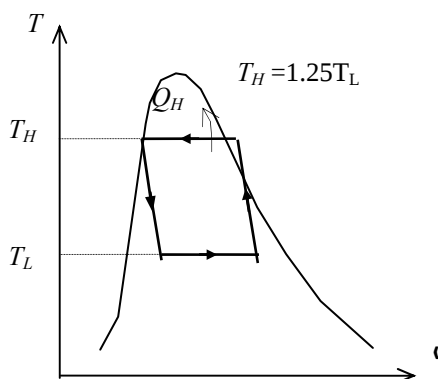
$$T_L = \frac{T_H}{1.25} = \frac{337.8 \text{ K}}{1.25} = 293.7 \text{ K} \cong 20.6^\circ\text{C}$$

Also,

$$P_{\text{min}} = P_{\text{sat}@20.6^\circ\text{C}} = 582 \text{ kPa}$$

Then the ratio of the maximum to minimum pressures in the cycle is

$$\frac{P_{\text{max}}}{P_{\text{min}}} = \frac{1875 \text{ kPa}}{582 \text{ kPa}} = 3.22$$



7-137 A Carnot heat engine is operating between specified temperature limits. The source temperature that will double the efficiency is to be determined.

Analysis Denoting the new source temperature by T_H^* , the thermal efficiency of the Carnot heat engine for both cases can be expressed as

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} \quad \text{and} \quad \eta_{\text{th,C}}^* = 1 - \frac{T_L}{T_H^*} = 2\eta_{\text{th,C}}$$

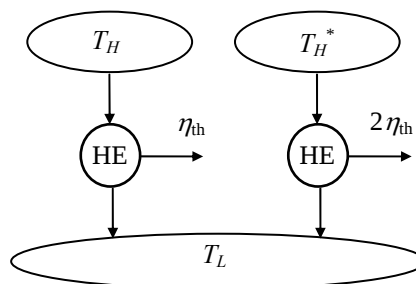
Substituting,

$$1 - \frac{T_L}{T_H^*} = 2 \left(1 - \frac{T_L}{T_H} \right)$$

Solving for T_H^* ,

$$T_H^* = \frac{T_H T_L}{T_H - 2T_L}$$

which is the desired relation.



7-138E A washing machine uses \$33/year worth of hot water heated by a gas water heater. The amount of hot water an average family uses per week is to be determined.

Assumptions 1 The electricity consumed by the motor of the washer is negligible. 2 Water is an incompressible substance with constant properties at room temperature.

Properties The density and specific heat of water at room temperature are $\rho = 62.1 \text{ lbm/ft}^3$ and $c = 1.00 \text{ Btu/lbm} \cdot ^\circ\text{F}$ (Table A-3E).

Analysis The amount of electricity used to heat the water and the net amount transferred to water are

$$\text{Total energy used (gas)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$33/\text{year}}{\$1.21/\text{therm}} = 27.27 \text{ therms/year}$$

$$\begin{aligned} \text{Total energy transfer to water} &= \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.58 \times 27.27 \text{ therms/year} \\ &= 15.82 \text{ therms/year} = (15.82 \text{ therms/year}) \left(\frac{100,000 \text{ Btu}}{1 \text{ therm}} \right) \left(\frac{1 \text{ year}}{52 \text{ weeks}} \right) \\ &= 30,420 \text{ Btu/week} \end{aligned}$$

Then the mass and the volume of hot water used per week become

$$\dot{E}_{\text{in}} = \dot{m}c(T_{\text{out}} - T_{\text{in}}) \rightarrow \dot{m} = \frac{\dot{E}_{\text{in}}}{c(T_{\text{out}} - T_{\text{in}})} = \frac{30,420 \text{ Btu/week}}{(1.0 \text{ Btu/lbm} \cdot ^\circ\text{F})(130 - 60)^\circ\text{F}} = 434.6 \text{ lbm/week}$$

and

$$\dot{V}_{\text{water}} = \frac{\dot{m}}{\rho} = \frac{434.6 \text{ lbm/week}}{62.1 \text{ lbm/ft}^3} = (7.0 \text{ ft}^3/\text{week}) \left(\frac{7.4804 \text{ gal}}{1 \text{ ft}^3} \right) = 52.4 \text{ gal/week}$$

Therefore, an average family uses about 52 gallons of hot water per week for washing clothes.

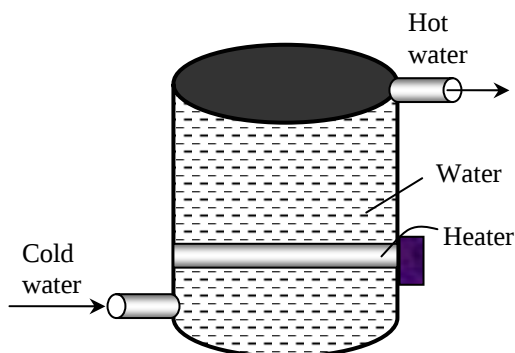
7-139 CD EES A typical heat pump powered water heater costs about \$800 more to install than a typical electric water heater. The number of years it will take for the heat pump water heater to pay for its cost differential from the energy it saves is to be determined.

Assumptions **1** The price of electricity remains constant. **2** Water is an incompressible substance with constant properties at room temperature. **3** Time value of money (interest, inflation) is not considered.

Analysis The amount of electricity used to heat the water and the net amount transferred to water are

$$\begin{aligned}\text{Total energy used (electrical)} &= \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} \\ &= \frac{\$390/\text{year}}{\$0.080/\text{kWh}} \\ &= 4875 \text{ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Total energy transfer to water} &= \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.9 \times 4875 \text{ kWh/year} \\ &= 4388 \text{ kWh/year}\end{aligned}$$



The amount of electricity consumed by the heat pump and its cost are

$$\text{Energy usage (of heat pump)} = \frac{\text{Energy transfer to water}}{\text{COP}_{\text{HP}}} = \frac{4388 \text{ kWh/year}}{2.2} = 1995 \text{ kWh/year}$$

$$\begin{aligned}\text{Energy cost (of heat pump)} &= (\text{Energy usage})(\text{Unit cost of energy}) = (1995 \text{ kWh/year})(\$0.08/\text{kWh}) \\ &= \$159.6/\text{year}\end{aligned}$$

Then the money saved per year by the heat pump and the simple payback period become

$$\begin{aligned}\text{Money saved} &= (\text{Energy cost of electric heater}) - (\text{Energy cost of heat pump}) \\ &= \$390 - \$159.60 = \$230.40\end{aligned}$$

$$\text{Simple payback period} = \frac{\text{Additional installation cost}}{\text{Money saved}} = \frac{\$800}{\$230.40/\text{year}} = 3.5 \text{ years}$$

Discussion The economics of heat pump water heater will be even better if the air in the house is used as the heat source for the heat pump in summer, and thus also serving as an air-conditioner.

7-140 EES Problem 7-139 is reconsidered. The effect of the heat pump COP on the yearly operation costs and the number of years required to break even are to be considered.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Energy supplied by the water heater to the water per year is E_ElecHeater"

"Cost per year to operate electric water heater for one year is:"

Cost_ElecHeater = 390 [\$/year]

"Energy supplied to the water by electric heater is 90% of energy purchased"

E_ElecHeater = 0.9*Cost_ElecHeater /UnitCost "[kWh/year]"

UnitCost=0.08 [\$/kWh]

"For the same amount of heated water and assuming that all the heat energy leaving the heat pump goes into the water, then"

"Energy supplied by heat pump heater = Energy supplied by electric heater"

E_HeatPump = E_ElecHeater "[kWh/year]"

"Electrical Work energy supplied to heat pump = Heat added to water/COP"

COP=2.2

W_HeatPump = E_HeatPump/COP "[kWh/year]"

"Cost per year to operate the heat pump is"

Cost_HeatPump=W_HeatPump*UnitCost

"Let N_BrkEven be the number of years to break even:"

"At the break even point, the total cost difference between the two water heaters is zero."

"Years to break even, neglecting the cost to borrow the extra \$800 to install heat pump"

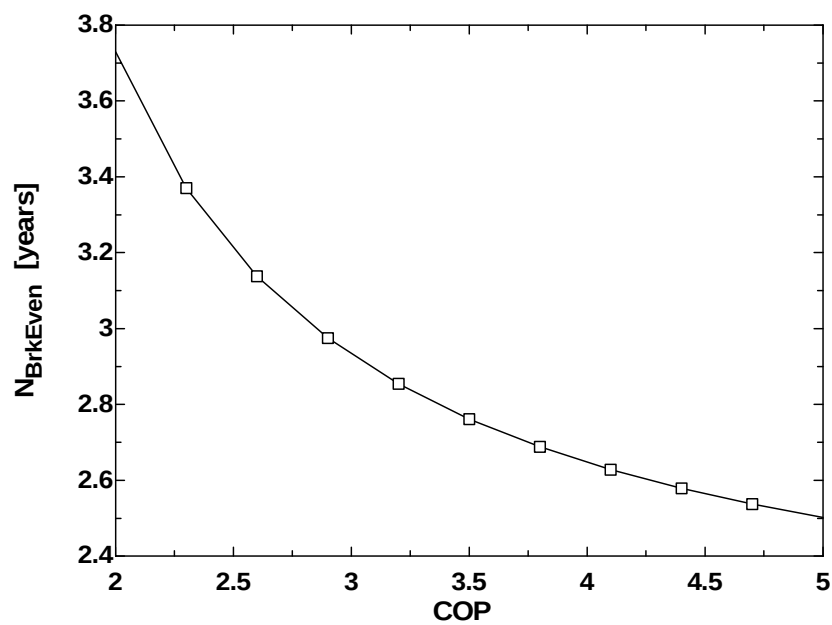
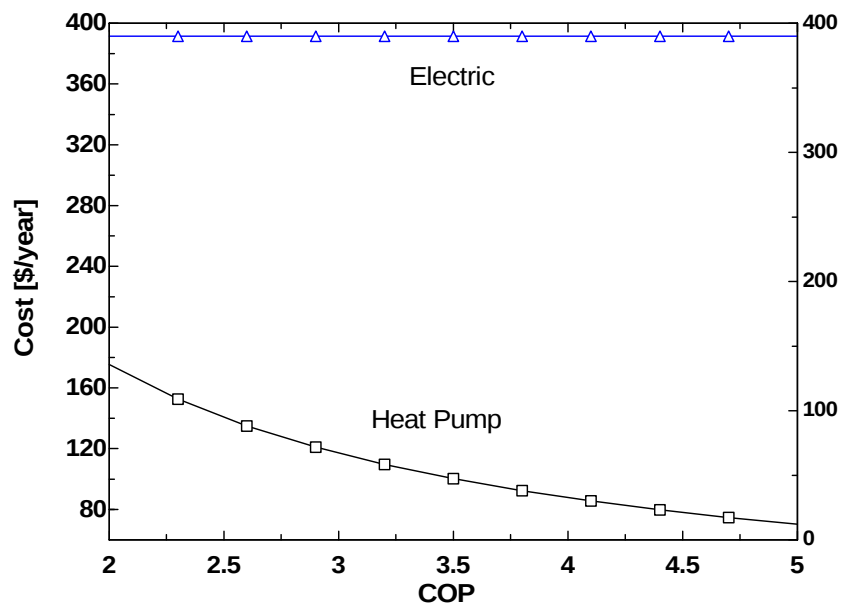
CostDiff_total = 0 [\$]

CostDiff_total=AddCost+N_BrkEven*(Cost_HeatPump-Cost_ElecHeater)

AddCost=800 [\$]

"The plot windows show the effect of heat pump COP on the yearly operation costs and the number of years required to break even. The data for these plots were obtained by placing '{' and '}' around the COP = 2.2 line, setting the COP values in the Parametric Table, and pressing F3 or selecting Solve Table from the Calculate menu"

COP	B _{BrkEven} [years]	Cost _{HeatPump} [\$/year]	Cost _{ElektHeater} [\$/year]
2	3.73	175.5	390
2.3	3.37	152.6	390
2.6	3.137	135	390
2.9	2.974	121	390
3.2	2.854	109.7	390
3.5	2.761	100.3	390
3.8	2.688	92.37	390
4.1	2.628	85.61	390
4.4	2.579	79.77	390
4.7	2.537	74.68	390
5	2.502	70.2	390



7-141 A home owner is to choose between a high-efficiency natural gas furnace and a ground-source heat pump. The system with the lower energy cost is to be determined.

Assumptions The two heater are comparable in all aspects other than the cost of energy.

Analysis The unit cost of each kJ of useful energy supplied to the house by each system is

$$\text{Natural gas furnace:} \quad \text{Unit cost of useful energy} = \frac{(\$1.42/\text{therm})}{0.97} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = \$13.8 \times 10^{-6} / \text{kJ}$$

$$\text{Heat Pump System:} \quad \text{Unit cost of useful energy} = \frac{(\$0.092/\text{kWh})}{3.5} \left(\frac{1 \text{ kWh}}{3600 \text{ kJ}} \right) = \$7.3 \times 10^{-6} / \text{kJ}$$

The energy cost of **ground-source heat pump system** will be lower.

7-142 ... 7-147 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 8
ENTROPY

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Entropy and the Increase of Entropy Principle

8-1C No. The $\oint \delta Q$ represents the net heat transfer during a cycle, which could be positive.

8-2C No. A system may produce more (or less) work than it receives during a cycle. A steam power plant, for example, produces more work than it receives during a cycle, the difference being the net work output.

8-3C The entropy change will be the same for both cases since entropy is a property and it has a fixed value at a fixed state.

8-4C No. In general, that integral will have a different value for different processes. However, it will have the same value for all reversible processes.

8-5C Yes.

8-6C That integral should be performed along a reversible path to determine the entropy change.

8-7C No. An isothermal process can be irreversible. Example: A system that involves paddle-wheel work while losing an equivalent amount of heat.

8-8C The value of this integral is always larger for reversible processes.

8-9C No. Because the entropy of the surrounding air increases even more during that process, making the total entropy change positive.

8-10C It is possible to create entropy, but it is not possible to destroy it.

8-11C If the system undergoes a reversible process, the entropy of the system cannot change without a heat transfer. Otherwise, the entropy must increase since there are no offsetting entropy changes associated with reservoirs exchanging heat with the system.

8-12C The claim that work will not change the entropy of a fluid passing through an adiabatic steady-flow system with a single inlet and outlet is true only if the process is also reversible. Since no real process is reversible, there will be an entropy increase in the fluid during the adiabatic process in devices such as pumps, compressors, and turbines.

8-13C Sometimes.

8-14C Never.

8-15C Always.

8-16C Increase.

8-17C Increases.

8-18C Decreases.

8-19C Sometimes.

8-20C Yes. This will happen when the system is losing heat, and the decrease in entropy as a result of this heat loss is equal to the increase in entropy as a result of irreversibilities.

8-21C They are heat transfer, irreversibilities, and entropy transport with mass.

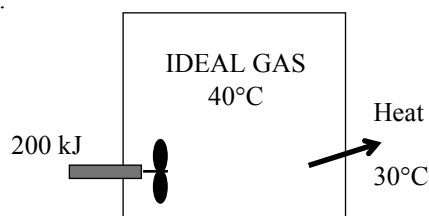
8-22C Greater than.

8-23 A rigid tank contains an ideal gas that is being stirred by a paddle wheel. The temperature of the gas remains constant as a result of heat transfer out. The entropy change of the gas is to be determined.

Assumptions The gas in the tank is given to be an ideal gas.

Analysis The temperature and the specific volume of the gas remain constant during this process. Therefore, the initial and the final states of the gas are the same. Then $s_2 = s_1$ since entropy is a property. Therefore,

$$\Delta S_{\text{sys}} = 0$$



8-24 Air is compressed steadily by a compressor. The air temperature is maintained constant by heat rejection to the surroundings. The rate of entropy change of air is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 Air is an ideal gas. 4 The process involves no internal irreversibilities such as friction, and thus it is an isothermal, internally reversible process.

Properties Noting that $h = h(T)$ for ideal gases, we have $h_1 = h_2$ since $T_1 = T_2 = 25^\circ\text{C}$.

Analysis We take the compressor as the system. Noting that the enthalpy of air remains constant, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

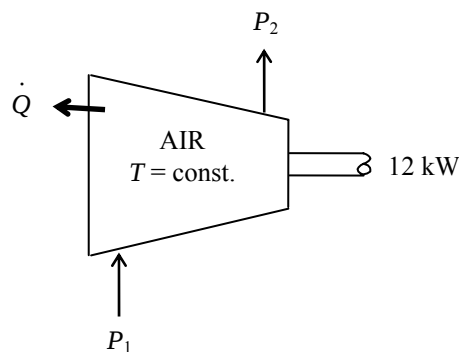
$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}}$$

Therefore,

$$\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 12 \text{ kW}$$

Noting that the process is assumed to be an isothermal and internally reversible process, the rate of entropy change of air is determined to be

$$\dot{\Delta S}_{\text{air}} = -\frac{\dot{Q}_{\text{out,air}}}{T_{\text{sys}}} = -\frac{12 \text{ kW}}{298 \text{ K}} = \mathbf{-0.0403 \text{ kW/K}}$$



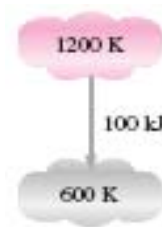
8-25 Heat is transferred directly from an energy-source reservoir to an energy-sink. The entropy change of the two reservoirs is to be calculated and it is to be determined if the increase of entropy principle is satisfied.

Assumptions The reservoirs operate steadily.

Analysis The entropy change of the source and sink is given by

$$\Delta S = \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-100 \text{ kJ}}{1200 \text{ K}} + \frac{100 \text{ kJ}}{600 \text{ K}} = \mathbf{0.0833 \text{ kJ/K}}$$

Since the entropy of everything involved in this process has increased, this transfer of heat is **possible**.



8-26 It is assumed that heat is transferred from a cold reservoir to the hot reservoir contrary to the Clausius statement of the second law. It is to be proven that this violates the increase in entropy principle.

Assumptions The reservoirs operate steadily.

Analysis According to the definition of the entropy, the entropy change of the high-temperature reservoir shown below is

$$\Delta S_H = \frac{Q}{T_H} = \frac{100 \text{ kJ}}{1200 \text{ K}} = 0.08333 \text{ kJ/K}$$

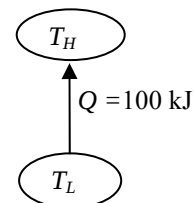
and the entropy change of the low-temperature reservoir is

$$\Delta S_L = \frac{Q}{T_L} = \frac{-100 \text{ kJ}}{600 \text{ K}} = -0.1667 \text{ kJ/K}$$

The total entropy change of everything involved with this system is then

$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L = 0.08333 - 0.1667 = -0.0833 \text{ kJ/K}$$

which violates the increase in entropy principle since the entropy is decreasing, not increasing or staying fixed.



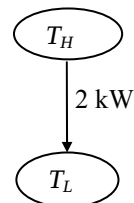
8-27 Heat is transferred from a hot reservoir to a cold reservoir. The entropy change of the two reservoirs is to be calculated and it is to be determined if the second law is satisfied.

Assumptions The reservoirs operate steadily.

Analysis The rate of entropy change of everything involved in this transfer of heat is given by

$$\Delta \dot{S} = \frac{\dot{Q}_H}{T_H} + \frac{\dot{Q}_L}{T_L} = \frac{-2 \text{ kW}}{800 \text{ K}} + \frac{2 \text{ kW}}{300 \text{ K}} = 0.00417 \text{ kW/K}$$

Since this rate is positive (i.e., the entropy increases as time passes), this transfer of heat is **possible**.



8-28E A reversible air conditioner with specified reservoir temperatures is considered. The entropy change of two reservoirs is to be calculated and it is to be determined if this air conditioner satisfies the increase in entropy principle.

Assumptions The air conditioner operates steadily.

Analysis According to the thermodynamic temperature scale,

$$\dot{Q}_H = \dot{Q}_L \frac{T_H}{T_L} = (36,000 \text{ Btu/h}) \frac{570 \text{ R}}{530 \text{ R}} = 38,720 \text{ Btu/h}$$

The rate of entropy change of the hot reservoir is then

$$\Delta \dot{S}_H = \frac{\dot{Q}_H}{T_H} = \frac{-38,720 \text{ Btu/h}}{570 \text{ R}} = -67.92 \text{ Btu/h} \cdot \text{R}$$

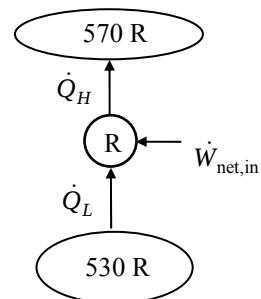
Similarly, the rate of entropy change of the cold reservoir is

$$\Delta \dot{S}_L = \frac{\dot{Q}_L}{T_L} = \frac{36,000 \text{ Btu/h}}{530 \text{ R}} = 67.92 \text{ Btu/h} \cdot \text{R}$$

The net rate of entropy change of everything in this system is

$$\Delta \dot{S}_{\text{total}} = \Delta \dot{S}_H + \Delta \dot{S}_L = -67.92 + 67.92 = \mathbf{0 \text{ Btu/h} \cdot \text{R}}$$

The net rate of entropy change is zero as it must be in order to satisfy the second law.



8-29 A reversible heat pump with specified reservoir temperatures is considered. The entropy change of two reservoirs is to be calculated and it is to be determined if this heat pump satisfies the increase in entropy principle.

Assumptions The heat pump operates steadily.

Analysis Since the heat pump is completely reversible, the combination of the coefficient of performance expression, first Law, and thermodynamic temperature scale gives

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (283 \text{ K}) / (294 \text{ K})} = 26.73$$

The power required to drive this heat pump, according to the coefficient of performance, is then

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,rev}}} = \frac{100 \text{ kW}}{26.73} = 3.741 \text{ kW}$$

According to the first law, the rate at which heat is removed from the low-temperature energy reservoir is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 100 \text{ kW} - 3.741 \text{ kW} = 96.26 \text{ kW}$$

The rate at which the entropy of the high temperature reservoir changes, according to the definition of the entropy, is

$$\Delta \dot{S}_H = \frac{\dot{Q}_H}{T_H} = \frac{100 \text{ kW}}{294 \text{ K}} = \mathbf{0.340 \text{ kW/K}}$$

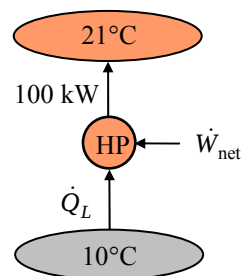
and that of the low-temperature reservoir is

$$\Delta \dot{S}_L = \frac{\dot{Q}_L}{T_L} = \frac{-96.26 \text{ kW}}{283 \text{ K}} = \mathbf{-0.340 \text{ kW/K}}$$

The net rate of entropy change of everything in this system is

$$\Delta \dot{S}_{\text{total}} = \Delta \dot{S}_H + \Delta \dot{S}_L = 0.340 - 0.340 = \mathbf{0 \text{ kW/K}}$$

as it must be since the heat pump is completely reversible.



8-30E Heat is transferred isothermally from the working fluid of a Carnot engine to a heat sink. The entropy change of the working fluid is given. The amount of heat transfer, the entropy change of the sink, and the total entropy change during the process are to be determined.

Analysis (a) This is a reversible isothermal process, and the entropy change during such a process is given by

$$\Delta S = \frac{Q}{T}$$

Noting that heat transferred from the working fluid is equal to the heat transferred to the sink, the heat transfer become

$$Q_{\text{fluid}} = T_{\text{fluid}} \Delta S_{\text{fluid}} = (555 \text{ R})(-0.7 \text{ Btu/R}) = -388.5 \text{ Btu} \rightarrow Q_{\text{fluid,out}} = \mathbf{388.5 \text{ Btu}}$$

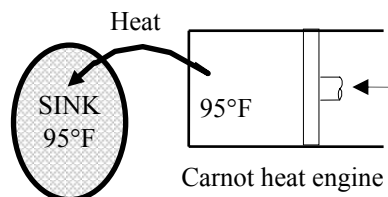
(b) The entropy change of the sink is determined from

$$\Delta S_{\text{sink}} = \frac{Q_{\text{sink,in}}}{T_{\text{sink}}} = \frac{388.5 \text{ Btu}}{555 \text{ R}} = \mathbf{0.7 \text{ Btu/R}}$$

(c) Thus the total entropy change of the process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{fluid}} + \Delta S_{\text{sink}} = -0.7 + 0.7 = \mathbf{0}$$

This is expected since all processes of the Carnot cycle are reversible processes, and no entropy is generated during a reversible process.



8-31 R-134a enters an evaporator as a saturated liquid-vapor at a specified pressure. Heat is transferred to the refrigerant from the cooled space, and the liquid is vaporized. The entropy change of the refrigerant, the entropy change of the cooled space, and the total entropy change for this process are to be determined.

Assumptions 1 Both the refrigerant and the cooled space involve no internal irreversibilities such as friction. 2 Any temperature change occurs within the wall of the tube, and thus both the refrigerant and the cooled space remain isothermal during this process. Thus it is an isothermal, internally reversible process.

Analysis Noting that both the refrigerant and the cooled space undergo reversible isothermal processes, the entropy change for them can be determined from

$$\Delta S = \frac{Q}{T}$$

(a) The pressure of the refrigerant is maintained constant. Therefore, the temperature of the refrigerant also remains constant at the saturation value,

$$T = T_{\text{sat}@160 \text{ kPa}} = -15.6^\circ\text{C} = 257.4 \text{ K} \quad (\text{Table A-12})$$

Then,

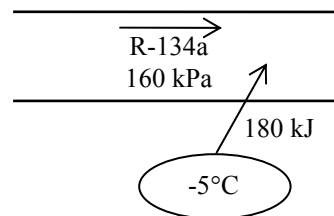
$$\Delta S_{\text{refrigerant}} = \frac{Q_{\text{refrigerant, in}}}{T_{\text{refrigerant}}} = \frac{180 \text{ kJ}}{257.4 \text{ K}} = \mathbf{0.699 \text{ kJ/K}}$$

(b) Similarly,

$$\Delta S_{\text{space}} = -\frac{Q_{\text{space, out}}}{T_{\text{space}}} = -\frac{180 \text{ kJ}}{268 \text{ K}} = \mathbf{-0.672 \text{ kJ/K}}$$

(c) The total entropy change of the process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{refrigerant}} + \Delta S_{\text{space}} = 0.699 - 0.672 = \mathbf{0.027 \text{ kJ/K}}$$



Entropy Changes of Pure Substances

8-32C Yes, because an internally reversible, adiabatic process involves no irreversibilities or heat transfer.

8-33C According to the conservation of mass principle,

$$\begin{aligned}\frac{dm_{CV}}{dt} &= \sum_{in} \dot{m} - \sum_{out} \dot{m} \\ \frac{dm}{dt} &= \dot{m}_{in}\end{aligned}$$

An entropy balance adapted to this system becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} - \dot{m}_{in} s_{in} \geq 0$$

When this is combined with the mass balance, and the constant entropies are removed from the derivatives, it becomes

$$\frac{dS_{surr}}{dt} + s \frac{dm}{dt} - s_{in} \frac{dm}{dt} \geq 0$$

Multiplying by dt and integrating the result yields

$$\Delta S_{surr} + (s - s_{in}) \Delta m \geq 0$$

or

$$\Delta S_{surr} \geq (s - s_{in}) \Delta m$$

8-34C According to the conservation of mass principle,

$$\begin{aligned}\frac{dm_{CV}}{dt} &= \sum_{in} \dot{m} - \sum_{out} \dot{m} \\ \frac{dm}{dt} &= -\dot{m}_{out}\end{aligned}$$

An entropy balance adapted to this system becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} + \dot{m}_{out} s \geq 0$$

When this is combined with the mass balance, it becomes

$$\frac{dS_{surr}}{dt} + s \frac{dm}{dt} - s \frac{dm}{dt} \geq 0$$

Multiplying by dt and integrating the result yields

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s(m_2 - m_1) \geq 0$$

Since all the entropies are same, this reduces to

$$\Delta S_{surr} \geq 0$$

Hence, the entropy of the surroundings can only increase or remain fixed.

8-35 R-134a is expanded in a turbine during which the entropy remains constant. The enthalpy difference is to be determined.

Analysis The initial state is superheated vapor and thus

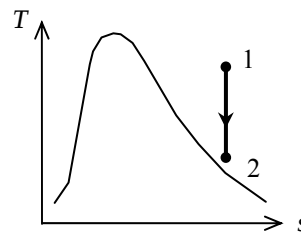
$$\left. \begin{array}{l} P_1 = 250 \text{ psia} \\ T_1 = 175^\circ\text{F} \end{array} \right\} \begin{array}{l} h_1 = 129.95 \text{ Btu/lbm} \\ s_1 = 0.23281 \text{ Btu/lbm} \cdot \text{R} \end{array} \quad (\text{from EES})$$

The entropy is constant during the process. The final state is also superheated vapor and the enthalpy at this state is

$$\left. \begin{array}{l} T_2 = 20^\circ\text{F} \\ s_2 = s_1 = 0.23281 \text{ Btu/lbm} \cdot \text{R} \end{array} \right\} h_2 = 106.95 \text{ Btu/lbm} \quad (\text{from EES})$$

Note that the properties at the inlet and exit states can also be determined from Table A-13E by interpolation but the values will not be as accurate as those by EES. The change in the enthalpy across the turbine is then

$$\Delta h = h_2 - h_1 = 106.95 - 129.95 = -23.0 \text{ Btu/lbm}$$



8-36E A piston-cylinder device that is filled with water is heated. The total entropy change is to be determined.

Analysis The initial specific volume is

$$\nu_1 = \frac{\nu_1}{m} = \frac{2.5 \text{ ft}^3}{2 \text{ lbm}} = 1.25 \text{ ft}^3/\text{lbm}$$

which is between ν_f and ν_g for 300 psia. The initial quality and the entropy are then (Table A-5E)

$$x_1 = \frac{\nu_1 - \nu_f}{\nu_{fg}} = \frac{(1.25 - 0.01890) \text{ ft}^3/\text{lbm}}{(1.5435 - 0.01890) \text{ ft}^3/\text{lbm}} = 0.8075$$

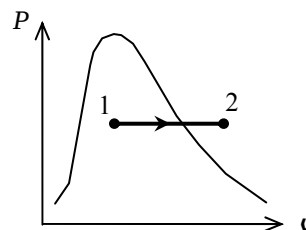
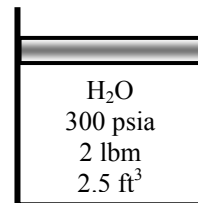
$$s_1 = s_f + x_1 s_{fg} = 0.58818 \text{ Btu/lbm} \cdot \text{R} + (0.8075)(0.92289 \text{ Btu/lbm} \cdot \text{R}) = 1.3334 \text{ Btu/lbm} \cdot \text{R}$$

The final state is superheated vapor and

$$\left. \begin{array}{l} T_2 = 500^\circ\text{F} \\ P_2 = P_1 = 300 \text{ psia} \end{array} \right\} s_2 = 1.5706 \text{ Btu/lbm} \cdot \text{R} \quad (\text{Table A - 6E})$$

Hence, the change in the total entropy is

$$\begin{aligned} \Delta S &= m(s_2 - s_1) \\ &= (2 \text{ lbm})(1.5706 - 1.3334) \text{ Btu/lbm} \cdot \text{R} \\ &= \mathbf{0.4744 \text{ Btu/R}} \end{aligned}$$



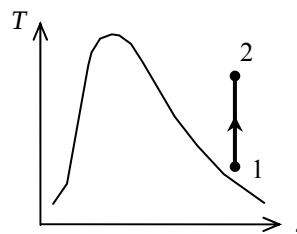
8-37 Water is compressed in a compressor during which the entropy remains constant. The final temperature and enthalpy are to be determined.

Analysis The initial state is superheated vapor and the entropy is

$$\left. \begin{array}{l} T_1 = 160^\circ\text{C} \\ P_1 = 35 \text{ kPa} \end{array} \right\} \begin{array}{l} h_1 = 2800.7 \text{ kJ/kg} \\ s_1 = 8.1531 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{from EES})$$

Note that the properties can also be determined from Table A-6 by interpolation but the values will not be as accurate as those by EES. The final state is superheated vapor and the properties are (Table A-6)

$$\left. \begin{array}{l} P_2 = 300 \text{ kPa} \\ s_2 = s_1 = 8.1531 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \begin{array}{l} T_2 = \mathbf{440.5^\circ\text{C}} \\ h_2 = \mathbf{3361.0 \text{ kJ/kg}} \end{array}$$



8-38E R-134a is expanded isentropically in a closed system. The heat transfer and work production are to be determined.

Assumptions 1 The system is stationary and thus the kinetic and potential energy changes are zero. 2 There are no work interactions involved other than the boundary work. 3 The thermal energy stored in the cylinder itself is negligible. 4 The compression or expansion process is quasi-equilibrium.

Analysis As there is no area under the process line shown on the T - s diagram and this process is reversible,

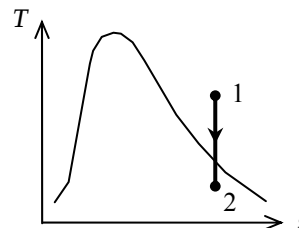
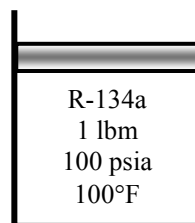
$$Q = 0 \text{ Btu}$$

The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\ -W_{\text{out}} = \Delta U = m(u_2 - u_1) \\ W_{\text{out}} = m(u_1 - u_2)$$

The initial state properties are

$$\left. \begin{array}{l} T_1 = 100^\circ\text{F} \\ P_1 = 100 \text{ psia} \end{array} \right\} \begin{array}{l} u_1 = 109.45 \text{ Btu/lbm} \\ s_1 = 0.22900 \text{ Btu/lbm} \cdot \text{R} \end{array} \quad (\text{Table A-13E})$$



The final state properties for this isentropic process are (Table A-12E)

$$\left. \begin{array}{l} P_2 = 10 \text{ psia} \\ s_2 = s_1 = 0.22900 \text{ Btu/lbm} \cdot \text{R} \end{array} \right\} \begin{array}{l} x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{0.22900 - 0.00742}{0.22206} = 0.99784 \\ u_2 = u_f + x_2 u_{fg} = 3.135 + (0.99784)(87.453) = 90.40 \text{ Btu/lbm} \end{array}$$

Substituting,

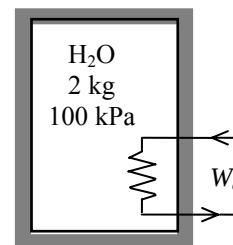
$$W_{\text{out}} = m(u_1 - u_2) = (1 \text{ lbm})(109.45 - 90.40) \text{ Btu/lbm} = \mathbf{19.05 \text{ Btu}}$$

8-39 An insulated rigid tank contains a saturated liquid-vapor mixture of water at a specified pressure. An electric heater inside is turned on and kept on until all the liquid vaporized. The entropy change of the water during this process is to be determined.

Analysis From the steam tables (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 0.25 \end{array} \right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.001 + (0.25)(1.6941 - 0.001) = 0.4243 \text{ m}^3/\text{kg} \\ s_1 = s_f + x_1 s_{fg} = 1.3028 + (0.25)(6.0562) = 2.8168 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} v_2 = v_1 \\ \text{sat. vapor} \end{array} \right\} s_2 = 6.8649 \text{ kJ/kg} \cdot \text{K}$$



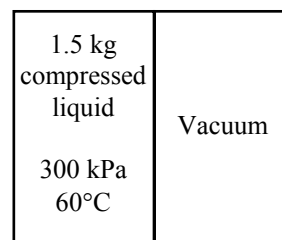
Then the entropy change of the steam becomes

$$\Delta S = m(s_2 - s_1) = (2 \text{ kg})(6.8649 - 2.8168) \text{ kJ/kg} \cdot \text{K} = \mathbf{8.10 \text{ kJ/K}}$$

8-40 CD EES A rigid tank is divided into two equal parts by a partition. One part is filled with compressed liquid water while the other side is evacuated. The partition is removed and water expands into the entire tank. The entropy change of the water during this process is to be determined.

Analysis The properties of the water are (Table A-4)

$$\left. \begin{array}{l} P_1 = 300 \text{ kPa} \\ T_1 = 60^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 \cong v_{f@60^\circ\text{C}} = 0.001017 \text{ m}^3/\text{kg} \\ s_1 = s_{f@60^\circ\text{C}} = 0.8313 \text{ kJ/kg} \cdot \text{K} \end{array}$$



Noting that

$$v_2 = 2v_1 = (2)(0.001017) = 0.002034 \text{ m}^3/\text{kg}$$

$$\left. \begin{array}{l} P_2 = 15 \text{ kPa} \\ v_2 = 0.002034 \text{ m}^3/\text{kg} \end{array} \right\} \begin{array}{l} x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002034 - 0.001014}{10.02 - 0.001014} = 0.0001018 \\ s_2 = s_f + x_2 s_{fg} = 0.7549 + (0.0001018)(7.2522) = 0.7556 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Then the entropy change of the water becomes

$$\Delta S = m(s_2 - s_1) = (1.5 \text{ kg})(0.7556 - 0.8313) \text{ kJ/kg} \cdot \text{K} = \mathbf{-0.114 \text{ kJ/K}}$$

8-41 EES Problem 8-40 is reconsidered. The entropy generated is to be evaluated and plotted as a function of surroundings temperature, and the values of the surroundings temperatures that are valid for this problem are to be determined. The surrounding temperature is to vary from 0°C to 100°C.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Input Data"

P[1]=300 [kPa]

T[1]=60 [C]

m=1.5 [kg]

P[2]=15 [kPa]

Fluid\$='Steam_IAPWS'

V[1]=m*spv[1]

spv[1]=volume(Fluid\$,T=T[1], P=P[1]) "specific volume of steam at state 1, m³/kg"

s[1]=entropy(Fluid\$,T=T[1],P=P[1]) "entropy of steam at state 1, kJ/kgK"

V[2]=2*V[1] "Steam expands to fill entire volume at state 2"

"State 2 is identified by P[2] and spv[2]"

spv[2]=V[2]/m "specific volume of steam at state 2, m³/kg"

s[2]=entropy(Fluid\$,P=P[2],v=spv[2]) "entropy of steam at state 2, kJ/kgK"

T[2]=temperature(Fluid\$,P=P[2],v=spv[2])

DELTAS_sys=m*(s[2]-s[1]) "Total entropy change of steam, kJ/K"

"What does the first law tell us about this problem?"

"Conservation of Energy for the entire, closed system"

E_in - E_out = DELTAE_sys

"neglecting changes in KE and PE for the system:"

DELTAE_sys=m*(intenergy(Fluid\$, P=P[2], v=spv[2]) - intenergy(Fluid\$,T=T[1],P=P[1]))

E_in = 0

"How do you interpret the energy leaving the system, E_out? Recall this is a constant volume system."

Q_out = E_out

"What is the maximum value of the Surroundings temperature?"

"The maximum possible value for the surroundings temperature occurs when we set

S_gen = 0=Delta S_sys+sum(DeltaS_surr)"

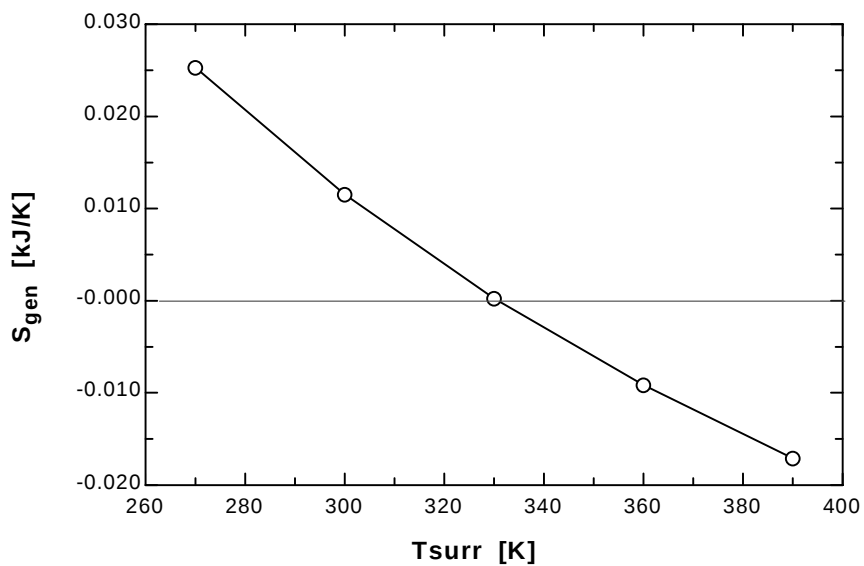
Q_net_surr=Q_out

S_gen = 0

S_gen = DELTAS_sys+Q_net_surr/Tsurr

"Establish a parametric table for the variables S_gen, Q_net_surr, T_surr, and DELTAS_sys. In the Parametric Table window select T_surr and insert a range of values. Then place '{' and '}' about the S_gen = 0 line; press F3 to solve the table. The results are shown in Plot Window 1. What values of T_surr are valid for this problem?"

S_{gen} [kJ/K]	$Q_{\text{net,surr}}$ [kJ]	T_{surr} [K]	ΔS_{sys} [kJ/K]
0.02533	37.44	270	-0.1133
0.01146	37.44	300	-0.1133
0.0001205	37.44	330	-0.1133
-0.009333	37.44	360	-0.1133
-0.01733	37.44	390	-0.1133



8-42E A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled and condensed at constant pressure. The entropy change of refrigerant during this process is to be determined

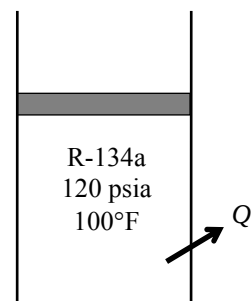
Analysis From the refrigerant tables (Tables A-11E through A-13E),

$$\left. \begin{array}{l} P_1 = 120 \text{ psia} \\ T_1 = 100^\circ\text{F} \end{array} \right\} s_1 = 0.22361 \text{ Btu/lbm} \cdot \text{R}$$

$$\left. \begin{array}{l} T_2 = 50^\circ\text{F} \\ P_2 = 120 \text{ psia} \end{array} \right\} s_2 \cong s_{f@90^\circ\text{F}} = 0.06039 \text{ Btu/lbm} \cdot \text{R}$$

Then the entropy change of the refrigerant becomes

$$\Delta S = m(s_2 - s_1) = (2 \text{ lbm})(0.06039 - 0.22361) \text{ Btu/lbm} \cdot \text{R} = \mathbf{-0.3264 \text{ Btu/R}}$$



8-43 An insulated cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically at constant pressure. The entropy change of the water during this process is to be determined.

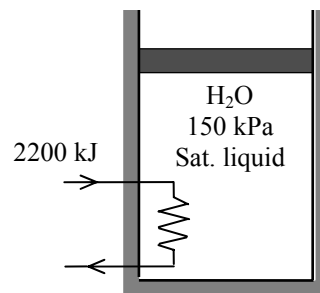
Assumptions 1 The kinetic and potential energy changes are negligible. 2 The cylinder is well-insulated and thus heat transfer is negligible. 3 The thermal energy stored in the cylinder itself is negligible. 4 The compression or expansion process is quasi-equilibrium.

Analysis From the steam tables (Tables A-4 through A-6),

$$P_1 = 150 \text{ kPa} \quad \left\{ \begin{array}{l} \nu_1 = \nu_{f@150 \text{ kPa}} = 0.001053 \text{ m}^3/\text{kg} \\ h_1 = h_{f@150 \text{ kPa}} = 467.13 \text{ kJ/kg} \\ s_1 = s_{f@150 \text{ kPa}} = 1.4337 \text{ kJ/kg} \cdot \text{K} \end{array} \right. \quad \text{sat. liquid}$$

Also,

$$m = \frac{\nu}{\nu_1} = \frac{0.005 \text{ m}^3}{0.001053 \text{ m}^3/\text{kg}} = 4.75 \text{ kg}$$



We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1)$$

since $\Delta U + W_{\text{b}} = \Delta H$ during a constant pressure quasi-equilibrium process. Solving for h_2 ,

$$h_2 = h_1 + \frac{W_{\text{e,in}}}{m} = 467.13 + \frac{2200 \text{ kJ}}{4.75 \text{ kg}} = 930.33 \text{ kJ/kg}$$

Thus,

$$\left. \begin{array}{l} P_2 = 150 \text{ kPa} \\ h_2 = 930.33 \text{ kJ/kg} \end{array} \right\} \begin{array}{l} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{930.33 - 467.13}{2226.0} = 0.2081 \\ s_2 = s_f + x_2 s_{fg} = 1.4337 + (0.2081)(5.7894) = 2.6384 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Then the entropy change of the water becomes

$$\Delta S = m(s_2 - s_1) = (4.75 \text{ kg})(2.6384 - 1.4337) \text{ kJ/kg} \cdot \text{K} = 5.72 \text{ kJ/K}$$

8-44 An insulated cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant expands in a reversible manner until the pressure drops to a specified value. The final temperature in the cylinder and the work done by the refrigerant are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

Analysis (a) This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$P_1 = 0.8 \text{ MPa} \left\{ \begin{array}{l} v_1 = v_{g@0.8 \text{ MPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_1 = u_{g@0.8 \text{ MPa}} = 246.79 \text{ kJ/kg} \\ s_1 = s_{g@0.8 \text{ MPa}} = 0.91835 \text{ kJ/kg} \cdot \text{K} \end{array} \right. \text{ sat. vapor}$$

Also,

$$m = \frac{V}{v_1} = \frac{0.05 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 1.952 \text{ kg}$$

and

$$P_2 = 0.4 \text{ MPa} \left\{ \begin{array}{l} x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{0.91835 - 0.24761}{0.67929} = 0.9874 \\ u_2 = u_f + x_2 u_{fg} = 63.62 + (0.9874)(171.45) = 232.91 \text{ kJ/kg} \end{array} \right. \quad s_2 = s_1$$

$$T_2 = T_{\text{sat}@0.4 \text{ MPa}} = \mathbf{8.91^\circ\text{C}}$$

(b) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this adiabatic closed system can be expressed as

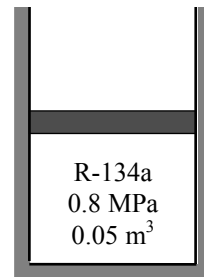
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-W_{\text{b,out}} = \Delta U$$

$$W_{\text{b,out}} = m(u_1 - u_2)$$

Substituting, the work done during this isentropic process is determined to be

$$W_{\text{b,out}} = m(u_1 - u_2) = (1.952 \text{ kg})(246.79 - 232.91) \text{ kJ/kg} = \mathbf{27.09 \text{ kJ}}$$



8-45 EES Problem 8-44 is reconsidered. The work done by the refrigerant is to be calculated and plotted as a function of final pressure as the pressure varies from 0.8 MPa to 0.4 MPa. The work done for this process is to be compared to one for which the temperature is constant over the same pressure range.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

Procedure

```
IsothermWork(P_1,x_1,m_sys,P_2:Work_out_Isotherm,Q_isotherm,DELTAE_isotherm,T_isotherm)
```

```
T_isotherm=Temperature(R134a,P=P_1,x=x_1)
```

```
T=T_isotherm
```

```
u_1 = INTENERGY(R134a,P=P_1,x=x_1)
```

```
v_1 = volume(R134a,P=P_1,x=x_1)
```

```
s_1 = entropy(R134a,P=P_1,x=x_1)
```

```
u_2 = INTENERGY(R134a,P=P_2,T=T)
```

```
s_2 = entropy(R134a,P=P_2,T=T)
```

"The process is reversible and Isothermal thus the heat transfer is determined by:"

```
Q_isotherm = (T+273)*m_sys*(s_2 - s_1)
```

```
DELTAE_isotherm = m_sys*(u_2 - u_1)
```

```
E_in = Q_isotherm
```

```
E_out = DELTAE_isotherm+E_in
```

```
Work_out_isotherm=E_out
```

```
END
```

"Knowns:"

```
P_1 = 800 [kPa]
```

```
x_1 = 1.0
```

```
V_sys = 0.05[m^3]
```

```
"P_2 = 400 [kPa]"
```

"Analysis: "

" Treat the rigid tank as a closed system, with no heat transfer in, neglect changes in KE and PE of the R134a."

"The isentropic work is determined from:"

```
E_in - E_out = DELTAE_sys
```

```
E_out = Work_out_isen
```

```
E_in = 0
```

```
DELTAE_sys = m_sys*(u_2 - u_1)
```

```
u_1 = INTENERGY(R134a,P=P_1,x=x_1)
```

```
v_1 = volume(R134a,P=P_1,x=x_1)
```

```
s_1 = entropy(R134a,P=P_1,x=x_1)
```

```
V_sys = m_sys*v_1
```

"Rigid Tank: The process is reversible and adiabatic or isentropic.

Then P_2 and s_2 specify state 2."

```
s_2 = s_1
```

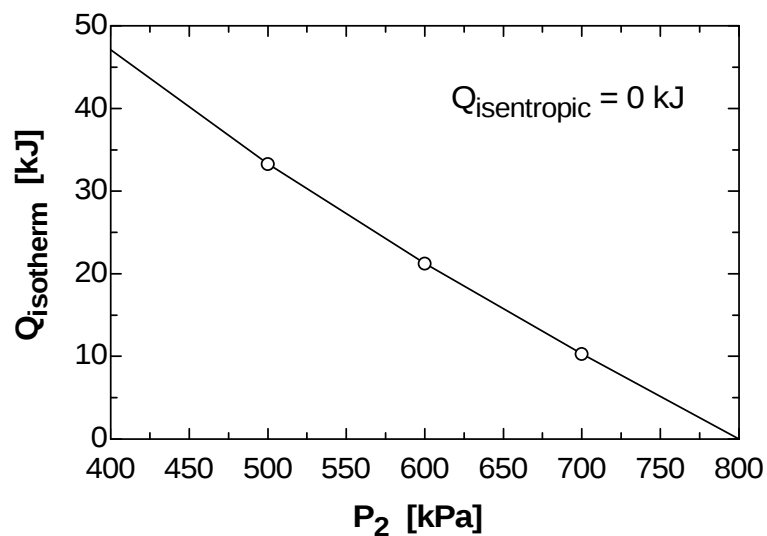
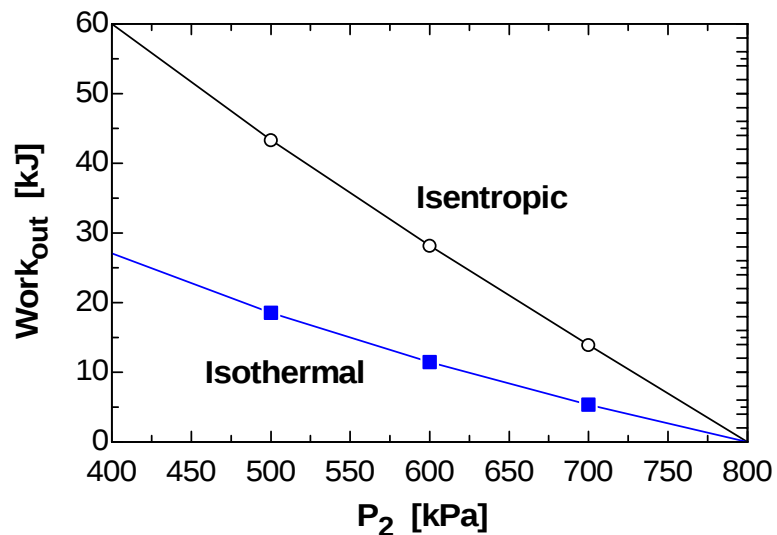
```
u_2 = INTENERGY(R134a,P=P_2,s=s_2)
```

```
T_2_isen = temperature(R134a,P=P_2,s=s_2)
```

Call

```
IsothermWork(P_1,x_1,m_sys,P_2:Work_out_Isotherm,Q_isotherm,DELTAE_isotherm,T_isotherm)
```

P_2 [kPa]	$W_{out,isen}$ [kJ]	$W_{out,isotherm}$ [kJ]	$Q_{isotherm}$ [kJ]
400	27.09	60.02	47.08
500	18.55	43.33	33.29
600	11.44	28.2	21.25
700	5.347	13.93	10.3
800	0	0	0



8-46 Saturated Refrigerant-134a vapor at 160 kPa is compressed steadily by an adiabatic compressor. The minimum power input to the compressor is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

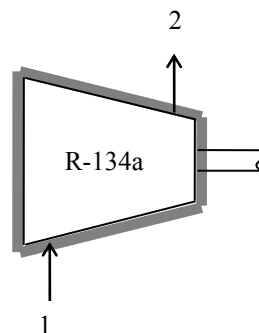
Analysis The power input to an adiabatic compressor will be a minimum when the compression process is reversible. For the reversible adiabatic process we have $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left. \begin{array}{l} P_1 = 160 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} v_1 = v_{g@160 \text{ kPa}} = 0.12348 \text{ m}^3/\text{kg} \\ h_1 = h_{g@160 \text{ kPa}} = 241.11 \text{ kJ/kg} \\ s_1 = s_{g@160 \text{ kPa}} = 0.9419 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 900 \text{ kPa} \\ s_2 = s_1 \end{array} \right\} h_2 = 277.06 \text{ kJ/kg}$$

Also,

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{2 \text{ m}^3/\text{min}}{0.12348 \text{ m}^3/\text{kg}} = 16.20 \text{ kg/min} = 0.27 \text{ kg/s}$$



There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting, the minimum power supplied to the compressor is determined to be

$$\dot{W}_{\text{in}} = (0.27 \text{ kg/s})(277.06 - 241.11) \text{ kJ/kg} = \mathbf{9.71 \text{ kW}}$$

8-47 An insulated cylinder is initially filled with superheated steam at a specified state. The steam is compressed in a reversible manner until the pressure drops to a specified value. The work input during this process is to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

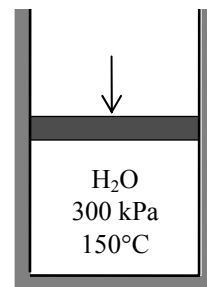
Analysis This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left. \begin{array}{l} P_1 = 300 \text{ kPa} \\ T_1 = 150^\circ\text{C} \end{array} \right\} \begin{array}{l} \nu_1 = 0.63402 \text{ m}^3/\text{kg} \\ u_1 = 2571.0 \text{ kJ/kg} \\ s_1 = 7.0792 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 1 \text{ MPa} \\ s_2 = s_1 \end{array} \right\} \begin{array}{l} u_2 = 2773.8 \text{ kJ/kg} \end{array}$$

Also,

$$m = \frac{\nu}{\nu_1} = \frac{0.05 \text{ m}^3}{0.63402 \text{ m}^3/\text{kg}} = 0.0789 \text{ kg}$$



We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this adiabatic closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

Substituting, the work input during this adiabatic process is determined to be

$$W_{\text{b,in}} = m(u_2 - u_1) = (0.0789 \text{ kg})(2773.8 - 2571.0) \text{ kJ/kg} = \mathbf{16.0 \text{ kJ}}$$

8-48 EES Problem 8-47 is reconsidered. The work done on the steam is to be determined and plotted as a function of final pressure as the pressure varies from 300 kPa to 1 MPa.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Knowns:"

P_1 = 300 [kPa]

T_1 = 150 [C]

V_sys = 0.05 [m^3]

"P_2 = 1000 [kPa]"

"Analysis: "

Fluid\$='Steam_IAPWS'

" Treat the piston-cylinder as a closed system, with no heat transfer in, neglect changes in KE and PE of the Steam. The process is reversible and adiabatic thus isentropic."

"The isentropic work is determined from:"

E_in - E_out = DELTAE_sys

E_out = 0 [kJ]

E_in = Work_in

DELTAE_sys = m_sys*(u_2 - u_1)

u_1 = INTENERGY(Fluid\$,P=P_1,T=T_1)

v_1 = volume(Fluid\$,P=P_1,T=T_1)

s_1 = entropy(Fluid\$,P=P_1,T=T_1)

V_sys = m_sys*v_1

" The process is reversible and adiabatic or isentropic.

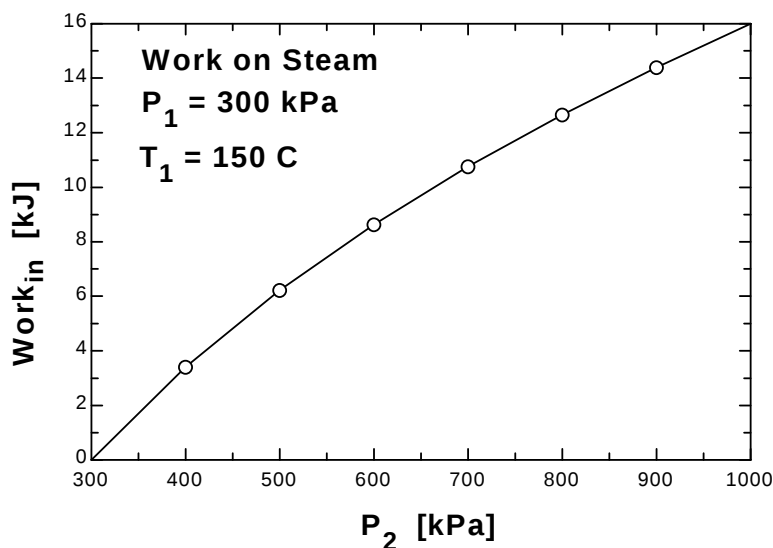
Then P_2 and s_2 specify state 2."

s_2 = s_1

u_2 = INTENERGY(Fluid\$,P=P_2,s=s_2)

T_2_isen = temperature(Fluid\$,P=P_2,s=s_2)

P ₂ [kPa]	Work _{in} [kJ]
300	0
400	3.411
500	6.224
600	8.638
700	10.76
800	12.67
900	14.4
1000	16



8-49 A cylinder is initially filled with saturated water vapor at a specified temperature. Heat is transferred to the steam, and it expands in a reversible and isothermal manner until the pressure drops to a specified value. The heat transfer and the work output for this process are to be determined.

Assumptions 1 The kinetic and potential energy changes are negligible. 2 The cylinder is well-insulated and thus heat transfer is negligible. 3 The thermal energy stored in the cylinder itself is negligible. 4 The process is stated to be reversible and isothermal.

Analysis From the steam tables (Tables A-4 through A-6),

$$\left. \begin{array}{l} T_1 = 200^\circ\text{C} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} u_1 = u_{g@200^\circ\text{C}} = 2594.2 \text{ kJ/kg} \\ s_1 = s_{g@200^\circ\text{C}} = 6.4302 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = T_1 \end{array} \right\} \begin{array}{l} u_2 = 2631.1 \text{ kJ/kg} \\ s_2 = 6.8177 \text{ kJ/kg} \cdot \text{K} \end{array}$$

The heat transfer for this reversible isothermal process can be determined from

$$Q = T\Delta S = Tm(s_2 - s_1) = (473 \text{ K})(1.2 \text{ kg})(6.8177 - 6.4302) \text{ kJ/kg} \cdot \text{K} = \mathbf{219.9 \text{ kJ}}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

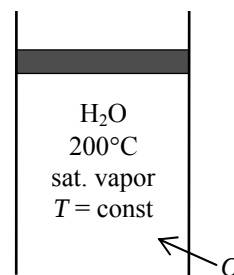
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$W_{\text{b,out}} = Q_{\text{in}} - m(u_2 - u_1)$$

Substituting, the work done during this process is determined to be

$$W_{\text{b,out}} = 219.9 \text{ kJ} - (1.2 \text{ kg})(2631.1 - 2594.2) \text{ kJ/kg} = \mathbf{175.6 \text{ kJ}}$$



8-50 EES Problem 8-49 is reconsidered. The heat transferred to the steam and the work done are to be determined and plotted as a function of final pressure as the pressure varies from the initial value to the final value of 800 kPa.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Knowns:"

$T_1 = 200$ [C]

$x_1 = 1.0$

$m_{\text{sys}} = 1.2$ [kg]

{ $P_2 = 800$ "[kPa]"}

"Analysis: "

Fluid\$='Steam_IAPWS'

" Treat the piston-cylinder as a closed system, neglect changes in KE and PE of the Steam. The process is reversible and isothermal ."

$T_2 = T_1$

$E_{\text{in}} - E_{\text{out}} = \text{DELTA}E_{\text{sys}}$

$E_{\text{in}} = Q_{\text{in}}$

$E_{\text{out}} = \text{Work}_{\text{out}}$

$\text{DELTA}E_{\text{sys}} = m_{\text{sys}}(u_2 - u_1)$

$P_1 = \text{pressure}(\text{Fluid}\$, T=T_1, x=1.0)$

$u_1 = \text{INTENERGY}(\text{Fluid}\$, T=T_1, x=1.0)$

$v_1 = \text{volume}(\text{Fluid}\$, T=T_1, x=1.0)$

$s_1 = \text{entropy}(\text{Fluid}\$, T=T_1, x=1.0)$

$V_{\text{sys}} = m_{\text{sys}}v_1$

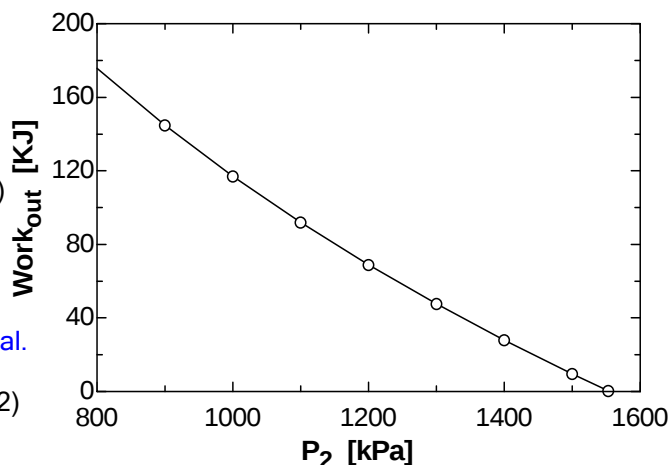
" The process is reversible and isothermal.

Then P_2 and T_2 specify state 2."

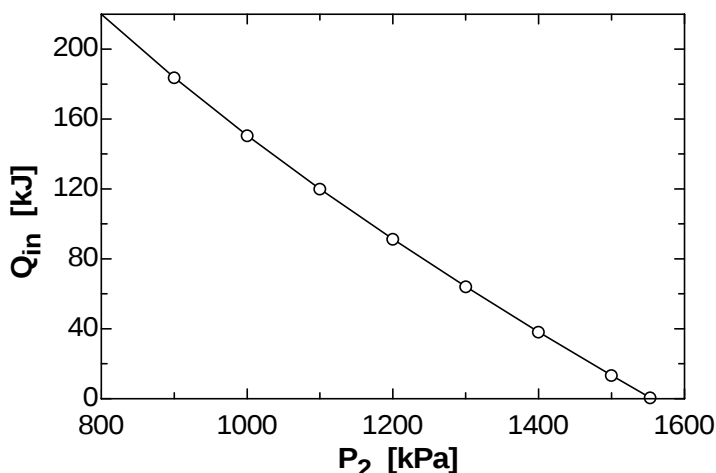
$u_2 = \text{INTENERGY}(\text{Fluid}\$, P=P_2, T=T_2)$

$s_2 = \text{entropy}(\text{Fluid}\$, P=P_2, T=T_2)$

$Q_{\text{in}} = (T_1 + 273)m_{\text{sys}}(s_2 - s_1)$



P ₂ [kPa]	Q _{in} [kJ]	Work _{out} [kJ]
800	219.9	175.7
900	183.7	144.7
1000	150.6	117
1100	120	91.84
1200	91.23	68.85
1300	64.08	47.65
1400	38.2	27.98
1500	13.32	9.605
1553	219.9	175.7



8-51 Water is compressed isentropically in a closed system. The work required is to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{in}} = \Delta U = m(u_2 - u_1)$$

The initial state properties are

$$u_1 = u_f + x_1 u_{fg} = 42.020 + (0.814)(2346.6) = 1952.2 \text{ kJ/kg}$$

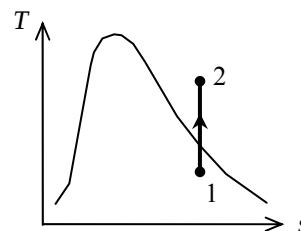
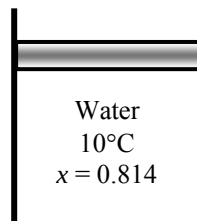
$$s_1 = s_f + x_1 s_{fg} = 0.1511 + (0.814)(8.7488) = 7.2726 \text{ kJ/kg} \cdot \text{K}$$

Since the entropy is constant during this process,

$$\left. \begin{array}{l} P_2 = 3 \text{ MPa} \\ s_2 = s_1 = 7.2726 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} u_2 = 3132.6 \text{ kJ/kg} \quad (\text{Table A - 6})$$

Substituting,

$$w_{\text{in}} = u_2 - u_1 = (3132.6 - 1952.2) \text{ kJ/kg} = \mathbf{1180.4 \text{ kJ/kg}}$$



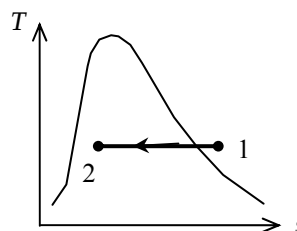
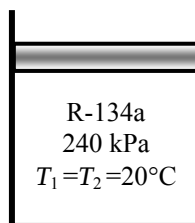
8-52 R-134a undergoes an isothermal process in a closed system. The work and heat transfer are to be determined.

Assumptions 1 The system is stationary and thus the kinetic and potential energy changes are zero. 2 There are no work interactions involved other than the boundary work. 3 The thermal energy stored in the cylinder itself is negligible. 4 The compression or expansion process is quasi-equilibrium.

Analysis The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$



The initial state properties are

$$\left. \begin{array}{l} P_1 = 240 \text{ kPa} \\ T_1 = 20^\circ\text{C} \end{array} \right\} \begin{array}{l} u_1 = 246.74 \text{ kJ/kg} \\ s_1 = 1.0134 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{Table A-13})$$

For this isothermal process, the final state properties are (Table A-11)

$$\left. \begin{array}{l} T_2 = T_1 = 20^\circ\text{C} \\ x_2 = 0.20 \end{array} \right\} \begin{array}{l} u_2 = u_f + x_2 u_{fg} = 78.86 + (0.20)(162.16) = 111.29 \text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 0.30063 + (0.20)(0.62172) = 0.42497 \text{ kJ/kg} \cdot \text{K} \end{array}$$

The heat transfer is determined from

$$q_{\text{in}} = T_0(s_2 - s_1) = (293 \text{ K})(0.42497 - 1.0134) \text{ kJ/kg} \cdot \text{K} = -172.4 \text{ kJ/kg}$$

The negative sign shows that the heat is actually transferred from the system. That is,

$$q_{\text{out}} = \mathbf{172.4 \text{ kJ/kg}}$$

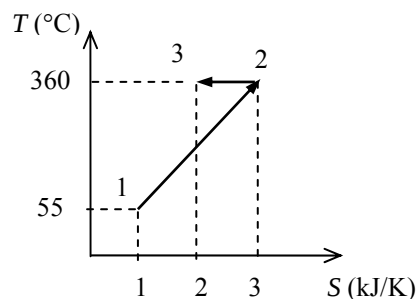
The work required is determined from the energy balance to be

$$w_{\text{in}} = q_{\text{out}} + (u_2 - u_1) = 172.4 \text{ kJ/kg} + (111.29 - 246.74) \text{ kJ/kg} = \mathbf{36.95 \text{ kJ/kg}}$$

8-53 The total heat transfer for the process 1-3 shown in the figure is to be determined.

Analysis For a reversible process, the area under the process line in $T-s$ diagram is equal to the heat transfer during that process. Then,

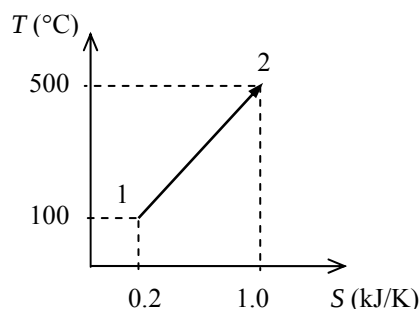
$$\begin{aligned} Q_{1-3} &= Q_{1-2} + Q_{2-3} \\ &= \int_1^2 T dS + \int_2^3 T dS \\ &= \frac{T_1 + T_2}{2}(S_2 - S_1) + T_2(S_3 - S_2) \\ &= \frac{(360 + 273) + (55 + 273) \text{ K}}{2}(3 - 1) \text{ kJ/K} + (360 + 273 \text{ K})(2 - 3) \text{ kJ/K} \\ &= \mathbf{328 \text{ kJ}} \end{aligned}$$



8-54 The total heat transfer for the process 1-2 shown in the figure is to be determined.

Analysis For a reversible process, the area under the process line in T - s diagram is equal to the heat transfer during that process. Then,

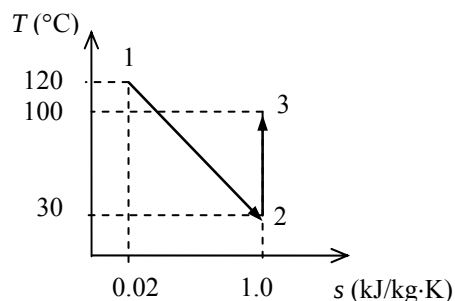
$$\begin{aligned} Q_{1-2} &= \int_1^2 T dS \\ &= \frac{T_1 + T_2}{2} (S_2 - S_1) \\ &= \frac{(500 + 273) + (100 + 273) \text{ K}}{2} (1.0 - 0.2) \text{ kJ/K} \\ &= \mathbf{458.4 \text{ kJ}} \end{aligned}$$



8-55 The heat transfer for the process 1-3 shown in the figure is to be determined.

Analysis For a reversible process, the area under the process line in T - s diagram is equal to the heat transfer during that process. Then,

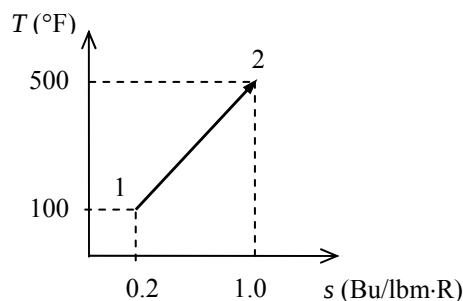
$$\begin{aligned} q_{1-3} &= \int_1^2 T ds + \int_2^3 T ds \\ &= \frac{T_1 + T_2}{2} (s_2 - s_1) + 0 \\ &= \frac{(120 + 273) + (30 + 273) \text{ K}}{2} (1.0 - 0.02) \text{ kJ/kg} \cdot \text{K} \\ &= \mathbf{341.0 \text{ kJ/kg}} \end{aligned}$$



8-56E The total heat transfer for the process 1-2 shown in the figure is to be determined.

Analysis For a reversible process, the area under the process line in T - s diagram is equal to the heat transfer during that process. Then,

$$\begin{aligned} Q_{1-2} &= \int_1^2 T dS \\ &= \frac{T_1 + T_2}{2} (s_2 - s_1) \\ &= \frac{(500 + 460) + (100 + 460) \text{ R}}{2} (1.0 - 0.2) \text{ Btu/lbm} \cdot \text{R} \\ &= \mathbf{608 \text{ Btu/lbm}} \end{aligned}$$



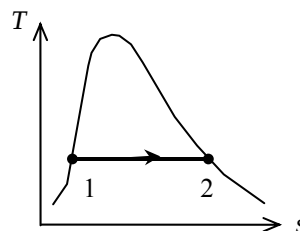
8-57 The change in the entropy of R-134a as it is heated at constant pressure is to be calculated using the relation $ds = (\delta Q / T)_{\text{int rev}}$, and it is to be verified by using R-134a tables.

Analysis As R-134a is converted from a saturated liquid to a saturated vapor, both the pressure and temperature remains constant. Then, the relation $ds = (\delta Q / T)_{\text{int rev}}$ reduces to

$$ds = \frac{dh}{T}$$

When this result is integrated between the saturated liquid and saturated vapor states, the result is (Table A-12)

$$\begin{aligned} s_g - s_f &= \frac{h_g - h_f}{T} = \frac{h_{fg @ 200 \text{ kPa}}}{T_{\text{sat @ 200 kPa}}} \\ &= \frac{206.03 \text{ kJ/kg}}{(-10.09 + 273.15) \text{ K}} = \mathbf{0.78321 \text{ kJ/kg} \cdot \text{K}} \end{aligned}$$



Finding the result directly from the R-134a tables

$$s_g - s_f = s_{fg @ 200 \text{ kPa}} = \mathbf{0.78316 \text{ kJ/kg} \cdot \text{K}} \quad (\text{Table A-12})$$

The two results are practically identical.

8-58 Steam is expanded in an isentropic turbine. The work produced is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

The inlet state properties are

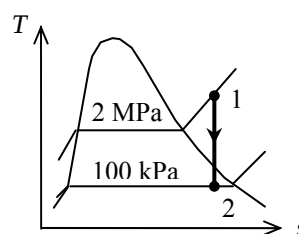
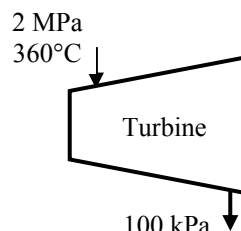
$$\left. \begin{array}{l} P_1 = 2 \text{ MPa} \\ T_1 = 360^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 3159.9 \text{ kJ/kg} \\ s_1 = 6.9938 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{Table A-6})$$

For this isentropic process, the final state properties are (Table A-5)

$$\left. \begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 = 6.9938 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \begin{array}{l} x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{6.9938 - 1.3028}{6.0562} = 0.9397 \\ h_2 = h_f + x_2 h_{fg} = 417.51 + (0.9397)(2257.5) = 2538.9 \text{ kJ/kg} \end{array}$$

Substituting,

$$w_{\text{out}} = h_1 - h_2 = (3159.9 - 2538.9) \text{ kJ/kg} = \mathbf{621.0 \text{ kJ/kg}}$$



8-59 R-134a is compressed in an isentropic compressor. The work required is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\dot{\Delta E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

The inlet state properties are

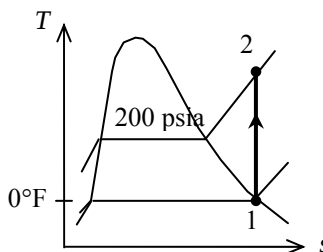
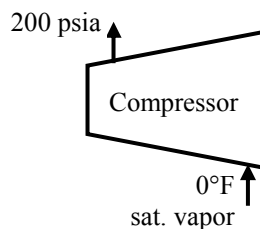
$$\left. \begin{array}{l} T_1 = 0^\circ\text{F} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} h_1 = 103.08 \text{ Btu/lbm} \\ s_1 = 0.22539 \text{ Btu/lbm} \cdot \text{R} \end{array} \quad (\text{Table A-11E})$$

For this isentropic process, the final state enthalpy is

$$\left. \begin{array}{l} P_2 = 200 \text{ psia} \\ s_2 = s_1 = 0.22539 \text{ Btu/lbm} \cdot \text{R} \end{array} \right\} h_2 = 123.28 \text{ Btu/lbm} \quad (\text{Table A-13E})$$

Substituting,

$$w_{\text{in}} = h_2 - h_1 = (123.28 - 103.08) \text{ Btu/lbm} = \mathbf{20.2 \text{ Btu/lbm}}$$



8-60 Steam is expanded in an isentropic turbine. The work produced is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

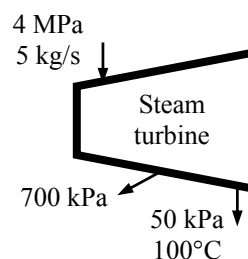
Analysis There is one inlet and two exits. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$



From a mass balance,

$$\dot{m}_2 = 0.05 \dot{m}_1 = (0.05)(5 \text{ kg/s}) = 0.25 \text{ kg/s}$$

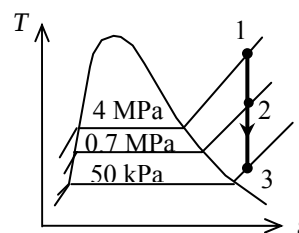
$$\dot{m}_3 = 0.95 \dot{m}_1 = (0.95)(5 \text{ kg/s}) = 4.75 \text{ kg/s}$$

Noting that the expansion process is isentropic, the enthalpies at three states are determined as follows:

$$\left. \begin{array}{l} P_3 = 50 \text{ kPa} \\ T_3 = 100^\circ\text{C} \end{array} \right\} \begin{array}{l} h_3 = 2682.4 \text{ kJ/kg} \\ s_3 = 7.6953 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{Table A - 6})$$

$$\left. \begin{array}{l} P_1 = 4 \text{ MPa} \\ s_1 = s_3 = 7.6953 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} h_1 = 3979.3 \text{ kJ/kg} \quad (\text{Table A - 6})$$

$$\left. \begin{array}{l} P_2 = 700 \text{ kPa} \\ s_2 = s_3 = 7.6953 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} h_2 = 3309.1 \text{ kJ/kg} \quad (\text{Table A - 6})$$



Substituting,

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3 \\ &= (5 \text{ kg/s})(3979.3 \text{ kJ/kg}) - (0.25 \text{ kg/s})(3309.1 \text{ kJ/kg}) - (4.75 \text{ kg/s})(2682.4 \text{ kJ/kg}) \\ &= \mathbf{6328 \text{ kW}} \end{aligned}$$

8-61 Heat is added to a pressure cooker that is maintained at a specified pressure. The minimum entropy change of the thermal-energy reservoir supplying this heat is to be determined.

Assumptions 1 Only water vapor escapes through the pressure relief valve.

Analysis According to the conservation of mass principle,

$$\frac{dm_{CV}}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m}$$

$$\frac{dm}{dt} = -\dot{m}_{out}$$

An entropy balance adapted to this system becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} + \dot{m}_{out} s \geq 0$$

When this is combined with the mass balance, it becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} - s \frac{dm}{dt} \geq 0$$

Multiplying by dt and integrating the result yields

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out} (m_2 - m_1) \geq 0$$

The properties at the initial and final states are (from Table A-5 at 200 kPa)

$$\nu_1 = \nu_f + x \nu_{fg} = 0.001061 + (0.10)(0.88578 - 0.001061) = 0.08953 \text{ m}^3/\text{kg}$$

$$s_1 = s_f + x s_{fg} = 1.5302 + (0.10)(5.5968) = 2.0899 \text{ kJ/kg} \cdot \text{K}$$

$$\nu_2 = \nu_f + x \nu_{fg} = 0.001061 + (0.50)(0.88578 - 0.001061) = 0.4434 \text{ m}^3/\text{kg}$$

$$s_2 = s_f + x s_{fg} = 1.5302 + (0.50)(5.5968) = 4.3286 \text{ kJ/kg} \cdot \text{K}$$

The initial and final masses are

$$m_1 = \frac{\nu}{\nu_1} = \frac{\nu}{\nu_f + x \nu_{fg}} = \frac{0.1 \text{ m}^3}{0.08953 \text{ m}^3/\text{kg}} = 1.117 \text{ kg}$$

$$m_2 = \frac{\nu}{\nu_2} = \frac{\nu}{\nu_f + x \nu_{fg}} = \frac{0.1 \text{ m}^3}{0.4434 \text{ m}^3/\text{kg}} = 0.2255 \text{ kg}$$

The entropy of escaping water vapor is

$$s_{out} = s_g @ 200 \text{ kPa} = 7.1270 \text{ kJ/kg} \cdot \text{K}$$

Substituting,

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out} (m_2 - m_1) \geq 0$$

$$\Delta S_{surr} + (0.2255)(4.3286) - (1.117)(2.0899) - (7.1270)(0.2255 - 1.117) \geq 0$$

$$\Delta S_{surr} + 4.995 \geq 0$$

The entropy change of the thermal energy reservoir must then satisfy

$$\Delta S_{surr} \geq -4.995 \text{ kJ/K}$$

8-62 Heat is added to a pressure cooker that is maintained at a specified pressure. Work is also done on water. The minimum entropy change of the thermal-energy reservoir supplying this heat is to be determined.

Assumptions 1 Only water vapor escapes through the pressure relief valve.

Analysis According to the conservation of mass principle,

$$\frac{dm_{CV}}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m}$$

$$\frac{dm}{dt} = -\dot{m}_{out}$$

An entropy balance adapted to this system becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} + \dot{m}_{out} s \geq 0$$

When this is combined with the mass balance, it becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} - s \frac{dm}{dt} \geq 0$$

Multiplying by dt and integrating the result yields

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out} (m_2 - m_1) \geq 0$$

The properties at the initial and final states are (from Table A-5 at 200 kPa)

$$\nu_1 = \nu_f + x \nu_{fg} = 0.001061 + (0.10)(0.88578 - 0.001061) = 0.08953 \text{ m}^3/\text{kg}$$

$$s_1 = s_f + x s_{fg} = 1.5302 + (0.10)(5.5968) = 2.0899 \text{ kJ/kg} \cdot \text{K}$$

$$\nu_2 = \nu_f + x \nu_{fg} = 0.001061 + (0.50)(0.88578 - 0.001061) = 0.4434 \text{ m}^3/\text{kg}$$

$$s_2 = s_f + x s_{fg} = 1.5302 + (0.50)(5.5968) = 4.3286 \text{ kJ/kg} \cdot \text{K}$$

The initial and final masses are

$$m_1 = \frac{\nu}{\nu_1} = \frac{\nu}{\nu_f + x \nu_{fg}} = \frac{0.1 \text{ m}^3}{0.08953 \text{ m}^3/\text{kg}} = 1.117 \text{ kg}$$

$$m_2 = \frac{\nu}{\nu_2} = \frac{\nu}{\nu_f + x \nu_{fg}} = \frac{0.1 \text{ m}^3}{0.4434 \text{ m}^3/\text{kg}} = 0.2255 \text{ kg}$$

The entropy of escaping water vapor is

$$s_{out} = s_g @ 200 \text{ kPa} = 7.1270 \text{ kJ/kg} \cdot \text{K}$$

Substituting,

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out} (m_2 - m_1) \geq 0$$

$$\Delta S_{surr} + (0.2255)(4.3286) - (1.117)(2.0899) - (7.1270)(0.2255 - 1.117) \geq 0$$

$$\Delta S_{surr} + 4.995 \geq 0$$

The entropy change of the thermal energy reservoir must then satisfy

$$\Delta S_{surr} \geq -4.995 \text{ kJ/K}$$

8-63 A cylinder is initially filled with saturated water vapor mixture at a specified temperature. Steam undergoes a reversible heat addition and an isentropic process. The processes are to be sketched and heat transfer for the first process and work done during the second process are to be determined.

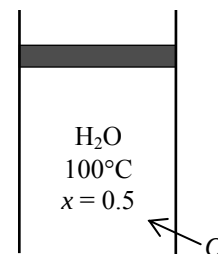
Assumptions 1 The kinetic and potential energy changes are negligible. 2 The thermal energy stored in the cylinder itself is negligible. 3 Both processes are reversible.

Analysis (b) From the steam tables (Tables A-4 through A-6),

$$\left. \begin{array}{l} T_1 = 100^\circ\text{C} \\ x = 0.5 \end{array} \right\} h_1 = h_f + xh_{fg} = 419.17 + (0.5)(2256.4) = 1547.4 \text{ kJ/kg}$$

$$\left. \begin{array}{l} T_2 = 100^\circ\text{C} \\ x_2 = 1 \end{array} \right\} \begin{array}{l} h_2 = h_g = 2675.6 \text{ kJ/kg} \\ u_2 = u_g = 2506.0 \text{ kJ/kg} \\ s_2 = s_g = 7.3542 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_3 = 15 \text{ kPa} \\ s_3 = s_2 \end{array} \right\} u_3 = 2247.9 \text{ kJ/kg}$$



We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

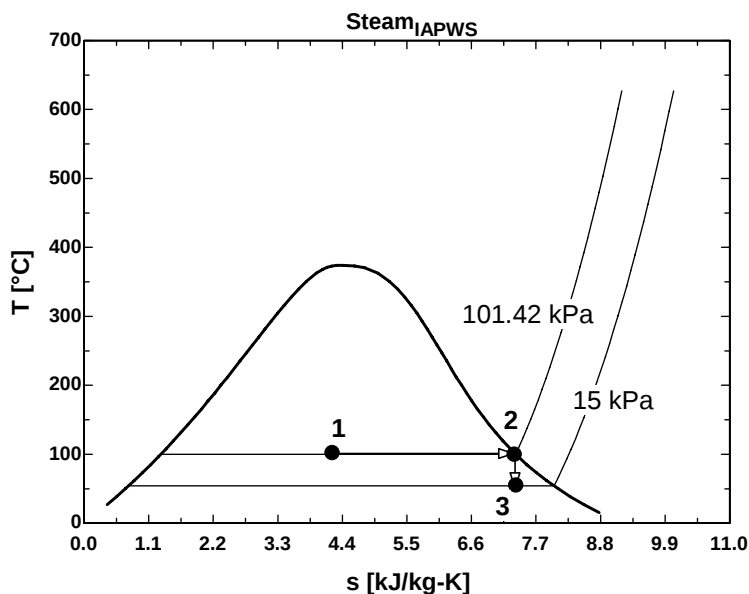
$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

For process 1-2, it reduces to

$$Q_{12,\text{in}} = m(h_2 - h_1) = (5 \text{ kg})(2675.6 - 1547.4) \text{ kJ/kg} = \mathbf{5641 \text{ kJ}}$$

(c) For process 2-3, it reduces to

$$W_{23,\text{b,out}} = m(u_2 - u_3) = (5 \text{ kg})(2506.0 - 2247.9) \text{ kJ/kg} = \mathbf{1291 \text{ kJ}}$$



8-64 Steam expands in an adiabatic turbine. Steam leaves the turbine at two different pressures. The process is to be sketched on a T - s diagram and the work done by the steam per unit mass of the steam at the inlet are to be determined.

Assumptions 1 The kinetic and potential energy changes are negligible.

Analysis (b) From the steam tables (Tables A-4 through A-6),

$$\left. \begin{array}{l} T_1 = 500^\circ\text{C} \\ P_1 = 6 \text{ MPa} \end{array} \right\} \begin{array}{l} h_1 = 3423.1 \text{ kJ/kg} \\ s_1 = 6.8826 \text{ kJ/kg}\cdot\text{K} \end{array} \quad \left. \begin{array}{l} P_2 = 1 \text{ MPa} \\ s_2 = s_1 \end{array} \right\} h_{2s} = 2921.3 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_3 = 10 \text{ kPa} \\ s_3 = s_1 \end{array} \right\} \begin{array}{l} h_{3s} = 2179.6 \text{ kJ/kg} \\ x_{3s} = 0.831 \end{array}$$

A mass balance on the control volume gives

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3 \quad \text{where} \quad \begin{array}{l} \dot{m}_2 = 0.1\dot{m}_1 \\ \dot{m}_3 = 0.9\dot{m}_1 \end{array}$$

We take the turbine as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

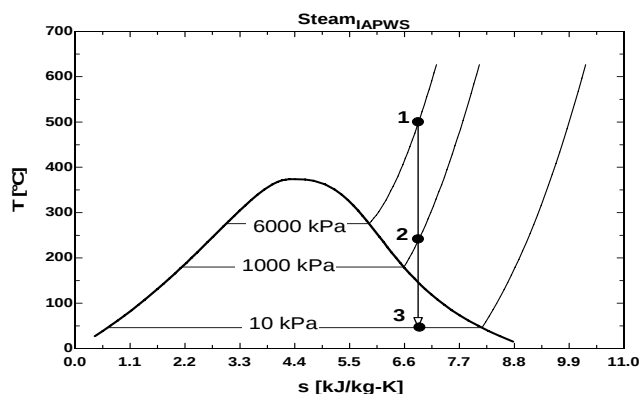
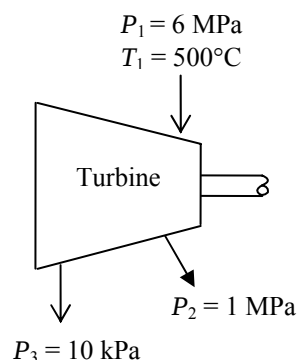
$$\begin{aligned} \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}_1 h_1 &= \dot{W}_{s,\text{out}} + \dot{m}_2 h_2 + \dot{m}_3 h_3 \\ \dot{m}_1 h_1 &= \dot{W}_{s,\text{out}} + 0.1\dot{m}_1 h_2 + 0.9\dot{m}_1 h_3 \end{aligned}$$

or

$$\begin{aligned} h_1 &= w_{s,\text{out}} + 0.1h_2 + 0.9h_3 \\ w_{s,\text{out}} &= h_1 - 0.1h_2 - 0.9h_3 \\ &= 3423.1 - (0.1)(2921.3) - (0.9)(2179.6) = 1169.3 \text{ kJ/kg} \end{aligned}$$

The actual work output per unit mass of steam at the inlet is

$$w_{\text{out}} = \eta_T w_{s,\text{out}} = (0.85)(1169.3 \text{ kJ/kg}) = \mathbf{993.9 \text{ kJ/kg}}$$



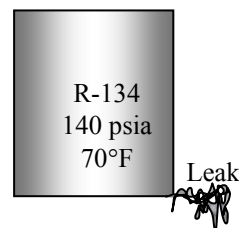
8-65E An insulated rigid can initially contains R-134a at a specified state. A crack develops, and refrigerant escapes slowly. The final mass in the can is to be determined when the pressure inside drops to a specified value.

Assumptions 1 The can is well-insulated and thus heat transfer is negligible. 2 The refrigerant that remains in the can underwent a reversible adiabatic process.

Analysis Noting that for a reversible adiabatic (i.e., isentropic) process, $s_1 = s_2$, the properties of the refrigerant in the can are (Tables A-11E through A-13E)

$$\left. \begin{array}{l} P_1 = 140 \text{ psia} \\ T_1 = 70^\circ\text{F} \end{array} \right\} s_1 \cong s_{f@70^\circ\text{F}} = 0.07306 \text{ Btu/lbm} \cdot \text{R}$$

$$\left. \begin{array}{l} P_2 = 20 \text{ psia} \\ s_2 = s_1 \end{array} \right\} \begin{aligned} x_2 &= \frac{s_2 - s_f}{s_{fg}} = \frac{0.07306 - 0.02605}{0.19962} = 0.2355 \\ \nu_2 &= \nu_f + x_2 \nu_{fg} = 0.01182 + (0.2355)(2.2772 - 0.01182) = 0.5453 \text{ ft}^3/\text{lbm} \end{aligned}$$



Thus the final mass of the refrigerant in the can is

$$m = \frac{\nu}{\nu_2} = \frac{1.2 \text{ ft}^3}{0.5453 \text{ ft}^3/\text{lbm}} = \mathbf{2.201 \text{ lbm}}$$

Entropy Change of Incompressible Substances

8-66C No, because entropy is not a conserved property.

8-67 A hot copper block is dropped into water in an insulated tank. The final equilibrium temperature of the tank and the total entropy change are to be determined.

Assumptions **1** Both the water and the copper block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

Properties The density and specific heat of water at 25°C are $\rho = 997 \text{ kg/m}^3$ and $c_p = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. The specific heat of copper at 27°C is $c_p = 0.386 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis We take the entire contents of the tank, water + copper block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

or,

$$\Delta U_{\text{Cu}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{Cu}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

where

$$m_{\text{water}} = \rho V = (997 \text{ kg/m}^3)(0.120 \text{ m}^3) = 119.6 \text{ kg}$$

Using specific heat values for copper and liquid water at room temperature and substituting,

$$(50 \text{ kg})(0.386 \text{ kJ/kg} \cdot ^\circ\text{C})(T_2 - 80)^\circ\text{C} + (119.6 \text{ kg})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(T_2 - 25)^\circ\text{C} = 0$$

$$T_2 = 27.0^\circ\text{C}$$

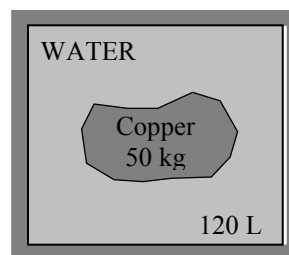
The entropy generated during this process is determined from

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50 \text{ kg})(0.386 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{300.0 \text{ K}}{353 \text{ K}}\right) = -3.140 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (119.6 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{300.0 \text{ K}}{298 \text{ K}}\right) = 3.344 \text{ kJ/K}$$

Thus,

$$\Delta S_{\text{total}} = \Delta S_{\text{copper}} + \Delta S_{\text{water}} = -3.140 + 3.344 = \mathbf{0.204 \text{ kJ/K}}$$



8-68 Computer chips are cooled by placing them in saturated liquid R-134a. The entropy changes of the chips, R-134a, and the entire system are to be determined.

Assumptions 1 The system is stationary and thus the kinetic and potential energy changes are zero. 2 There are no work interactions involved. 3 There is no heat transfer between the system and the surroundings.

Analysis (a) The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$0 = \Delta U = [m(u_2 - u_1)]_{\text{chips}} + [m(u_2 - u_1)]_{\text{R-134a}}$$

$$[m(u_1 - u_2)]_{\text{chips}} = [m(u_2 - u_1)]_{\text{R-134a}}$$

The heat released by the chips is

$$Q_{\text{chips}} = mc(T_1 - T_2) = (0.010 \text{ kg})(0.3 \text{ kJ/kg} \cdot \text{K})[20 - (-40)]\text{K} = 0.18 \text{ kJ}$$

The mass of the refrigerant vaporized during this heat exchange process is

$$m_{g,2} = \frac{Q_{\text{R-134a}}}{u_g - u_f} = \frac{Q_{\text{R-134a}}}{u_{fg @ -40^\circ\text{C}}} = \frac{0.18 \text{ kJ}}{207.40 \text{ kJ/kg}} = 0.0008679 \text{ kg}$$

Only a small fraction of R-134a is vaporized during the process. Therefore, the temperature of R-134a remains constant during the process. The change in the entropy of the R-134a is (at -40°F from Table A-11)

$$\begin{aligned}\Delta S_{\text{R-134a}} &= m_{g,2}s_{g,2} + m_{f,2}s_{f,2} - m_{f,1}s_{f,1} \\ &= (0.0008679)(0.96866) + (0.005 - 0.0008679)(0) - (0.005)(0) \\ &= \mathbf{0.000841 \text{ kJ/K}}\end{aligned}$$

(b) The entropy change of the chips is

$$\Delta S_{\text{chips}} = mc \ln \frac{T_2}{T_1} = (0.010 \text{ kg})(0.3 \text{ kJ/kg} \cdot \text{K}) \ln \frac{(-40 + 273)\text{K}}{(20 + 273)\text{K}} = \mathbf{-0.000687 \text{ kJ/K}}$$

(c) The total entropy change is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{R-134a}} + \Delta S_{\text{chips}} = 0.000841 + (-0.000687) = \mathbf{0.000154 \text{ kJ/K}}$$

The positive result for the total entropy change (i.e., entropy generation) indicates that this process is possible.

8-69 A hot iron block is dropped into water in an insulated tank. The total entropy change during this process is to be determined.

Assumptions 1 Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer. **4** The water that evaporates, condenses back.

Properties The specific heat of water at 25°C is $c_p = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. The specific heat of iron at room temperature is $c_p = 0.45 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis We take the entire contents of the tank, water + iron block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

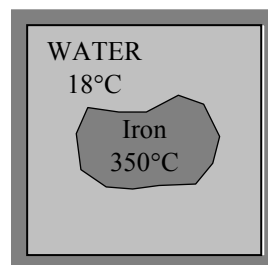
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

or,

$$\Delta U_{\text{iron}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{iron}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$



Substituting,

$$(25 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K})(T_2 - 350^\circ\text{C}) + (100 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K})(T_2 - 18^\circ\text{C}) = 0$$

$$T_2 = \mathbf{26.7^\circ\text{C}}$$

The entropy generated during this process is determined from

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (25 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{299.7 \text{ K}}{623 \text{ K}}\right) = -8.232 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (100 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{299.7 \text{ K}}{291 \text{ K}}\right) = 12.314 \text{ kJ/K}$$

Thus,

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}} = -8.232 + 12.314 = \mathbf{4.08 \text{ kJ/K}}$$

Discussion The results can be improved somewhat by using specific heats at average temperature.

8-70 An aluminum block is brought into contact with an iron block in an insulated enclosure. The final equilibrium temperature and the total entropy change for this process are to be determined.

Assumptions 1 Both the aluminum and the iron block are incompressible substances with constant specific heats. 2 The system is stationary and thus the kinetic and potential energies are negligible. 3 The system is well-insulated and thus there is no heat transfer.

Properties The specific heat of aluminum at the anticipated average temperature of 450 K is $c_p = 0.973$ kJ/kg·°C. The specific heat of iron at room temperature (the only value available in the tables) is $c_p = 0.45$ kJ/kg·°C (Table A-3).

Analysis We take the iron+aluminum blocks as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

Iron 20 kg 100°C	Aluminum 20 kg 200°C
------------------------	----------------------------

or,

$$\Delta U_{\text{alum}} + \Delta U_{\text{iron}} = 0$$

$$[mc(T_2 - T_1)]_{\text{alum}} + [mc(T_2 - T_1)]_{\text{iron}} = 0$$

Substituting,

$$(20 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K})(T_2 - 100^\circ \text{C}) + (20 \text{ kg})(0.973 \text{ kJ/kg} \cdot \text{K})(T_2 - 200^\circ \text{C}) = 0$$

$$T_2 = \mathbf{168.4^\circ \text{C}} = 441.4 \text{ K}$$

The total entropy change for this process is determined from

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln \left(\frac{T_2}{T_1} \right) = (20 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K}) \ln \left(\frac{441.4 \text{ K}}{373 \text{ K}} \right) = 1.515 \text{ kJ/K}$$

$$\Delta S_{\text{alum}} = mc_{\text{avg}} \ln \left(\frac{T_2}{T_1} \right) = (20 \text{ kg})(0.973 \text{ kJ/kg} \cdot \text{K}) \ln \left(\frac{441.4 \text{ K}}{473 \text{ K}} \right) = -1.346 \text{ kJ/K}$$

Thus,

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{alum}} = 1.515 - 1.346 = \mathbf{0.169 \text{ kJ/K}}$$

8-71 EES Problem 8-70 is reconsidered. The effect of the mass of the iron block on the final equilibrium temperature and the total entropy change for the process is to be studied. The mass of the iron is to vary from 1 to 10 kg. The equilibrium temperature and the total entropy change are to be plotted as a function of iron mass.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Knowns:"

$$T_{1_iron} = 100 \text{ [C]}$$

$$\{m_{iron} = 20 \text{ [kg]}\}$$

$$T_{1_al} = 200 \text{ [C]}$$

$$m_{al} = 20 \text{ [kg]}$$

$$C_{al} = 0.973 \text{ [kJ/kg-K]} \text{ "From Table A-3 at the anticipated average temperature of 450 K."}$$

$$C_{iron} = 0.45 \text{ [kJ/kg-K]} \text{ "From Table A-3 at room temperature, the only value available."}$$

"Analysis: "

"Treat the iron plus aluminum as a closed system, with no heat transfer in, no work out, neglect changes in KE and PE of the system. "

"The final temperature is found from the energy balance."

$$E_{in} - E_{out} = \Delta E_{sys}$$

$$E_{out} = 0$$

$$E_{in} = 0$$

$$\Delta E_{sys} = m_{iron} \Delta E_{iron} + m_{al} \Delta E_{al}$$

$$\Delta E_{iron} = C_{iron} (T_{2_iron} - T_{1_iron})$$

$$\Delta E_{al} = C_{al} (T_{2_al} - T_{1_al})$$

"the iron and aluminum reach thermal equilibrium:"

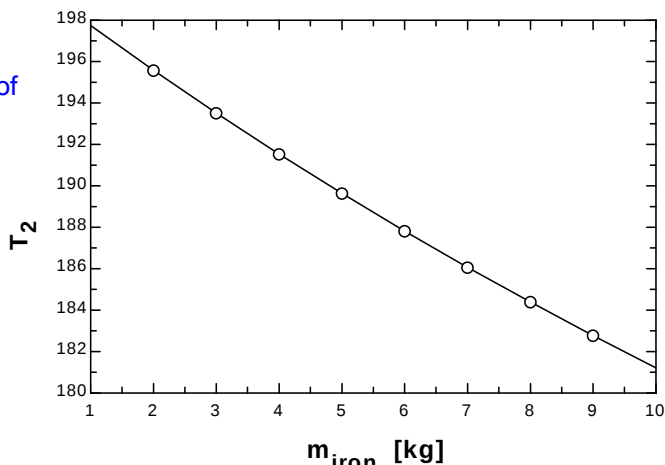
$$T_{2_iron} = T_{2}$$

$$T_{2_al} = T_{2}$$

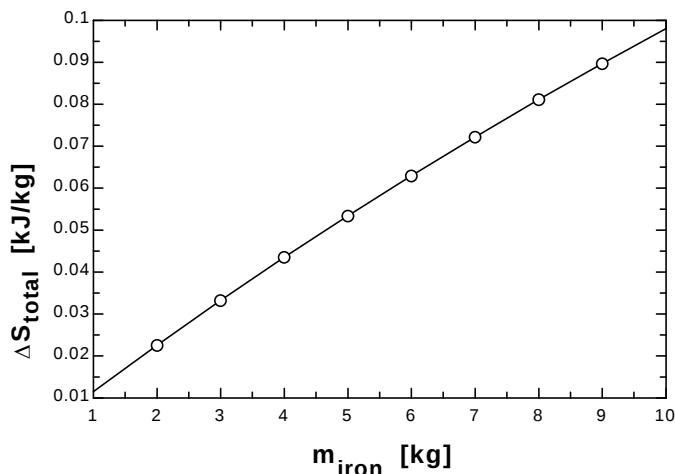
$$\Delta S_{iron} = m_{iron} C_{iron} \ln((T_{2_iron} + 273) / (T_{1_iron} + 273))$$

$$\Delta S_{al} = m_{al} C_{al} \ln((T_{2_al} + 273) / (T_{1_al} + 273))$$

$$\Delta S_{total} = \Delta S_{iron} + \Delta S_{al}$$



ΔS_{total} [kJ/kg]	m_{iron} [kg]	T_2 [C]
0.01152	1	197.7
0.0226	2	195.6
0.03326	3	193.5
0.04353	4	191.5
0.05344	5	189.6
0.06299	6	187.8
0.07221	7	186.1
0.08112	8	184.4
0.08973	9	182.8
0.09805	10	181.2



8-72 An iron block and a copper block are dropped into a large lake. The total amount of entropy change when both blocks cool to the lake temperature is to be determined.

Assumptions 1 The water, the iron block and the copper block are incompressible substances with constant specific heats at room temperature. **2** Kinetic and potential energies are negligible.

Properties The specific heats of iron and copper at room temperature are $c_{\text{iron}} = 0.45 \text{ kJ/kg} \cdot ^\circ\text{C}$ and $c_{\text{copper}} = 0.386 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis The thermal-energy capacity of the lake is very large, and thus the temperatures of both the iron and the copper blocks will drop to the lake temperature (15°C) when the thermal equilibrium is established. Then the entropy changes of the blocks become

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{288 \text{ K}}{353 \text{ K}}\right) = -4.579 \text{ kJ/K}$$

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20 \text{ kg})(0.386 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{288 \text{ K}}{353 \text{ K}}\right) = -1.571 \text{ kJ/K}$$

We take both the iron and the copper blocks, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\ -Q_{\text{out}} = \Delta U = \Delta U_{\text{iron}} + \Delta U_{\text{copper}}$$

or,

$$Q_{\text{out}} = [mc(T_1 - T_2)]_{\text{iron}} + [mc(T_1 - T_2)]_{\text{copper}}$$

Substituting,

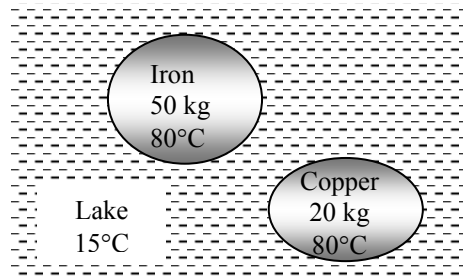
$$Q_{\text{out}} = (50 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K})(353 - 288) \text{ K} + (20 \text{ kg})(0.386 \text{ kJ/kg} \cdot \text{K})(353 - 288) \text{ K} \\ = 1964 \text{ kJ}$$

Thus,

$$\Delta S_{\text{lake}} = \frac{Q_{\text{lake, in}}}{T_{\text{lake}}} = \frac{1964 \text{ kJ}}{288 \text{ K}} = 6.820 \text{ kJ/K}$$

Then the total entropy change for this process is

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{copper}} + \Delta S_{\text{lake}} = -4.579 - 1.571 + 6.820 = \mathbf{0.670 \text{ kJ/K}}$$

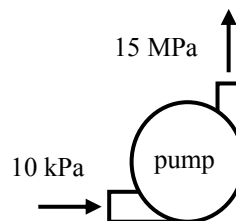


8-73 An adiabatic pump is used to compress saturated liquid water in a reversible manner. The work input is to be determined by different approaches.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible. 3 Heat transfer to or from the fluid is negligible.

Analysis The properties of water at the inlet and exit of the pump are (Tables A-4 through A-6)

$$\begin{aligned} P_1 = 10 \text{ kPa} \quad \left\{ \begin{array}{l} h_1 = 191.81 \text{ kJ/kg} \\ s_1 = 0.6492 \text{ kJ/kg} \\ \nu_1 = 0.001010 \text{ m}^3/\text{kg} \end{array} \right. \\ x_1 = 0 \\ P_2 = 15 \text{ MPa} \quad \left\{ \begin{array}{l} h_2 = 206.90 \text{ kJ/kg} \\ s_2 = s_1 \\ \nu_2 = 0.001004 \text{ m}^3/\text{kg} \end{array} \right. \end{aligned}$$



(a) Using the entropy data from the compressed liquid water table

$$w_p = h_2 - h_1 = 206.90 - 191.81 = \mathbf{15.10 \text{ kJ/kg}}$$

(b) Using inlet specific volume and pressure values

$$w_p = \nu_1 (P_2 - P_1) = (0.001010 \text{ m}^3/\text{kg})(15,000 - 10) \text{ kPa} = \mathbf{15.14 \text{ kJ/kg}}$$

$$\text{Error} = \mathbf{0.3\%}$$

(b) Using average specific volume and pressure values

$$w_p = \nu_{\text{avg}} (P_2 - P_1) = \left[1/2(0.001010 + 0.001004) \text{ m}^3/\text{kg} \right] (15,000 - 10) \text{ kPa} = \mathbf{15.10 \text{ kJ/kg}}$$

$$\text{Error} = \mathbf{0\%}$$

Discussion The results show that any of the method may be used to calculate reversible pump work.

Entropy Changes of Ideal Gases

8-74C For ideal gases, $c_p = c_v + R$ and

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow \frac{v_2}{v_1} = \frac{T_2 P_1}{T_1 P_2}$$

Thus,

$$\begin{aligned} s_2 - s_1 &= c_v \ln \left(\frac{T_2}{T_1} \right) + R \ln \left(\frac{v_2}{v_1} \right) \\ &= c_v \ln \left(\frac{T_2}{T_1} \right) + R \ln \left(\frac{T_2 P_1}{T_1 P_2} \right) \\ &= c_v \ln \left(\frac{T_2}{T_1} \right) + R \ln \left(\frac{T_2}{T_1} \right) - R \ln \left(\frac{P_2}{P_1} \right) \\ &= c_p \ln \left(\frac{T_2}{T_1} \right) - R \ln \left(\frac{P_2}{P_1} \right) \end{aligned}$$

8-75C For an ideal gas, $dh = c_p dT$ and $v = RT/P$. From the second Tds relation,

$$ds = \frac{dh}{T} - \frac{vdP}{T} = \frac{c_p dT}{T} - \frac{RT}{P} \frac{dP}{T} = c_p \frac{dT}{T} - R \frac{dP}{P}$$

Integrating,

$$s_2 - s_1 = c_p \ln \left(\frac{T_2}{T_1} \right) - R \ln \left(\frac{P_2}{P_1} \right)$$

Since c_p is assumed to be constant.

8-76C No. The entropy of an ideal gas depends on the pressure as well as the temperature.

8-77C Setting $\Delta s = 0$ gives

$$c_p \ln \left(\frac{T_2}{T_1} \right) - R \ln \left(\frac{P_2}{P_1} \right) = 0 \longrightarrow \ln \left(\frac{T_2}{T_1} \right) = \frac{R}{c_p} \ln \left(\frac{P_2}{P_1} \right) \longrightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{R/c_p}$$

But

$$\frac{R}{c_p} = \frac{c_p - c_v}{c_p} = 1 - \frac{1}{k} = \frac{k-1}{k} \quad \text{since} \quad k = c_p / c_v. \quad \text{Thus,} \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{(k-1)/k}$$

8-78C The P_r and v_r are called relative pressure and relative specific volume, respectively. They are derived for isentropic processes of ideal gases, and thus their use is limited to isentropic processes only.

8-79C The entropy of a gas *can* change during an isothermal process since entropy of an ideal gas depends on the pressure as well as the temperature.

8-80C The entropy change relations of an ideal gas simplify to

$$\Delta s = c_p \ln(T_2/T_1) \text{ for a constant pressure process}$$

and $\Delta s = c_v \ln(T_2/T_1)$ for a constant volume process.

Noting that $c_p > c_v$, the entropy change will be larger for a constant pressure process.

8-81 The entropy difference between the two states of oxygen is to be determined.

Assumptions Oxygen is an ideal gas with constant specific heats.

Properties The specific heat of oxygen at the average temperature of $(39+337)/2=188^\circ\text{C}=461\text{ K}$ is $c_p = 0.960\text{ kJ/kg}\cdot\text{K}$ (Table A-2b).

Analysis From the entropy change relation of an ideal gas,

$$\Delta s_{\text{oxygen}} = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} = (0.960\text{ kJ/kg}\cdot\text{K}) \ln \frac{(337+273)\text{K}}{(39+273)\text{K}} - 0 = \mathbf{0.6436\text{ kJ/kg}\cdot\text{K}}$$

since the pressure is same at the initial and final states.

8-82 The entropy changes of helium and nitrogen is to be compared for the same initial and final states.

Assumptions Helium and nitrogen are ideal gases with constant specific heats.

Properties The properties of helium are $c_p = 5.1926\text{ kJ/kg}\cdot\text{K}$, $R = 2.0769\text{ kJ/kg}\cdot\text{K}$ (Table A-2a). The specific heat of nitrogen at the average temperature of $(427+27)/2=227^\circ\text{C}=500\text{ K}$ is $c_p = 1.056\text{ kJ/kg}\cdot\text{K}$ (Table A-2b). The gas constant of nitrogen is $R = 0.2968\text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis From the entropy change relation of an ideal gas,

$$\begin{aligned} \Delta s_{\text{He}} &= c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \\ &= (5.1926\text{ kJ/kg}\cdot\text{K}) \ln \frac{(27+273)\text{K}}{(427+273)\text{K}} - (2.0769\text{ kJ/kg}\cdot\text{K}) \ln \frac{200\text{ kPa}}{2000\text{ kPa}} \\ &= \mathbf{0.3826\text{ kJ/kg}\cdot\text{K}} \\ \Delta s_{\text{N}_2} &= c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \\ &= (1.056\text{ kJ/kg}\cdot\text{K}) \ln \frac{(27+273)\text{K}}{(427+273)\text{K}} - (0.2968\text{ kJ/kg}\cdot\text{K}) \ln \frac{200\text{ kPa}}{2000\text{ kPa}} \\ &= \mathbf{-0.2113\text{ kJ/kg}\cdot\text{K}} \end{aligned}$$

Hence, helium undergoes the largest change in entropy.

8-83E The entropy change of air during an expansion process is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat of air at the average temperature of $(500+50)/2=275^\circ\text{F}$ is $c_p = 0.243 \text{ Btu/lbm}\cdot\text{R}$ (Table A-2Eb). The gas constant of air is $R = 0.06855 \text{ Btu/lbm}\cdot\text{R}$ (Table A-2Ea).

Analysis From the entropy change relation of an ideal gas,

$$\begin{aligned}\Delta s_{\text{air}} &= c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \\ &= (0.243 \text{ Btu/lbm}\cdot\text{R}) \ln \frac{(50+460)\text{R}}{(500+460)\text{R}} - (0.06855 \text{ Btu/lbm}\cdot\text{R}) \ln \frac{100 \text{ psia}}{200 \text{ psia}} \\ &= \mathbf{-0.1062 \text{ Btu/lbm}\cdot\text{R}}\end{aligned}$$

8-84 The final temperature of air when it is expanded isentropically is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air at an anticipated average temperature of 550 K is $k = 1.381$ (Table A-2b).

Analysis From the isentropic relation of an ideal gas under constant specific heat assumption,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (477 + 273 \text{ K}) \left(\frac{100 \text{ kPa}}{1000 \text{ kPa}} \right)^{0.381/1.381} = \mathbf{397 \text{ K}}$$

Discussion The average air temperature is $(750+397.4)/2=573.7 \text{ K}$, which is sufficiently close to the assumed average temperature of 550 K.

8-85E The final temperature of air when it is expanded isentropically is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air at an anticipated average temperature of 300°F is $k = 1.394$ (Table A-2Eb).

Analysis From the isentropic relation of an ideal gas under constant specific heat assumption,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (500 + 460 \text{ R}) \left(\frac{20 \text{ psia}}{100 \text{ psia}} \right)^{0.394/1.394} = \mathbf{609 \text{ R}}$$

Discussion The average air temperature is $(960+609)/2=785 \text{ R}=325^\circ\text{F}$, which is sufficiently close to the assumed average temperature of 300°F .

8-86 The final temperatures of helium and nitrogen when they are compressed isentropically are to be compared.

Assumptions Helium and nitrogen are ideal gases with constant specific heats.

Properties The specific heat ratios of helium and nitrogen at room temperature are $k = 1.667$ and $k = 1.4$, respectively (Table A-2a).

Analysis From the isentropic relation of an ideal gas under constant specific heat assumption,

$$T_{2,\text{He}} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (298 \text{ K}) \left(\frac{1000 \text{ kPa}}{100 \text{ kPa}} \right)^{0.667/1.667} = \mathbf{749 \text{ K}}$$

$$T_{2,\text{N}_2} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (298 \text{ K}) \left(\frac{1000 \text{ kPa}}{100 \text{ kPa}} \right)^{0.4/1.4} = \mathbf{575 \text{ K}}$$

Hence, the helium produces the greater temperature when it is compressed.

8-87 The final temperatures of neon and air when they are expanded isentropically are to be compared.

Assumptions Neon and air are ideal gases with constant specific heats.

Properties The specific heat ratios of neon and air at room temperature are $k = 1.667$ and $k = 1.4$, respectively (Table A-2a).

Analysis From the isentropic relation of an ideal gas under constant specific heat assumption,

$$T_{2,\text{Ne}} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (773 \text{ K}) \left(\frac{100 \text{ kPa}}{1000 \text{ kPa}} \right)^{0.667/1.667} = \mathbf{308 \text{ K}}$$

$$T_{2,\text{air}} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (773 \text{ K}) \left(\frac{100 \text{ kPa}}{1000 \text{ kPa}} \right)^{0.4/1.4} = \mathbf{400 \text{ K}}$$

Hence, the neon produces the smaller temperature when it is expanded.

8-88 An insulated cylinder initially contains air at a specified state. A resistance heater inside the cylinder is turned on, and air is heated for 15 min at constant pressure. The entropy change of air during this process is to be determined for the cases of constant and variable specific heats.

Assumptions At specified conditions, air can be treated as an ideal gas.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

Analysis The mass of the air and the electrical work done during this process are

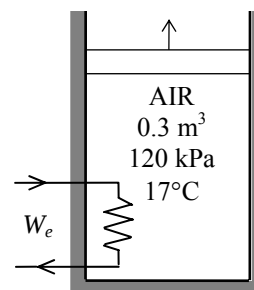
$$m = \frac{P_1 V_1}{RT_1} = \frac{(120 \text{ kPa})(0.3 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})} = 0.4325 \text{ kg}$$

$$W_{e,\text{in}} = \dot{W}_{e,\text{in}} \Delta t = (0.2 \text{ kJ/s})(15 \times 60 \text{ s}) = 180 \text{ kJ}$$

The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,\text{in}} - W_{b,\text{out}} = \Delta U \longrightarrow W_{e,\text{in}} = m(h_2 - h_1) \cong c_p(T_2 - T_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.

(a) Using a constant c_p value at the anticipated average temperature of 450 K, the final temperature becomes

$$\text{Thus, } T_2 = T_1 + \frac{W_{e,\text{in}}}{mc_p} = 290 \text{ K} + \frac{180 \text{ kJ}}{(0.4325 \text{ kg})(1.02 \text{ kJ/kg}\cdot\text{K})} = 698 \text{ K}$$

Then the entropy change becomes

$$\Delta S_{\text{sys}} = m(s_2 - s_1) = m \left(c_{p,\text{avg}} \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) = mc_{p,\text{avg}} \ln \frac{T_2}{T_1}$$

$$= (0.4325 \text{ kg})(1.020 \text{ kJ/kg}\cdot\text{K}) \ln \left(\frac{698 \text{ K}}{290 \text{ K}} \right) = \mathbf{0.387 \text{ kJ/K}}$$

(b) Assuming variable specific heats,

$$W_{e,\text{in}} = m(h_2 - h_1) \longrightarrow h_2 = h_1 + \frac{W_{e,\text{in}}}{m} = 290.16 \text{ kJ/kg} + \frac{180 \text{ kJ}}{0.4325 \text{ kg}} = 706.34 \text{ kJ/kg}$$

From the air table (Table A-21, we read $s_2^\circ = 2.5628 \text{ kJ/kg}\cdot\text{K}$ corresponding to this h_2 value. Then,

$$\Delta S_{\text{sys}} = m \left(s_2^\circ - s_1^\circ + R \ln \frac{P_2}{P_1} \right) = m(s_2^\circ - s_1^\circ) = (0.4325 \text{ kg})(2.5628 - 1.66802) \text{ kJ/kg}\cdot\text{K} = \mathbf{0.387 \text{ kJ/K}}$$

8-89 A cylinder contains N_2 gas at a specified pressure and temperature. The gas is compressed polytropically until the volume is reduced by half. The entropy change of nitrogen during this process is to be determined.

Assumptions 1 At specified conditions, N_2 can be treated as an ideal gas. 2 Nitrogen has constant specific heats at room temperature.

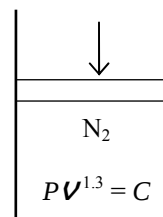
Properties The gas constant of nitrogen is $R = 0.297 \text{ kJ/kg}\cdot\text{K}$ (Table A-1). The constant volume specific heat of nitrogen at room temperature is $c_v = 0.743 \text{ kJ/kg}\cdot\text{K}$ (Table A-2).

Analysis From the polytropic relation,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1} \longrightarrow T_2 = T_1 \left(\frac{v_1}{v_2}\right)^{n-1} = (300 \text{ K})(2)^{1.3-1} = 369.3 \text{ K}$$

Then the entropy change of nitrogen becomes

$$\begin{aligned} \Delta S_{N_2} &= m \left(c_{v,\text{avg}} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1} \right) \\ &= (1.2 \text{ kg}) \left((0.743 \text{ kJ/kg}\cdot\text{K}) \ln \frac{369.3 \text{ K}}{300 \text{ K}} + (0.297 \text{ kJ/kg}\cdot\text{K}) \ln(0.5) \right) = -0.0617 \text{ kJ/K} \end{aligned}$$



8-90 EES Problem 8-89 is reconsidered. The effect of varying the polytropic exponent from 1 to 1.4 on the entropy change of the nitrogen is to be investigated, and the processes are to be shown on a common P - v diagram.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

Function BoundWork(P[1],V[1],P[2],V[2],n)

"This function returns the Boundary Work for the polytropic process. This function is required since the expression for boundary work depends on whether $n=1$ or $n \neq 1$ "

```
If n<>1 then
  BoundWork:=(P[2]*V[2]-P[1]*V[1])/(1-n)"Use Equation 3-22 when n=1"
else
  BoundWork:= P[1]*V[1]*ln(V[2]/V[1]) "Use Equation 3-20 when n=1"
endif
end
```

n=1

P[1] = 120 [kPa]

T[1] = 27 [C]

m = 1.2 [kg]

V[2]=V[1]/2

Gas\$='N2'

MM=molarmass(Gas\$)

R=R_u/MM

R_u=8.314 [kJ/kmol-K]

"System: The gas enclosed in the piston-cylinder device."

"Process: Polytropic expansion or compression, $P \cdot V^n = C$ "

P[1]*V[1]=m*R*(T[1]+273)

P[2]*V[2]^n=P[1]*V[1]^n

W_b = BoundWork(P[1],V[1],P[2],V[2],n)

"Find the temperature at state 2 from the pressure and specific volume."

T[2]=temperature(gas\$,P=P[2],v=V[2]/m)

"The entropy at states 1 and 2 is:"

s[1]=entropy(gas\$,P=P[1],v=V[1]/m)

s[2]=entropy(gas\$,P=P[2],v=V[2]/m)

DELTAS=m*(s[2] - s[1])

"Remove the {} to generate the P-v plot data"

{Nsteps = 10

VP[1]=V[1]

PP[1]=P[1]

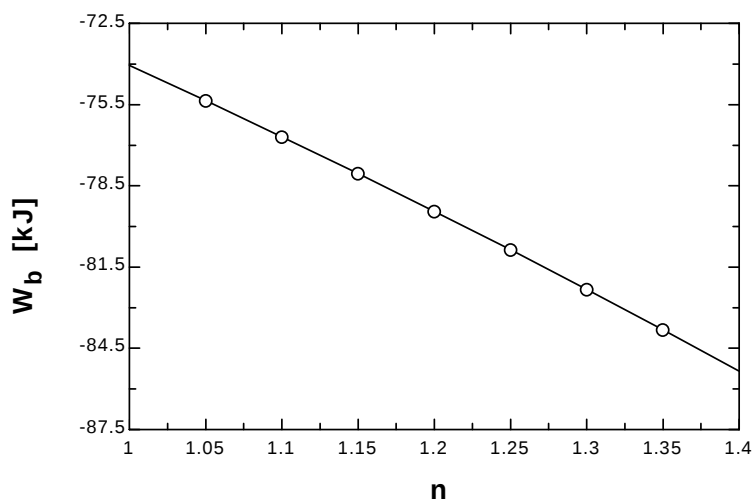
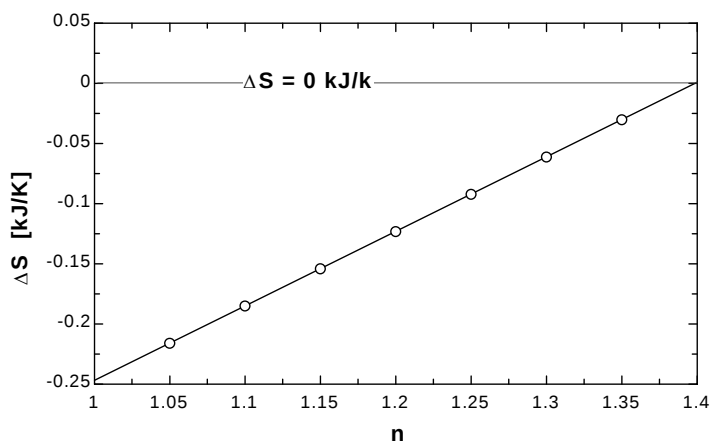
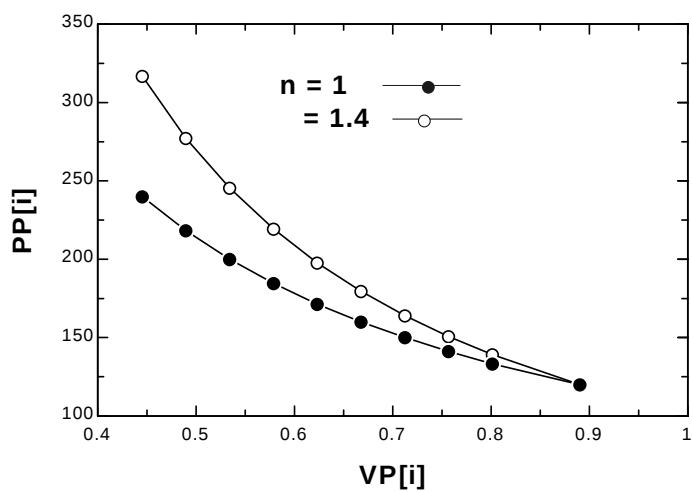
Duplicate i=2,Nsteps

VP[i]=V[1]-i*(V[1]-V[2])/Nsteps

PP[i]=P[1]*(V[1]/VP[i])^n

END }

ΔS [kJ/kg]	n	W_b [kJ]
-0.2469	1	-74.06
-0.2159	1.05	-75.36
-0.1849	1.1	-76.69
-0.1539	1.15	-78.05
-0.1229	1.2	-79.44
-0.09191	1.25	-80.86
-0.06095	1.3	-82.32
-0.02999	1.35	-83.82
0.0009849	1.4	-85.34



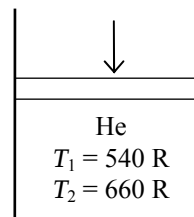
8-91E A fixed mass of helium undergoes a process from one specified state to another specified state. The entropy change of helium is to be determined for the cases of reversible and irreversible processes.

Assumptions 1 At specified conditions, helium can be treated as an ideal gas. **2** Helium has constant specific heats at room temperature.

Properties The gas constant of helium is $R = 0.4961 \text{ Btu/lbm} \cdot \text{R}$ (Table A-1E). The constant volume specific heat of helium is $c_v = 0.753 \text{ Btu/lbm} \cdot \text{R}$ (Table A-2E).

Analysis From the ideal-gas entropy change relation,

$$\begin{aligned}\Delta S_{\text{He}} &= m \left(c_{v,\text{ave}} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1} \right) \\ &= (15 \text{ lbm}) \left((0.753 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{660 \text{ R}}{540 \text{ R}} + (0.4961 \text{ Btu/lbm} \cdot \text{R}) \ln \left(\frac{10 \text{ ft}^3/\text{lbm}}{50 \text{ ft}^3/\text{lbm}} \right) \right) \\ &= \mathbf{-9.71 \text{ Btu/R}}\end{aligned}$$



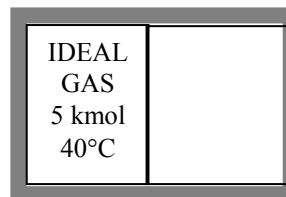
The entropy change will be the same for both cases.

8-92 One side of a partitioned insulated rigid tank contains an ideal gas at a specified temperature and pressure while the other side is evacuated. The partition is removed, and the gas fills the entire tank. The total entropy change during this process is to be determined.

Assumptions The gas in the tank is given to be an ideal gas, and thus ideal gas relations apply.

Analysis Taking the entire rigid tank as the system, the energy balance can be expressed as

$$\begin{aligned}\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\ 0 = \Delta U &= m(u_2 - u_1) \\ u_2 &= u_1 \\ T_2 &= T_1\end{aligned}$$



since $u = u(T)$ for an ideal gas. Then the entropy change of the gas becomes

$$\begin{aligned}\Delta S &= N \left(\bar{c}_{v,\text{avg}} \ln \frac{T_2}{T_1} + R_u \ln \frac{v_2}{v_1} \right) = NR_u \ln \frac{v_2}{v_1} \\ &= (5 \text{ kmol})(8.314 \text{ kJ/kmol} \cdot \text{K}) \ln(2) \\ &= \mathbf{28.81 \text{ kJ/K}}\end{aligned}$$

This also represents the **total entropy change** since the tank does not contain anything else, and there are no interactions with the surroundings.

8-93 Air is compressed in a piston-cylinder device in a reversible and adiabatic manner. The final temperature and the work are to be determined for the cases of constant and variable specific heats.

Assumptions 1 At specified conditions, air can be treated as an ideal gas. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-1). The specific heat ratio of air at low to moderately high temperatures is $k = 1.4$ (Table A-2).

Analysis (a) Assuming constant specific heats, the ideal gas isentropic relations give

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (290 \text{ K}) \left(\frac{800 \text{ kPa}}{100 \text{ kPa}} \right)^{0.4/1.4} = 525.3 \text{ K}$$

Then,

$$T_{\text{avg}} = (290 + 525.3)/2 = 407.7 \text{ K} \longrightarrow c_{v,\text{avg}} = 0.727 \text{ kJ/kg}\cdot\text{K}$$

We take the air in the cylinder as the system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{\text{in}} = \Delta U = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

Thus,

$$w_{\text{in}} = c_{v,\text{avg}}(T_2 - T_1) = (0.727 \text{ kJ/kg}\cdot\text{K})(525.3 - 290) \text{ K} = 171.1 \text{ kJ/kg}$$

(b) Assuming variable specific heats, the final temperature can be determined using the relative pressure data (Table A-21),

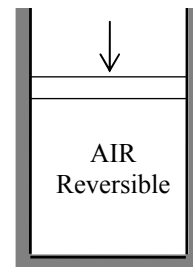
$$T_1 = 290 \text{ K} \longrightarrow \begin{aligned} P_{r_1} &= 1.2311 \\ u_1 &= 206.91 \text{ kJ/kg} \end{aligned}$$

and

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{800 \text{ kPa}}{100 \text{ kPa}} (1.2311) = 9.849 \longrightarrow \begin{aligned} T_2 &= 522.4 \text{ K} \\ u_2 &= 376.16 \text{ kJ/kg} \end{aligned}$$

Then the work input becomes

$$w_{\text{in}} = u_2 - u_1 = (376.16 - 206.91) \text{ kJ/kg} = 169.25 \text{ kJ/kg}$$



8-94 EES Problem 8-93 is reconsidered. The work done and final temperature during the compression process are to be calculated and plotted as functions of the final pressure for the two cases as the final pressure varies from 100 kPa to 800 kPa.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```

Procedure ConstPropSol(P_1,T_1,P_2,Gas$:Work_in_ConstProp,T2_ConstProp)
C_P=SPECHEAT(Gas$,T=27)
MM=MOLARMASS(Gas$)
R_u=8.314 [kJ/kmol-K]
R=R_u/MM
C_V = C_P - R
k = C_P/C_V
T2= (T_1+273)*(P_2/P_1)^((k-1)/k)
T2_ConstProp=T2-273 "[C]"
DELTau = C_v*(T2-(T_1+273))
Work_in_ConstProp = DELTAu
End

```

"Knowns:"

P_1 = 100 [kPa]

T_1 = 17 [C]

P_2 = 800 [kPa]

"Analysis: "

" Treat the piston-cylinder as a closed system, with no heat transfer in, neglect changes in KE and PE of the air. The process is reversible and adiabatic thus isentropic."

"The isentropic work is determined from:"

e_in - e_out = DELTAe_sys

e_out = 0 [kJ/kg]

e_in = Work_in

DELTAe_sys = (u_2 - u_1)

u_1 = INTENERGY(air,T=T_1)

v_1 = volume(air,P=P_1,T=T_1)

s_1 = entropy(air,P=P_1,T=T_1)

" The process is reversible and adiabatic or isentropic.

Then P_2 and s_2 specify state 2."

s_2 = s_1

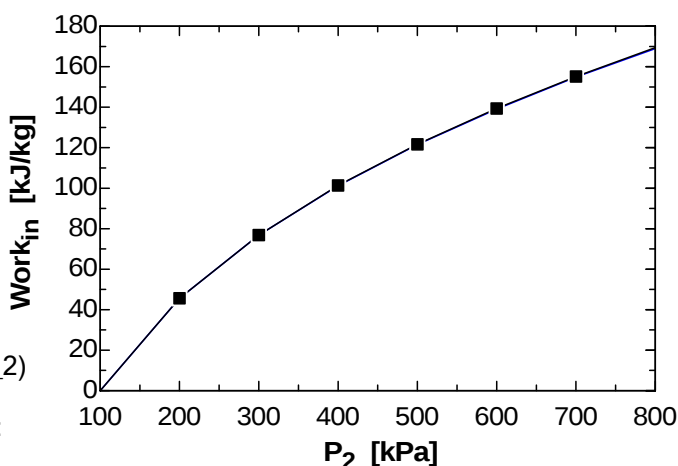
u_2 = INTENERGY(air,P=P_2,s=s_2)

T_2_isen=temperature(air,P=P_2,s=s_2)

Gas\$ = 'air'

Call ConstPropSol(P_1,T_1,P_2,Gas\$:

Work_in_ConstProp,T2_ConstProp)



P ₂ [kPa]	Work _{in} [kJ/kg]	Work _{in,ConstProp} [kJ/kg]
100	0	0
200	45.63	45.6
300	76.84	76.77
400	101.3	101.2
500	121.7	121.5
600	139.4	139.1
700	155.2	154.8
800	169.3	168.9

8-95 An insulated rigid tank contains argon gas at a specified pressure and temperature. A valve is opened, and argon escapes until the pressure drops to a specified value. The final mass in the tank is to be determined.

Assumptions 1 At specified conditions, argon can be treated as an ideal gas. 2 The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

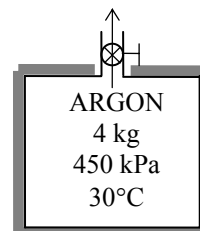
Properties The specific heat ratio of argon is $k = 1.667$ (Table A-2).

Analysis From the ideal gas isentropic relations,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (303 \text{ K}) \left(\frac{200 \text{ kPa}}{450 \text{ kPa}} \right)^{0.667/1.667} = 219.0 \text{ K}$$

The final mass in the tank is determined from the ideal gas relation,

$$\frac{P_1 V}{P_2 V} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow m_2 = \frac{P_2 T_1}{P_1 T_2} m_1 = \frac{(200 \text{ kPa})(303 \text{ K})}{(450 \text{ kPa})(219 \text{ K})} (4 \text{ kg}) = \mathbf{2.46 \text{ kg}}$$



8-96 EES Problem 8-95 is reconsidered. The effect of the final pressure on the final mass in the tank is to be investigated as the pressure varies from 450 kPa to 150 kPa, and the results are to be plotted.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"Knowns:"

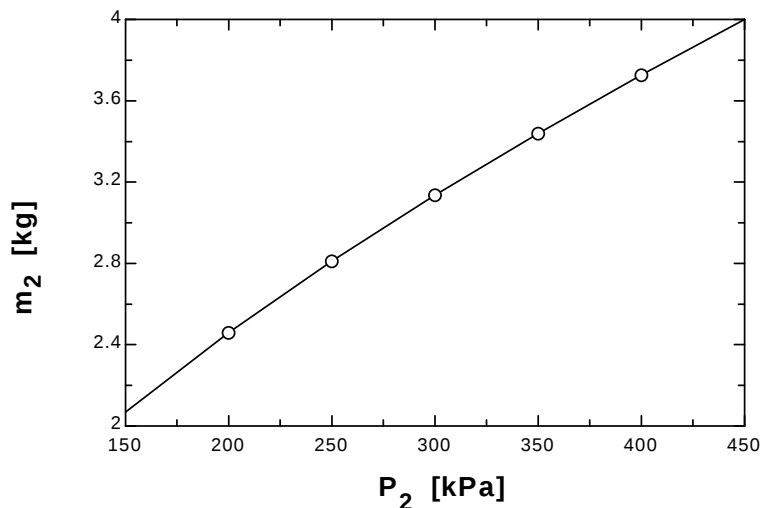
C_P = 0.5203 "[kJ/kg-K]"
 C_V = 0.3122 "[kJ/kg-K]"
 R = 0.2081 "[kPa-m^3/kg-K]"
 P_1 = 450 "[kPa]"
 T_1 = 30 "[C]"
 m_1 = 4 "[kg]"
 P_2 = 150 "[kPa]"

"Analysis:"

We assume the mass that stays in the tank undergoes an isentropic expansion process. This allows us to determine the final temperature of that gas at the final pressure in the tank by using the isentropic relation:"

$k = C_P / C_V$
 $T_2 = ((T_1 + 273) * (P_2 / P_1)^{((k-1)/k)} - 273) "[C]"$
 $V_2 = V_1$
 $P_1 * V_1 = m_1 * R * (T_1 + 273)$
 $P_2 * V_2 = m_2 * R * (T_2 + 273)$

m ₂ [kg]	P ₂ [kPa]
2.069	150
2.459	200
2.811	250
3.136	300
3.44	350
3.727	400
4	450



8-97E Air is accelerated in an adiabatic nozzle. Disregarding irreversibilities, the exit velocity of air is to be determined.

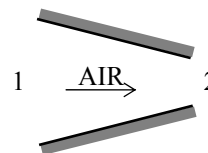
Assumptions 1 Air is an ideal gas with variable specific heats. 2 The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply. 2 The nozzle operates steadily.

Analysis Assuming variable specific heats, the inlet and exit properties are determined to be

$$T_1 = 1000 \text{ R} \longrightarrow \begin{matrix} P_{r_1} = 12.30 \\ h_1 = 240.98 \text{ Btu/lbm} \end{matrix}$$

and

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{12 \text{ psia}}{60 \text{ psia}} (12.30) = 2.46 \longrightarrow \begin{matrix} T_2 = 635.9 \text{ R} \\ h_2 = 152.11 \text{ Btu/lbm} \end{matrix}$$



We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = 0$$

Therefore,

$$V_2 = \sqrt{2(h_1 - h_2) + V_1^2} = \sqrt{2(240.98 - 152.11) \text{ Btu/lbm} \left(\frac{25,037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right) + (200 \text{ ft/s})^2}$$

$$= 2119 \text{ ft/s}$$

8-98E Air is expanded in an isentropic turbine. The exit temperature of the air and the power produced are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 The process is isentropic (i.e., reversible-adiabatic). 3 Air is an ideal gas with constant specific heats.

Properties The properties of air at an anticipated average temperature of 600°F are $c_p = 0.250$ Btu/lbm·R and $k = 1.377$ (Table A-2Eb). The gas constant of air is $R = 0.3704$ psia·ft³/lbm·R (Table A-1E).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right) + \dot{W}_{\text{out}}$$

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m} \left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} \right) \\ &= \dot{m} \left(c_p (T_1 - T_2) + \frac{V_1^2 - V_2^2}{2} \right) \end{aligned}$$

The exit temperature of the air for this isentropic process is

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (900 + 460 \text{ R}) \left(\frac{15 \text{ psia}}{150 \text{ psia}} \right)^{0.377/1.377} = \mathbf{724 \text{ R}}$$

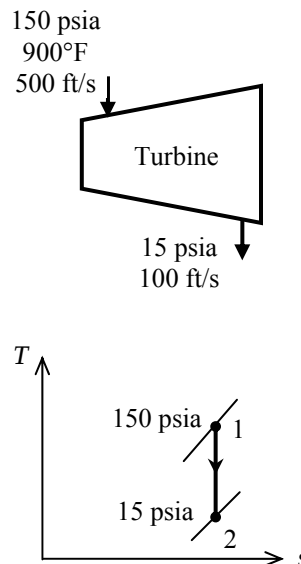
The specific volume of air at the inlet and the mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(900 + 460 \text{ R})}{150 \text{ psia}} = 3.358 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{(0.5 \text{ ft}^2)(500 \text{ ft/s})}{3.358 \text{ ft}^3/\text{lbm}} = 74.45 \text{ lbm/s}$$

Substituting into the energy balance equation gives

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m} \left(c_p (T_1 - T_2) + \frac{V_1^2 - V_2^2}{2} \right) \\ &= (74.45 \text{ lbm/s}) \left[(0.250 \text{ Btu/lbm} \cdot \text{R})(1360 - 724) \text{ R} + \frac{(500 \text{ ft/s})^2 - (100 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) \right] \\ &= 12,194 \text{ Btu/s} \left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}} \right) = \mathbf{17,250 \text{ hp}} \end{aligned}$$



8-99 Nitrogen is compressed in an adiabatic compressor. The minimum work input is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** The process is adiabatic, and thus there is no heat transfer. **3** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at an anticipated average temperature of 400 K are $c_p = 1.044$ kJ/kg·K and $k = 1.397$ (Table A-2b).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

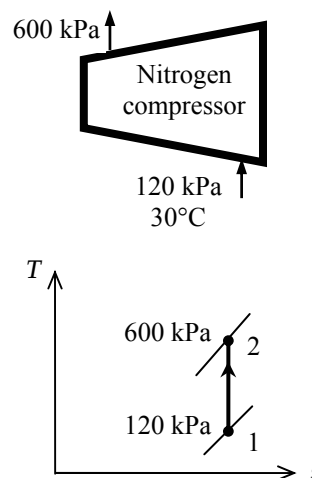
$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

For the minimum work input to the compressor, the process must be reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (303 \text{ K}) \left(\frac{600 \text{ kPa}}{120 \text{ kPa}} \right)^{0.397/1.397} = 479 \text{ K}$$

Substituting into the energy balance equation gives

$$w_{\text{in}} = h_2 - h_1 = c_p (T_2 - T_1) = (1.044 \text{ kJ/kg} \cdot \text{K})(479 - 303) \text{ K} = \mathbf{184 \text{ kJ/kg}}$$



8-100 Oxygen is expanded in an adiabatic nozzle. The maximum velocity at the exit is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** The process is adiabatic, and thus there is no heat transfer. **3** Oxygen is an ideal gas with constant specific heats.

Properties The properties of oxygen at room temperature are $c_p = 0.918 \text{ kJ/kg}\cdot\text{K}$ and $k = 1.395$ (Table A-2a).

Analysis For the maximum velocity at the exit, the process must be reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (363 \text{ K}) \left(\frac{120 \text{ kPa}}{300 \text{ kPa}} \right)^{0.395/1.395} = 280.0 \text{ K}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\text{no}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

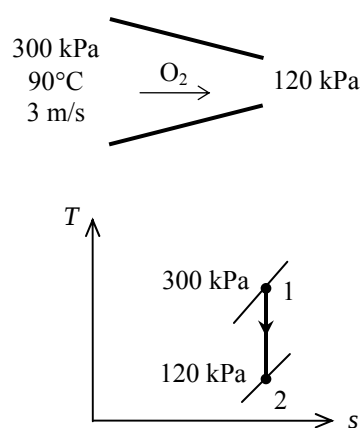
Solving for the exit velocity,

$$V_2 = \left[V_1^2 + 2(h_1 - h_2) \right]^{0.5}$$

$$= \left[V_1^2 + 2c_p (T_1 - T_2) \right]^{0.5}$$

$$= \left[(3 \text{ m/s})^2 + 2(0.918 \text{ kJ/kg}\cdot\text{K})(363 - 280) \text{ K} \left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}} \right) \right]^{0.5}$$

$$= \mathbf{390 \text{ m/s}}$$



8-101 Air is expanded in an adiabatic nozzle by a polytropic process. The temperature and velocity at the exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ and $k = 1.4$ (Table A-2a).

Analysis For the polytropic process of an ideal gas, $P\upsilon^n = \text{Constant}$, and the exit temperature is given by

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(n-1)/n} = (373 \text{ K}) \left(\frac{200 \text{ kPa}}{700 \text{ kPa}} \right)^{0.3/1.3} = \mathbf{279 \text{ K}}$$

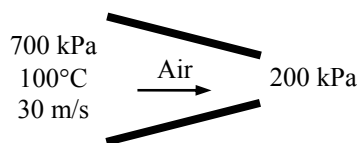
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$



Solving for the exit velocity,

$$\begin{aligned} V_2 &= \left[V_1^2 + 2(h_1 - h_2) \right]^{0.5} \\ &= \left[V_1^2 + 2c_p(T_1 - T_2) \right]^{0.5} \\ &= \left[(30 \text{ m/s})^2 + 2(1.005 \text{ kJ/kg}\cdot\text{K})(373 - 279) \text{ K} \left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}} \right) \right]^{0.5} \\ &= \mathbf{436 \text{ m/s}} \end{aligned}$$

8-102 Air is expanded in an adiabatic nozzle by a polytropic process. The temperature and velocity at the exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ and $k = 1.4$ (Table A-2a).

Analysis For the polytropic process of an ideal gas, $P\nu^n = \text{Constant}$, and the exit temperature is given by

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(n-1)/n} = (373 \text{ K}) \left(\frac{200 \text{ kPa}}{700 \text{ kPa}} \right)^{0.2/1.2} = \mathbf{303 \text{ K}}$$

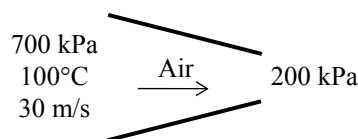
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$



Solving for the exit velocity,

$$\begin{aligned} V_2 &= \left[V_1^2 + 2(h_1 - h_2) \right]^{0.5} \\ &= \left[V_1^2 + 2c_p(T_1 - T_2) \right]^{0.5} \\ &= \left[(30 \text{ m/s})^2 + 2(1.005 \text{ kJ/kg}\cdot\text{K})(373 - 303) \text{ K} \left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}} \right) \right]^{0.5} \\ &= \mathbf{376 \text{ m/s}} \end{aligned}$$

8-103E Air is charged to an initially evacuated container from a supply line. The minimum temperature of the air in the container after it is filled is to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The specific heat of air at room temperature is $c_p = 0.240 \text{ Btu/lbm}\cdot\text{R}$ (Table A-2Ea).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and entropy balances for this uniform-flow system can be expressed as

Mass balance:

$$\begin{aligned} m_{\text{in}} - m_{\text{out}} &= \Delta m_{\text{system}} \\ m_i &= m_2 - m_1 \\ m_i &= m_2 \end{aligned}$$

Entropy balance:

$$\begin{aligned} m_2 s_2 - m_1 s_1 + m_e s_e - m_i s_i &\geq 0 \\ m_2 s_2 - m_i s_i &\geq 0 \end{aligned}$$

Combining the two balances,

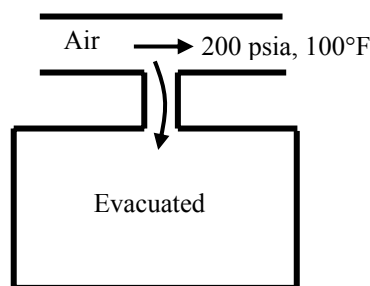
$$\begin{aligned} m_2 s_2 - m_2 s_i &\geq 0 \\ s_2 - s_i &\geq 0 \end{aligned}$$

The minimum temperature will result when the equal sign applies. Noting that $P_2 = P_i$, we have

$$s_2 - s_i = c_p \ln \frac{T_2}{T_i} - R \ln \frac{P_2}{P_i} = 0 \longrightarrow c_p \ln \frac{T_2}{T_i} = 0$$

Then,

$$T_2 = T_i = \mathbf{100^\circ\text{F}}$$



8-104 A container filled with liquid water is placed in a room and heat transfer takes place between the container and the air in the room until the thermal equilibrium is established. The final temperature, the amount of heat transfer between the water and the air, and the entropy generation are to be determined.

Assumptions 1 Kinetic and potential energy changes are negligible. 2 Air is an ideal gas with constant specific heats. 3 The room is well-sealed and there is no heat transfer from the room to the surroundings. 4 Sea level atmospheric pressure is assumed. $P = 101.3 \text{ kPa}$.

Properties The properties of air at room temperature are $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$, $c_p = 1.005 \text{ kJ/kg} \cdot \text{K}$, $c_v = 0.718 \text{ kJ/kg} \cdot \text{K}$. The specific heat of water at room temperature is $c_w = 4.18 \text{ kJ/kg} \cdot \text{K}$ (Tables A-2, A-3).

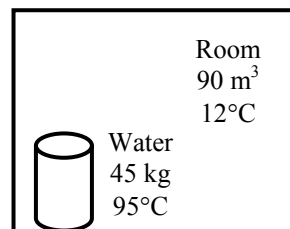
Analysis (a) The mass of the air in the room is

$$m_a = \frac{P\mathcal{V}}{RT_{a1}} = \frac{(101.3 \text{ kPa})(90 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(12 + 273 \text{ K})} = 111.5 \text{ kg}$$

An energy balance on the system that consists of the water in the container and the air in the room gives the final equilibrium temperature

$$0 = m_w c_w (T_2 - T_{w1}) + m_a c_v (T_2 - T_{a1})$$

$$0 = (45 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K})(T_2 - 95) + (111.5 \text{ kg})(0.718 \text{ kJ/kg} \cdot \text{K})(T_2 - 12) \longrightarrow T_2 = \mathbf{70.2^\circ\text{C}}$$



(b) The heat transfer to the air is

$$Q = m_a c_v (T_2 - T_{a1}) = (111.5 \text{ kg})(0.718 \text{ kJ/kg} \cdot \text{K})(70.2 - 12) = \mathbf{4660 \text{ kJ}}$$

(c) The entropy generation associated with this heat transfer process may be obtained by calculating total entropy change, which is the sum of the entropy changes of water and the air.

$$\Delta S_w = m_w c_w \ln \frac{T_2}{T_{w1}} = (45 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{(70.2 + 273) \text{ K}}{(95 + 273) \text{ K}} = -13.11 \text{ kJ/K}$$

$$P_2 = \frac{m_a R T_2}{\mathcal{V}} = \frac{(111.5 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(70.2 + 273 \text{ K})}{(90 \text{ m}^3)} = 122 \text{ kPa}$$

$$\begin{aligned} \Delta S_a &= m_a \left(c_p \ln \frac{T_2}{T_{a1}} - R \ln \frac{P_2}{P_1} \right) \\ &= (111.5 \text{ kg}) \left[(1.005 \text{ kJ/kg} \cdot \text{K}) \ln \frac{(70.2 + 273) \text{ K}}{(12 + 273) \text{ K}} - (0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{122 \text{ kPa}}{101.3 \text{ kPa}} \right] = 14.88 \text{ kJ/K} \end{aligned}$$

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_w + \Delta S_a = -13.11 + 14.88 = \mathbf{1.77 \text{ kJ/K}}$$

8-105 Air is accelerated in an isentropic nozzle. The maximum velocity at the exit is to be determined.

Assumptions 1 Air is an ideal gas with constant specific heats. **2** The nozzle operates steadily.

Properties The properties of air at room temperature are $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$, $k = 1.4$ (Table A-2a).

Analysis The exit temperature is determined from ideal gas isentropic relation to be,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (400 + 273 \text{ K}) \left(\frac{100 \text{ kPa}}{800 \text{ kPa}} \right)^{0.4/1.4} = 371.5 \text{ K}$$

We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

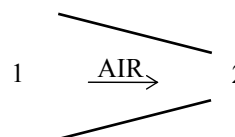
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{Steady}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - 0}{2}$$

$$0 = c_p(T_2 - T_1) + \frac{V_2^2}{2}$$



Therefore,

$$V_2 = \sqrt{2c_p(T_1 - T_2)} = \sqrt{2(1.005 \text{ kJ/kg}\cdot\text{K})(673 - 371.5)\text{K} \left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}} \right)} = \mathbf{778.5 \text{ m/s}}$$

8-106 An ideal gas is compressed in an isentropic compressor. 10% of gas is compressed to 400 kPa and 90% is compressed to 600 kPa. The compression process is to be sketched, and the exit temperatures at the two exits, and the mass flow rate into the compressor are to be determined.

Assumptions 1 The compressor operates steadily. 2 The process is reversible-adiabatic (isentropic)

Properties The properties of ideal gas are given to be $c_p = 1.1$ kJ/kg.K and $c_v = 0.8$ kJ/kg.K.

Analysis (b) The specific heat ratio of the gas is

$$k = \frac{c_p}{c_v} = \frac{1.1}{0.8} = 1.375$$

The exit temperatures are determined from ideal gas isentropic relations to be,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (27 + 273 \text{ K}) \left(\frac{400 \text{ kPa}}{100 \text{ kPa}} \right)^{0.375/1.375} = 437.8 \text{ K}$$

$$T_3 = T_1 \left(\frac{P_3}{P_1} \right)^{(k-1)/k} = (27 + 273 \text{ K}) \left(\frac{600 \text{ kPa}}{100 \text{ kPa}} \right)^{0.375/1.375} = 489.0 \text{ K}$$

(c) A mass balance on the control volume gives

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3$$

where

$$\dot{m}_2 = 0.1\dot{m}_1$$

$$\dot{m}_3 = 0.9\dot{m}_1$$

We take the compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{steady}}{=} 0$$

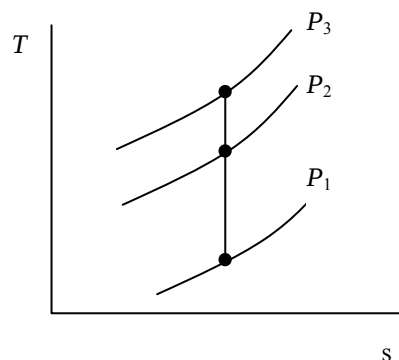
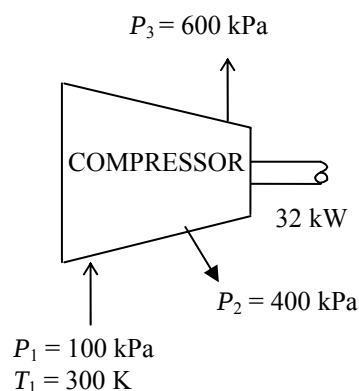
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{W}_{\text{in}} = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

$$\dot{m}_1 c_p T_1 + \dot{W}_{\text{in}} = 0.1 \dot{m}_1 c_p T_2 + 0.9 \dot{m}_1 c_p T_3$$

Solving for the inlet mass flow rate, we obtain

$$\begin{aligned} \dot{m}_1 &= \frac{\dot{W}_{\text{in}}}{c_p [0.1(T_2 - T_1) + 0.9(T_3 - T_1)]} \\ &= \frac{32 \text{ kW}}{(1.1 \text{ kJ/kg} \cdot \text{K}) [0.1(437.8 - 300) + 0.9(489.0 - 300)]} \\ &= \mathbf{0.158 \text{ kg/s}} \end{aligned}$$



8-107 Air contained in a constant-volume tank is cooled to ambient temperature. The entropy changes of the air and the universe due to this process are to be determined and the process is to be sketched on a T-s diagram.

Assumptions 1 Air is an ideal gas with constant specific heats.

Properties The specific heat of air at room temperature is $c_v = 0.718 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis (a) The entropy change of air is determined from

$$\begin{aligned}\Delta S_{\text{air}} &= mc_v \ln \frac{T_2}{T_1} \\ &= (5 \text{ kg})(0.718 \text{ kJ/kg}\cdot\text{K}) \ln \frac{(27 + 273) \text{ K}}{(327 + 273) \text{ K}} \\ &= \mathbf{-2.488 \text{ kJ/K}}\end{aligned}$$

(b) An energy balance on the system gives

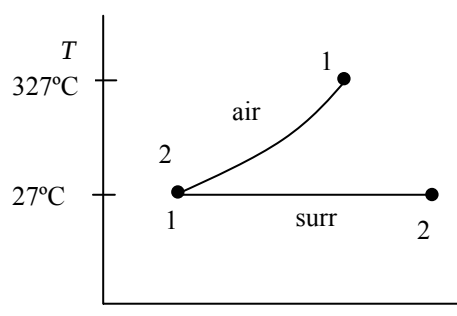
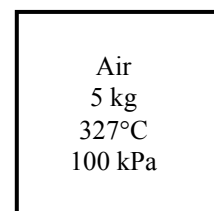
$$\begin{aligned}Q_{\text{out}} &= mc_v (T_1 - T_2) \\ &= (5 \text{ kg})(0.718 \text{ kJ/kg}\cdot\text{K})(327 - 27) \\ &= 1077 \text{ kJ}\end{aligned}$$

The entropy change of the surroundings is

$$\Delta S_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{1077 \text{ kJ}}{300 \text{ K}} = 3.59 \text{ kJ/K}$$

The entropy change of universe due to this process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{air}} + \Delta S_{\text{surr}} = -2.488 + 3.59 = \mathbf{1.10 \text{ kJ/K}}$$



Reversible Steady-Flow Work

8-108C The work associated with steady-flow devices is proportional to the specific volume of the gas. Cooling a gas during compression will reduce its specific volume, and thus the power consumed by the compressor.

8-109C Cooling the steam as it expands in a turbine will reduce its specific volume, and thus the work output of the turbine. Therefore, this is not a good proposal.

8-110C We would not support this proposal since the steady-flow work input to the pump is proportional to the specific volume of the liquid, and cooling will not affect the specific volume of a liquid significantly.

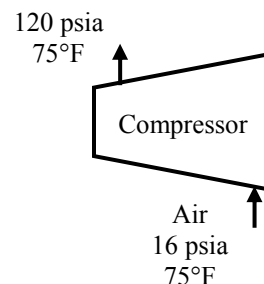
8-111 Air is compressed isothermally in a reversible steady-flow device. The work required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer associated with the process. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.06855 \text{ Btu/lbm} \cdot \text{R}$ (Table A-1E).

Analysis Substituting the ideal gas equation of state into the reversible steady-flow work expression gives

$$\begin{aligned} w_{\text{in}} &= \int_1^2 v dP = RT \int_1^2 \frac{dP}{P} = RT \ln \frac{P_2}{P_1} \\ &= (0.06855 \text{ Btu/lbm} \cdot \text{R})(75 + 460 \text{ K}) \ln \left(\frac{120 \text{ psia}}{16 \text{ psia}} \right) \\ &= \mathbf{73.9 \text{ Btu/lbm}} \end{aligned}$$



8-112 Saturated water vapor is compressed in a reversible steady-flow device. The work required is to be determined.

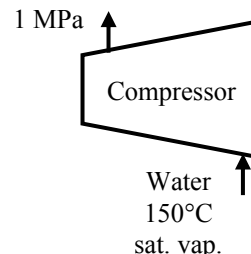
Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer associated with the process. **3** Kinetic and potential energy changes are negligible.

Analysis The properties of water at the inlet state are

$$\left. \begin{array}{l} T_1 = 150^\circ\text{C} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} P_1 = 476.16 \text{ kPa} \\ \nu_1 = 0.39248 \text{ m}^3/\text{kg} \end{array} \quad (\text{Table A - 4})$$

Noting that the specific volume remains constant, the reversible steady-flow work expression gives

$$\begin{aligned} w_{\text{in}} &= \int_1^2 \nu dP = \nu_1 (P_2 - P_1) \\ &= (0.39248 \text{ m}^3/\text{kg})(1000 - 476.16) \text{ kPa} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= \mathbf{205.6 \text{ kJ/kg}} \end{aligned}$$

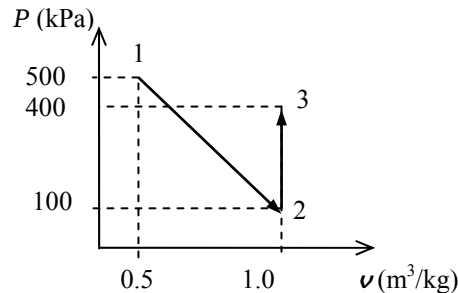


8-113 The work produced for the process 1-3 shown in the figure is to be determined.

Assumptions Kinetic and potential energy changes are negligible.

Analysis The work integral represents the area to the left of the reversible process line. Then,

$$\begin{aligned} w_{\text{in},1-3} &= \int_1^2 \nu dP + \int_2^3 \nu dP \\ &= \frac{\nu_1 + \nu_2}{2} (P_2 - P_1) + \nu_2 (P_3 - P_2) \\ &= \frac{(0.5 + 1.0) \text{ m}^3/\text{kg}}{2} (100 - 500) \text{ kPa} + (1.0 \text{ m}^3/\text{kg})(400 - 100) \text{ kPa} \\ &= \mathbf{0 \text{ kJ/kg}} \end{aligned}$$

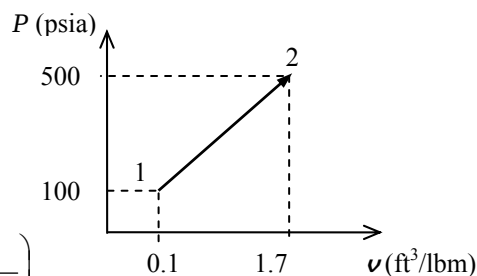


8-114E The work produced for the process 1-2 shown in the figure is to be determined.

Assumptions Kinetic and potential energy changes are negligible.

Analysis The work integral represents the area to the left of the reversible process line. Then,

$$\begin{aligned} w_{\text{in},1-2} &= \int_1^2 \nu dP \\ &= \frac{\nu_1 + \nu_2}{2} (P_2 - P_1) \\ &= \frac{(0.1 + 1.7) \text{ ft}^3/\text{lbm}}{2} (500 - 100) \text{ psia} \left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3} \right) \\ &= \mathbf{66.6 \text{ Btu/lbm}} \end{aligned}$$



8-115 Liquid water is to be pumped by a 25-kW pump at a specified rate. The highest pressure the water can be pumped to is to be determined.

Assumptions 1 Liquid water is an incompressible substance. 2 Kinetic and potential energy changes are negligible. 3 The process is assumed to be reversible since we will determine the limiting case.

Properties The specific volume of liquid water is given to be $\nu_1 = 0.001 \text{ m}^3/\text{kg}$.

Analysis The highest pressure the liquid can have at the pump exit can be determined from the reversible steady-flow work relation for a liquid,

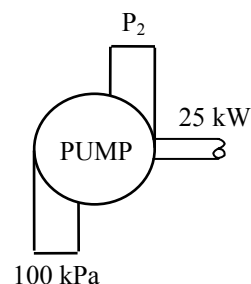
$$\dot{W}_{\text{in}} = \dot{m} \left(\int_1^2 \nu dP + \Delta ke^{\text{e}} + \Delta pe^{\text{e}} \right) = \dot{m} \nu_1 (P_2 - P_1)$$

Thus,

$$25 \text{ kJ/s} = (5 \text{ kg/s})(0.001 \text{ m}^3/\text{kg})(P_2 - 100) \text{ kPa} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right)$$

It yields

$$P_2 = \mathbf{5100 \text{ kPa}}$$



8-116 A steam power plant operates between the pressure limits of 10 MPa and 20 kPa. The ratio of the turbine work to the pump work is to be determined.

Assumptions 1 Liquid water is an incompressible substance. 2 Kinetic and potential energy changes are negligible. 3 The process is reversible. 4 The pump and the turbine are adiabatic.

Properties The specific volume of saturated liquid water at 20 kPa is $\nu_1 = \nu_f @ 20 \text{ kPa} = 0.001017 \text{ m}^3/\text{kg}$ (Table A-5).

Analysis Both the compression and expansion processes are reversible and adiabatic, and thus isentropic, $s_1 = s_2$ and $s_3 = s_4$. Then the properties of the steam are

$$\left. \begin{array}{l} P_4 = 20 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} h_4 = h_g @ 20 \text{ kPa} = 2608.9 \text{ kJ/kg} \\ s_4 = s_g @ 20 \text{ kPa} = 7.9073 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_3 = 10 \text{ MPa} \\ s_3 = s_4 \end{array} \right\} h_3 = 4707.2 \text{ kJ/kg}$$

Also, $\nu_1 = \nu_f @ 20 \text{ kPa} = 0.001017 \text{ m}^3/\text{kg}$.

The work output to this isentropic turbine is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\varphi 0} (\text{steady})}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_3 = \dot{m}h_4 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_3 - h_4)$$

Substituting,

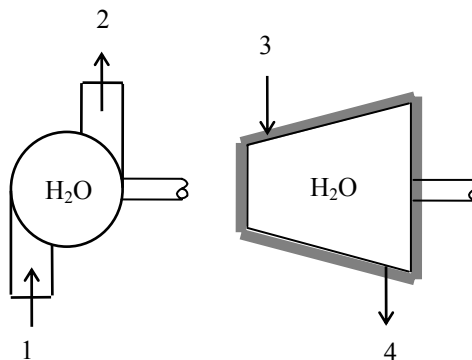
$$w_{\text{turb, out}} = h_3 - h_4 = 4707.2 - 2608.9 = 2098.3 \text{ kJ/kg}$$

The pump work input is determined from the steady-flow work relation to be

$$\begin{aligned} w_{\text{pump, in}} &= \int_1^2 \nu dP + \Delta ke^{\varphi 0} + \Delta pe^{\varphi 0} = \nu_1 (P_2 - P_1) \\ &= (0.001017 \text{ m}^3/\text{kg})(10,000 - 20) \text{ kPa} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= 10.15 \text{ kJ/kg} \end{aligned}$$

Thus,

$$\frac{w_{\text{turb, out}}}{w_{\text{pump, in}}} = \frac{2098.3}{10.15} = \mathbf{206.7}$$



8-117 EES Problem 8-116 is reconsidered. The effect of the quality of the steam at the turbine exit on the net work output is to be investigated as the quality is varied from 0.5 to 1.0, and the net work output is to be plotted as a function of this quality.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

"System: control volume for the pump and turbine"

"Property relation: Steam functions"

"Process: For Pump and Turbine: Steady state, steady flow, adiabatic, reversible or isentropic"

"Since we don't know the mass, we write the conservation of energy per unit mass."

"Conservation of mass: $\dot{m}_1 = \dot{m}_2$ "

"Knowns:"

WorkFluid\$ = 'Steam_IAPWS'

P[1] = 20 [kPa]

x[1] = 0

P[2] = 10000 [kPa]

x[4] = 1.0

"Pump Analysis:"

T[1]=temperature(WorkFluid\$,P=P[1],x=0)

v[1]=volume(workFluid\$,P=P[1],x=0)

h[1]=enthalpy(WorkFluid\$,P=P[1],x=0)

s[1]=entropy(WorkFluid\$,P=P[1],x=0)

s[2] = s[1]

h[2]=enthalpy(WorkFluid\$,P=P[2],s=s[2])

T[2]=temperature(WorkFluid\$,P=P[2],s=s[2])

"The Volume function has the same form for an ideal gas as for a real fluid."

v[2]=volume(WorkFluid\$,T=T[2],p=P[2])

"Conservation of Energy - SSSF energy balance for pump"

" -- neglect the change in potential energy, no heat transfer:"

$h[1] + W_{\text{pump}} = h[2]$

"Also the work of pump can be obtained from the incompressible fluid, steady-flow result:"

$W_{\text{pump_incomp}} = v[1](P[2] - P[1])$

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential energy, no heat transfer:"

P[4] = P[1]

P[3] = P[2]

h[4]=enthalpy(WorkFluid\$,P=P[4],x=x[4])

s[4]=entropy(WorkFluid\$,P=P[4],x=x[4])

T[4]=temperature(WorkFluid\$,P=P[4],x=x[4])

s[3] = s[4]

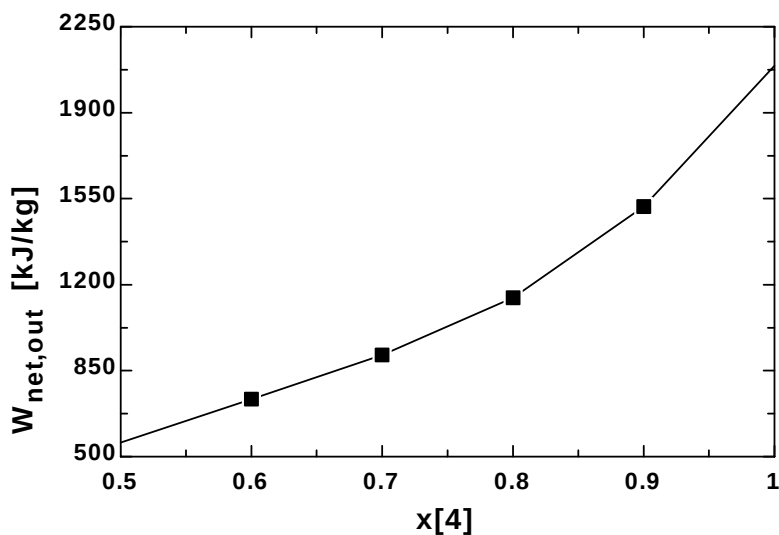
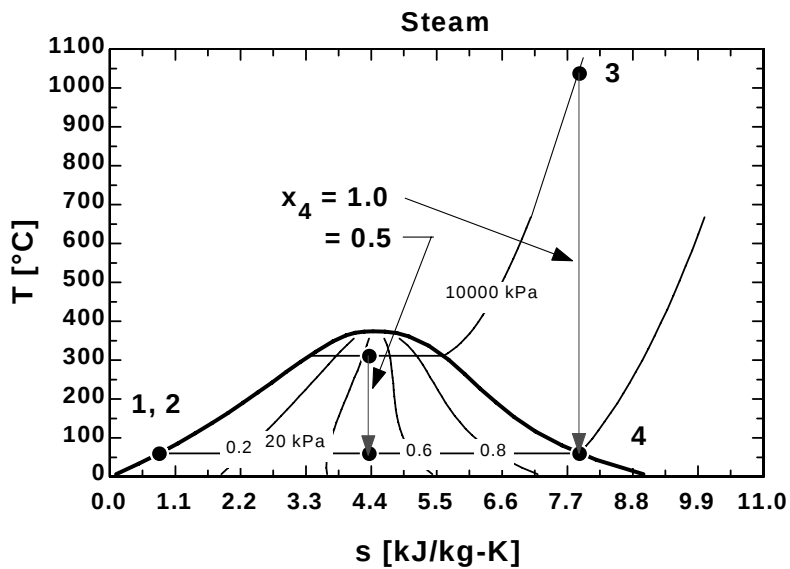
h[3]=enthalpy(WorkFluid\$,P=P[3],s=s[3])

T[3]=temperature(WorkFluid\$,P=P[3],s=s[3])

$h[3] = h[4] + W_{\text{turb}}$

$W_{\text{net_out}} = W_{\text{turb}} - W_{\text{pump}}$

$W_{\text{net,out}}$ [kJ/kg]	W_{pump} [kJ/kg]	$W_{\text{pump,incomp}}$ [kJ/kg]	W_{turb} [kJ/kg]	x_4
557.1	10.13	10.15	567.3	0.5
734.7	10.13	10.15	744.8	0.6
913.6	10.13	10.15	923.7	0.7
1146	10.13	10.15	1156	0.8
1516	10.13	10.15	1527	0.9
2088	10.13	10.15	2098	1



8-118 Liquid water is pumped by a 70-kW pump to a specified pressure at a specified level. The highest possible mass flow rate of water is to be determined.

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic energy changes are negligible, but potential energy changes may be significant. **3** The process is assumed to be reversible since we will determine the limiting case.

Properties The specific volume of liquid water is given to be $\nu_1 = 0.001 \text{ m}^3/\text{kg}$.

Analysis The highest mass flow rate will be realized when the entire process is reversible. Thus it is determined from the reversible steady-flow work relation for a liquid,

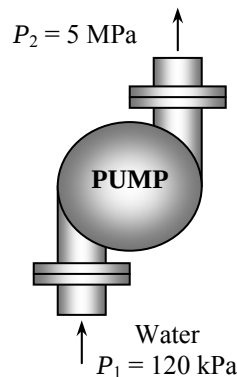
$$\dot{W}_{\text{in}} = \dot{m} \left(\int_1^2 \nu \, dP + \Delta ke^{\text{flow}} + \Delta pe \right) = \dot{m} \{ \nu (P_2 - P_1) + g(z_2 - z_1) \}$$

Thus,

$$7 \text{ kJ/s} = \dot{m} \left\{ (0.001 \text{ m}^3/\text{kg})(5000 - 120) \text{ kPa} \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) + (9.8 \text{ m/s}^2)(10 \text{ m}) \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right\}$$

It yields

$$\dot{m} = \mathbf{1.41 \text{ kg/s}}$$



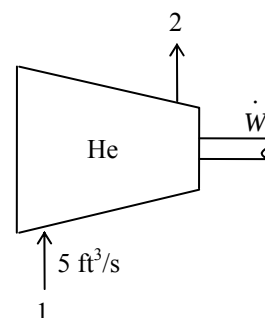
8-119E Helium gas is compressed from a specified state to a specified pressure at a specified rate. The power input to the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

Assumptions 1 Helium is an ideal gas with constant specific heats. 2 The process is reversible. 3 Kinetic and potential energy changes are negligible.

Properties The gas constant of helium is $R = 2.6805 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$
 $= 0.4961 \text{ Btu}/\text{lbm} \cdot \text{R}$. The specific heat ratio of helium is $k = 1.667$ (Table A-2E).

Analysis The mass flow rate of helium is

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(14 \text{ psia})(5 \text{ ft}^3/\text{s})}{(2.6805 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(530 \text{ R})} = 0.0493 \text{ lbm/s}$$



(a) Isentropic compression with $k = 1.667$:

$$\begin{aligned} \dot{W}_{\text{comp, in}} &= \dot{m} \frac{kRT_1}{k-1} \left\{ \left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right\} \\ &= (0.0493 \text{ lbm/s}) \frac{(1.667)(0.4961 \text{ Btu}/\text{lbm} \cdot \text{R})(530 \text{ R})}{1.667 - 1} \left\{ \left(\frac{120 \text{ psia}}{14 \text{ psia}} \right)^{0.667/1.667} - 1 \right\} \\ &= 44.11 \text{ Btu/s} \\ &= \mathbf{62.4 \text{ hp}} \quad \text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s} \end{aligned}$$

(b) Polytropic compression with $n = 1.2$:

$$\begin{aligned} \dot{W}_{\text{comp, in}} &= \dot{m} \frac{nRT_1}{n-1} \left\{ \left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right\} \\ &= (0.0493 \text{ lbm/s}) \frac{(1.2)(0.4961 \text{ Btu}/\text{lbm} \cdot \text{R})(530 \text{ R})}{1.2 - 1} \left\{ \left(\frac{120 \text{ psia}}{14 \text{ psia}} \right)^{0.2/1.2} - 1 \right\} \\ &= 33.47 \text{ Btu/s} \\ &= \mathbf{47.3 \text{ hp}} \quad \text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s} \end{aligned}$$

(c) Isothermal compression:

$$\dot{W}_{\text{comp, in}} = \dot{m}RT \ln \frac{P_2}{P_1} = (0.0493 \text{ lbm/s})(0.4961 \text{ Btu}/\text{lbm} \cdot \text{R})(530 \text{ R}) \ln \frac{120 \text{ psia}}{14 \text{ psia}} = 27.83 \text{ Btu/s} = \mathbf{39.4 \text{ hp}}$$

(d) Ideal two-stage compression with intercooling ($n = 1.2$): In this case, the pressure ratio across each stage is the same, and its value is determined from

$$P_x = \sqrt{P_1 P_2} = \sqrt{(14 \text{ psia})(120 \text{ psia})} = 41.0 \text{ psia}$$

The compressor work across each stage is also the same, thus total compressor work is twice the compression work for a single stage:

$$\begin{aligned} \dot{W}_{\text{comp, in}} &= 2\dot{m}w_{\text{comp, I}} = 2\dot{m} \frac{nRT_1}{n-1} \left\{ \left(\frac{P_x}{P_1} \right)^{(n-1)/n} - 1 \right\} \\ &= 2(0.0493 \text{ lbm/s}) \frac{(1.2)(0.4961 \text{ Btu}/\text{lbm} \cdot \text{R})(530 \text{ R})}{1.2 - 1} \left\{ \left(\frac{41 \text{ psia}}{14 \text{ psia}} \right)^{0.2/1.2} - 1 \right\} \\ &= 30.52 \text{ Btu/s} \\ &= \mathbf{43.2 \text{ hp}} \quad \text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s} \end{aligned}$$

8-120E EES Problem 8-119E is reconsidered. The work of compression and entropy change of the helium is to be evaluated and plotted as functions of the polytropic exponent as it varies from 1 to 1.667.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
Procedure FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out_2stagePoly)
```

```
If n =1 then
```

```
T2=T1
```

```
W_dot_comp_polytropic= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp) "[hp]"
```

```
W_dot_comp_2stagePoly = W_dot_comp_polytropic "[hp]"
```

```
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s) "[Btu/s]"
```

```
Q_dot_Out_2stagePoly = Q_dot_Out_polytropic*convert(hp,Btu/s) "[Btu/s]"
```

```
Else
```

```
C_P = k*R/(k-1) "[Btu/lbm-R]"
```

```
T2=(T1+460)*((P2/P1)^((n+1)/n)-460)"[F]"
```

```
W_dot_comp_polytropic = m_dot*n*R*(T1+460)/(n-1)*((P2/P1)^((n-1)/n) -
```

```
1)*convert(Btu/s,hp)"[hp]"
```

```
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s)+m_dot*C_P*(T1-T2)"[Btu/s]"
```

```
Px=(P1*P2)^0.5
```

```
T2x=(T1+460)*((Px/P1)^((n+1)/n)-460)"[F]"
```

```
W_dot_comp_2stagePoly = 2*m_dot*n*R*(T1+460)/(n-1)*((Px/P1)^((n-1)/n) -
```

```
1)*convert(Btu/s,hp)"[hp]"
```

```
Q_dot_Out_2stagePoly=W_dot_comp_2stagePoly*convert(hp,Btu/s)+2*m_dot*C_P*(T1-
T2x)"[Btu/s]"
```

```
endif
```

```
END
```

```
R=0.4961[Btu/lbm-R]
```

```
k=1.667
```

```
n=1.2
```

```
P1=14 [psia]
```

```
T1=70 [F]
```

```
P2=120 [psia]
```

```
V_dot = 5 [ft^3/s]
```

```
P1*V_dot=m_dot*R*(T1+460)*convert(Btu,psia-ft^3)
```

```
W_dot_comp_isentropic = m_dot*k*R*(T1+460)/(k-1)*((P2/P1)^((k-1)/k) -
```

```
1)*convert(Btu/s,hp)"[hp]"
```

```
Q_dot_Out_isentropic = 0"[Btu/s]"
```

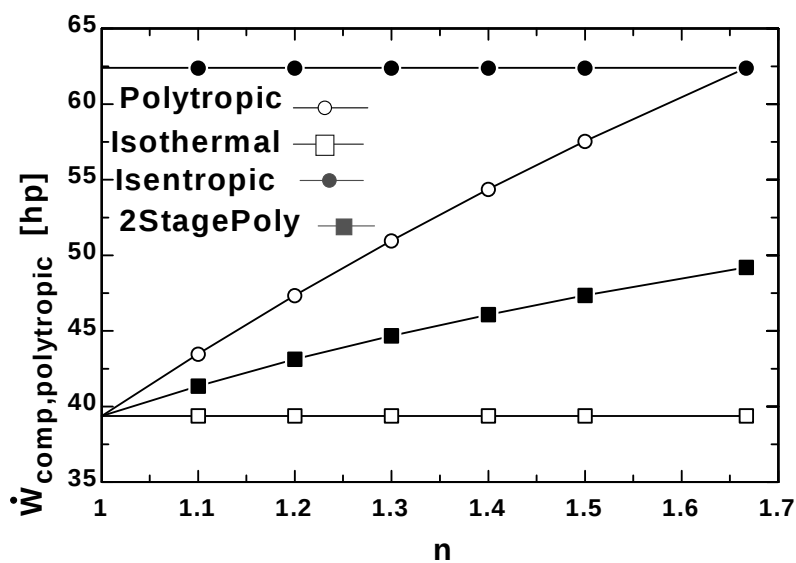
```
Call FuncPoly(m_dot,k, R,
```

```
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out_2stagePoly)
```

```
W_dot_comp_isothermal= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp)"[hp]"
```

```
Q_dot_Out_isothermal = W_dot_comp_isothermal*convert(hp,Btu/s)"[Btu/s]"
```

n	$W_{\text{comp2StagePoly}}$ [hp]	$W_{\text{compisentropic}}$ [hp]	$W_{\text{compisothermal}}$ [hp]	$W_{\text{comppolytropic}}$ [hp]
1	39.37	62.4	39.37	39.37
1.1	41.36	62.4	39.37	43.48
1.2	43.12	62.4	39.37	47.35
1.3	44.68	62.4	39.37	50.97
1.4	46.09	62.4	39.37	54.36
1.5	47.35	62.4	39.37	57.54
1.667	49.19	62.4	39.37	62.4



8-121 Water mist is to be sprayed into the air stream in the compressor to cool the air as the water evaporates and to reduce the compression power. The reduction in the exit temperature of the compressed air and the compressor power saved are to be determined.

Assumptions 1 Air is an ideal gas with variable specific heats. 2 The process is reversible. 3 Kinetic and potential energy changes are negligible. 3 Air is compressed isentropically. 4 Water vaporizes completely before leaving the compressor. 4 Air properties can be used for the air-vapor mixture.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-1). The specific heat ratio of air is $k = 1.4$. The inlet enthalpies of water and air are (Tables A-4 and A-17)

$$h_{w1} = h_{f@20^\circ\text{C}} = 83.29 \text{ kJ/kg}, h_{fg@20^\circ\text{C}} = 2453.9 \text{ kJ/kg} \text{ and } h_{a1} = h_{@300\text{ K}} = 300.19 \text{ kJ/kg}$$

Analysis In the case of isentropic operation (thus no cooling or water spray), the exit temperature and the power input to the compressor are

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{(k-1)/k} \rightarrow T_2 = (300 \text{ K}) \left(\frac{1200 \text{ kPa}}{100 \text{ kPa}} \right)^{(1.4-1)/1.4} = 610.2 \text{ K}$$

$$\begin{aligned} \dot{W}_{\text{comp, in}} &= \dot{m} \frac{kRT_1}{k-1} \left\{ \left(P_2/P_1 \right)^{(k-1)/k} - 1 \right\} \\ &= (2.1 \text{ kg/s}) \frac{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.4-1} \left\{ (1200 \text{ kPa}/100 \text{ kPa})^{0.4/1.4} - 1 \right\} = 654.3 \text{ kW} \end{aligned}$$

When water is sprayed, we first need to check the accuracy of the assumption that the water vaporizes completely in the compressor. In the limiting case, the compression will be isothermal at the compressor inlet temperature, and the water will be a saturated vapor. To avoid the complexity of dealing with two fluid streams and a gas mixture, we disregard water in the air stream (other than the mass flow rate), and assume air is cooled by an amount equal to the enthalpy change of water.

The rate of heat absorption of water as it evaporates at the inlet temperature completely is

$$\dot{Q}_{\text{cooling, max}} = \dot{m}_w h_{fg @ 20^\circ\text{C}} = (0.2 \text{ kg/s})(2453.9 \text{ kJ/kg}) = 490.8 \text{ kW}$$

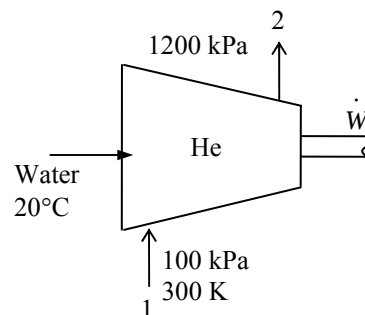
The minimum power input to the compressor is

$$\dot{W}_{\text{comp, in, min}} = \dot{m} R T \ln \frac{P_2}{P_1} = (2.1 \text{ kg/s})(0.287 \text{ kJ/kg}\cdot\text{K})(300 \text{ K}) \ln \left(\frac{1200 \text{ kPa}}{100 \text{ kPa}} \right) = 449.3 \text{ kW}$$

This corresponds to maximum cooling from the air since, at constant temperature, $\Delta h = 0$ and thus $\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 449.3 \text{ kW}$, which is close to 490.8 kW. Therefore, the assumption that all the water vaporizes is approximately valid. Then the reduction in required power input due to water spray becomes

$$\Delta \dot{W}_{\text{comp, in}} = \dot{W}_{\text{comp, isentropic}} - \dot{W}_{\text{comp, isothermal}} = 654.3 - 449.3 = \mathbf{205 \text{ kW}}$$

Discussion (can be ignored): At constant temperature, $\Delta h = 0$ and thus $\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 449.3 \text{ kW}$ corresponds to maximum cooling from the air, which is less than 490.8 kW. Therefore, the assumption that all the water vaporizes is only roughly valid. As an alternative, we can assume the compression process to be polytropic and the water to be a saturated vapor at the compressor exit temperature, and disregard the remaining liquid. But in this case there is not a unique solution, and we will have to select either the amount of water or the exit temperature or the polytropic exponent to obtain a solution. Of course we can also tabulate the results for different cases, and then make a selection.



Sample Analysis: We take the compressor exit temperature to be $T_2 = 200^\circ\text{C} = 473\text{ K}$. Then,

$$h_{w2} = h_{g@200^\circ\text{C}} = 2792.0\text{ kJ/kg and } h_{a2} = h_{@473\text{ K}} = 475.3\text{ kJ/kg}$$

Then,

$$\begin{aligned}\frac{T_2}{T_1} &= \left(\frac{P_2}{P_1}\right)^{(n-1)/n} \rightarrow \frac{473\text{ K}}{300\text{ K}} = \left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(n-1)/n} \rightarrow n = 1.224 \\ \dot{W}_{\text{comp},in} &= \dot{m} \frac{nRT_1}{n-1} \left\{ \left(P_2/P_1\right)^{(n-1)/n} - 1 \right\} = \dot{m} \frac{nR}{n-1} (T_2 - T_1) \\ &= (2.1\text{ kg/s}) \frac{(1.224)(0.287\text{ kJ/kg} \cdot \text{K})}{1.224 - 1} (473 - 300)\text{K} = 570\text{ kW}\end{aligned}$$

Energy balance:

$$\begin{aligned}\dot{W}_{\text{comp},in} - \dot{Q}_{\text{out}} &= \dot{m}(h_2 - h_1) \rightarrow \dot{Q}_{\text{out}} = \dot{W}_{\text{comp},in} - \dot{m}(h_2 - h_1) \\ &= 569.7\text{ kW} - (2.1\text{ kg/s})(475.3 - 300.19) = 202.0\text{ kW}\end{aligned}$$

Noting that this heat is absorbed by water, the rate at which water evaporates in the compressor becomes

$$\dot{Q}_{\text{out},\text{air}} = \dot{Q}_{\text{in},\text{water}} = \dot{m}_w(h_{w2} - h_{w1}) \longrightarrow \dot{m}_w = \frac{\dot{Q}_{\text{in},\text{water}}}{h_{w2} - h_{w1}} = \frac{202.0\text{ kJ/s}}{(2792.0 - 83.29)\text{ kJ/kg}} = 0.0746\text{ kg/s}$$

Then the reductions in the exit temperature and compressor power input become

$$\begin{aligned}\Delta T_2 &= T_{2,\text{isentropic}} - T_{2,\text{water cooled}} = 610.2 - 473 = \mathbf{137.2^\circ\text{C}} \\ \Delta \dot{W}_{\text{comp},in} &= \dot{W}_{\text{comp},\text{isentropic}} - \dot{W}_{\text{comp},\text{water cooled}} = 654.3 - 570 = \mathbf{84.3\text{ kW}}\end{aligned}$$

Note that selecting a different compressor exit temperature T_2 will result in different values.

8-122 A water-injected compressor is used in a gas turbine power plant. It is claimed that the power output of a gas turbine will increase when water is injected into the compressor because of the increase in the mass flow rate of the gas (air + water vapor) through the turbine. This, however, is **not necessarily right** since the compressed air in this case enters the combustor at a low temperature, and thus it absorbs much more heat. In fact, the cooling effect will most likely dominate and cause the cyclic efficiency to drop.

Isentropic Efficiencies of Steady-Flow Devices

8-123C The ideal process for all three devices is the reversible adiabatic (i.e., isentropic) process. The adiabatic efficiencies of these devices are defined as

$$\eta_T = \frac{\text{actual work output}}{\text{isentropic work output}}, \quad \eta_C = \frac{\text{isentropic work input}}{\text{actual work input}}, \quad \text{and} \quad \eta_N = \frac{\text{actual exit kinetic energy}}{\text{isentropic exit kinetic energy}}$$

8-124C No, because the isentropic process is not the model or ideal process for compressors that are cooled intentionally.

8-125C Yes. Because the entropy of the fluid must increase during an actual adiabatic process as a result of irreversibilities. Therefore, the actual exit state has to be on the right-hand side of the isentropic exit state

8-126 Saturated steam is compressed in an adiabatic process with an isentropic efficiency of 0.90. The work required is to be determined.

Assumptions 1 Kinetic and potential energy changes are negligible. 2 The device is adiabatic and thus heat transfer is negligible.

Analysis We take the steam as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{in}} = \Delta U = m(u_2 - u_1)$$

From the steam tables (Tables A-5 and A-6),

$$\left. \begin{array}{l} P_1 = 100 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} u_1 = u_{g@100 \text{ kPa}} = 2505.6 \text{ kJ/kg} \\ s_1 = s_{g@100 \text{ kPa}} = 7.3589 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array} \right\} u_{2s} = 2903.0 \text{ kJ/kg}$$

The work input during the isentropic process is

$$W_{s,\text{in}} = m(u_{2s} - u_1) = (100 \text{ kg})(2903.0 - 2505.6) \text{ kJ/kg} = 39,740 \text{ kJ}$$

The actual work input is then

$$W_{a,\text{in}} = \frac{W_{s,\text{in}}}{\eta_C} = \frac{39,740 \text{ kJ}}{0.90} = \mathbf{44,160 \text{ kJ}}$$

8-127E R-134a is expanded in an adiabatic process with an isentropic efficiency of 0.95. The final volume is to be determined.

Assumptions 1 Kinetic and potential energy changes are negligible. 2 The device is adiabatic and thus heat transfer is negligible.

Analysis We take the R-134a as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

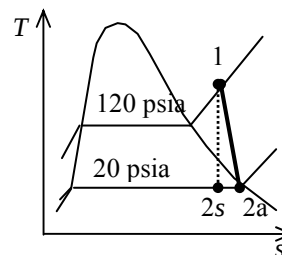
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1)$$

From the R-134a tables (Tables A-11E through A-13E),

$$\left. \begin{array}{l} P_1 = 120 \text{ psia} \\ T_1 = 100^\circ\text{F} \end{array} \right\} \begin{array}{l} u_1 = 108.48 \text{ Btu/lbm} \\ s_1 = 0.22362 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left. \begin{array}{l} P_2 = 20 \text{ psia} \\ s_{2s} = s_1 \end{array} \right\} \begin{array}{l} x_{2s} = \frac{s_{2s} - s_f}{s_{fg}} = \frac{0.22362 - 0.02605}{0.19962} = 0.9897 \\ u_{2s} = u_f + x_{2s} u_{fg} = 11.401 + (0.9897)(82.898) = 93.445 \text{ Btu/lbm} \end{array}$$



The actual work input is

$$w_{a,\text{out}} = \eta_T w_{s,\text{out}} = \eta_T (u_1 - u_{2s}) = (0.95)(108.48 - 93.448) \text{ Btu/lbm} = 14.28 \text{ Btu/lbm}$$

The actual internal energy at the end of the expansion process is

$$w_{a,\text{out}} = (u_1 - u_{2a}) \longrightarrow u_{2a} = u_1 - w_{a,\text{out}} = 108.48 - 14.28 = 94.20 \text{ Btu/lbm}$$

The specific volume at the final state is (Table A-12E)

$$\left. \begin{array}{l} P_2 = 20 \text{ psia} \\ u_{2a} = 94.20 \text{ Btu/lbm} \end{array} \right\} \begin{array}{l} x_2 = \frac{u_{2a} - u_f}{u_{fg}} = \frac{94.20 - 11.401}{82.898} = 0.9988 \\ v_2 = v_f + x_2 v_{fg} = 0.01182 + (0.9988)(2.2772 - 0.01182) = 2.2745 \text{ ft}^3/\text{lbm} \end{array}$$

The final volume is then

$$V_2 = m v_2 = (10 \text{ lbm})(2.2745 \text{ ft}^3/\text{lbm}) = \mathbf{22.75 \text{ ft}^3}$$

8-128 Steam is expanded in an adiabatic turbine with an isentropic efficiency of 0.92. The power output of the turbine is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

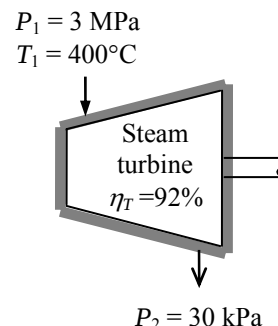
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\phi=0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \equiv \Delta \text{ke} \equiv \Delta \text{pe} \equiv 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2)$$



From the steam tables (Tables A-4 through A-6),

$$\left. \begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 3231.7 \text{ kJ/kg} \\ s_1 = 6.9235 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_{2s} = 30 \text{ kPa} \\ s_{2s} = s_1 \end{array} \right\} \begin{array}{l} x_{2s} = \frac{s_{2s} - s_f}{s_{fg}} = \frac{6.9235 - 0.9441}{6.8234} = 0.8763 \\ h_{2s} = h_f + x_{2s} h_{fg} = 289.27 + (0.8763)(2335.3) = 2335.7 \text{ kJ/kg} \end{array}$$

The actual power output may be determined by multiplying the isentropic power output with the isentropic efficiency. Then,

$$\begin{aligned} \dot{W}_{a,\text{out}} &= \eta_T \dot{W}_{s,\text{out}} \\ &= \eta_T \dot{m}(h_1 - h_{2s}) \\ &= (0.92)(2 \text{ kg/s})(3231.7 - 2335.7) \text{ kJ/kg} \\ &= \mathbf{1649 \text{ kW}} \end{aligned}$$

8-129 Steam is expanded in an adiabatic turbine with an isentropic efficiency of 0.90. The power output of the turbine is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

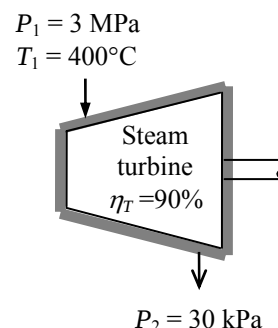
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\phi=0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \equiv \Delta \text{ke} \equiv \Delta \text{pe} \equiv 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2)$$



From the steam tables (Tables A-4 through A-6),

$$\left. \begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 3231.7 \text{ kJ/kg} \\ s_1 = 6.9235 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_{2s} = 30 \text{ kPa} \\ s_{2s} = s_1 \end{array} \right\} \begin{array}{l} x_{2s} = \frac{s_{2s} - s_f}{s_{fg}} = \frac{6.9235 - 0.9441}{6.8234} = 0.8763 \\ h_{2s} = h_f + x_{2s} h_{fg} = 289.27 + (0.8763)(2335.3) = 2335.7 \text{ kJ/kg} \end{array}$$

The actual power output may be determined by multiplying the isentropic power output with the isentropic efficiency. Then,

$$\begin{aligned} \dot{W}_{a,\text{out}} &= \eta_T \dot{W}_{s,\text{out}} \\ &= \eta_T \dot{m}(h_1 - h_{2s}) \\ &= (0.90)(2 \text{ kg/s})(3231.7 - 2335.7) \text{ kJ/kg} \\ &= \mathbf{1613 \text{ kW}} \end{aligned}$$

8-130 Argon gas is compressed by an adiabatic compressor. The isentropic efficiency of the compressor is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

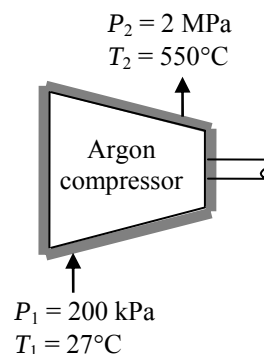
Properties The specific heat ratio of argon is $k = 1.667$ (Table A-2).

Analysis The isentropic exit temperature is

$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (300 \text{ K}) \left(\frac{2000 \text{ kPa}}{200 \text{ kPa}} \right)^{0.667/1.667} = 753.8 \text{ K}$$

From the isentropic efficiency relation,

$$\begin{aligned} \eta_C &= \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{c_p (T_{2s} - T_1)}{c_p (T_{2a} - T_1)} = \frac{T_{2s} - T_1}{T_{2a} - T_1} \\ &= \frac{753.8 - 300}{823 - 300} = 0.867 = \mathbf{86.7\%} \end{aligned}$$



8-131 R-134a is compressed by an adiabatic compressor. The isentropic efficiency of the compressor is to be determined.

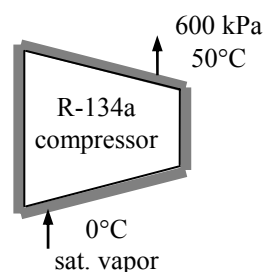
Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis From the R-134a tables (Tables A-11 through A-13),

$$\begin{aligned} \left. \begin{array}{l} T_1 = 0^\circ\text{C} \\ x = 1 \text{ (sat. vap.)} \end{array} \right\} & \begin{array}{l} h_1 = 250.45 \text{ kJ/kg} \\ s_1 = 0.93139 \text{ kJ/kg} \cdot \text{K} \end{array} \\ \left. \begin{array}{l} P_2 = 600 \text{ kPa} \\ T_2 = 50^\circ\text{C} \end{array} \right\} & h_{2a} = 290.28 \text{ kJ/kg} \\ \left. \begin{array}{l} P_2 = 600 \text{ kPa} \\ s_2 = s_1 = 0.93139 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} & h_{2s} = 265.25 \text{ kJ/kg} \end{aligned}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{265.25 - 250.45}{290.28 - 250.45} = 0.372 = \mathbf{37.2\%}$$



8-132 Steam enters an adiabatic turbine at a specified state, and leaves at a specified state. The mass flow rate of the steam and the isentropic efficiency are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis (a) From the steam tables (Tables A-4 and A-6),

$$\left. \begin{array}{l} P_1 = 7 \text{ MPa} \\ T_1 = 600^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 3650.6 \text{ kJ/kg} \\ s_1 = 7.0910 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 50 \text{ kPa} \\ T_2 = 150^\circ\text{C} \end{array} \right\} h_{2a} = 2780.2 \text{ kJ/kg}$$

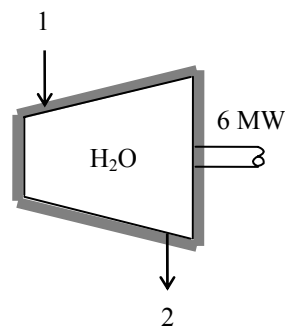
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{steady}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{a,\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta p_e \cong 0)$$

$$\dot{W}_{a,\text{out}} = -\dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$



Substituting, the mass flow rate of the steam is determined to be

$$6000 \text{ kJ/s} = -\dot{m} \left(2780.2 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right)$$

$$\dot{m} = \mathbf{6.95 \text{ kg/s}}$$

(b) The isentropic exit enthalpy of the steam and the power output of the isentropic turbine are

$$\left. \begin{array}{l} P_{2s} = 50 \text{ kPa} \\ s_{2s} = s_1 \end{array} \right\} \begin{array}{l} x_{2s} = \frac{s_{2s} - s_f}{s_{fg}} = \frac{7.0910 - 1.0912}{6.5019} = 0.9228 \\ h_{2s} = h_f + x_{2s} h_{fg} = 340.54 + (0.9228)(2304.7) = 2467.3 \text{ kJ/kg} \end{array}$$

and

$$\dot{W}_{s,\text{out}} = -\dot{m} \left(h_{2s} - h_1 + \frac{(V_2^2 - V_1^2)}{2} \right)$$

$$\dot{W}_{s,\text{out}} = -(6.95 \text{ kg/s}) \left(2467.3 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right)$$

$$= \mathbf{8174 \text{ kW}}$$

Then the isentropic efficiency of the turbine becomes

$$\eta_\tau = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000 \text{ kW}}{8174 \text{ kW}} = 0.734 = \mathbf{73.4\%}$$

8-133 Argon enters an adiabatic turbine at a specified state with a specified mass flow rate, and leaves at a specified pressure. The isentropic efficiency of the turbine is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 The device is adiabatic and thus heat transfer is negligible. 4 Argon is an ideal gas with constant specific heats.

Properties The specific heat ratio of argon is $k = 1.667$. The constant pressure specific heat of argon is $c_p = 0.5203 \text{ kJ/kg} \cdot \text{K}$ (Table A-2).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the isentropic turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\begin{aligned}\dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}h_1 &= \dot{W}_{s,\text{out}} + \dot{m}h_{2s} \quad (\text{since } \dot{Q} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0) \\ \dot{W}_{s,\text{out}} &= \dot{m}(h_1 - h_{2s})\end{aligned}$$

From the isentropic relations,

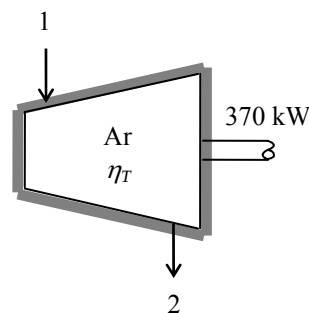
$$T_{2s} = T_1 \left(\frac{P_{2s}}{P_1} \right)^{(k-1)/k} = (1073 \text{ K}) \left(\frac{200 \text{ kPa}}{1500 \text{ kPa}} \right)^{0.667/1.667} = 479 \text{ K}$$

Then the power output of the isentropic turbine becomes

$$\dot{W}_{s,\text{out}} = \dot{m}c_p(T_1 - T_{2s}) = (80/60 \text{ kg/min})(0.5203 \text{ kJ/kg} \cdot \text{K})(1073 - 479) = 412.1 \text{ kW}$$

Then the isentropic efficiency of the turbine is determined from

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{370 \text{ kW}}{412.1 \text{ kW}} = 0.898 = \mathbf{89.8\%}$$



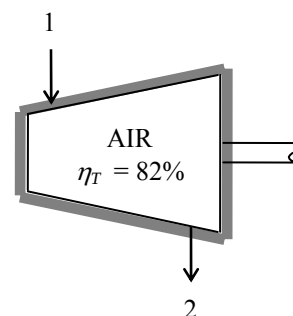
8-134E Combustion gases enter an adiabatic gas turbine with an isentropic efficiency of 82% at a specified state, and leave at a specified pressure. The work output of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

Analysis From the air table and isentropic relations,

$$T_1 = 2000 \text{ R} \longrightarrow \begin{array}{l} h_1 = 504.71 \text{ Btu/lbm} \\ P_{r_1} = 174.0 \end{array}$$

$$P_{r_2} = \left(\frac{P_2}{P_1} \right) P_{r_1} = \left(\frac{60 \text{ psia}}{120 \text{ psia}} \right) (174.0) = 87.0 \longrightarrow h_{2s} = 417.3 \text{ Btu/lbm}$$



There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\begin{aligned} \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}h_1 &= \dot{W}_{\text{a,out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0) \\ \dot{W}_{\text{a,out}} &= \dot{m}(h_1 - h_2) \end{aligned}$$

Noting that $w_a = \eta_T w_s$, the work output of the turbine per unit mass is determined from

$$w_a = (0.82)(504.71 - 417.3) \text{ Btu/lbm} = \mathbf{71.7 \text{ Btu/lbm}}$$

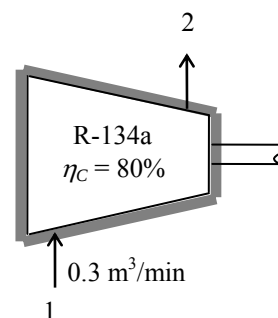
8-135 CD EES Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The compressor exit temperature and power input to the compressor are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis (a) From the refrigerant tables (Tables A-11E through A-13E),

$$\left. \begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} h_1 = h_{g@120 \text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g@120 \text{ kPa}} = 0.94779 \text{ kJ/kg} \cdot \text{K} \\ v_1 = v_{g@120 \text{ kPa}} = 0.16212 \text{ m}^3/\text{kg} \end{array}$$

$$\left. \begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array} \right\} h_{2s} = 281.21 \text{ kJ/kg}$$



From the isentropic efficiency relation,

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_c = 236.97 + (281.21 - 236.97)/0.80 = 292.26 \text{ kJ/kg}$$

Thus,

$$\left. \begin{array}{l} P_{2a} = 1 \text{ MPa} \\ h_{2a} = 292.26 \text{ kJ/kg} \end{array} \right\} T_{2a} = \mathbf{58.9^\circ\text{C}}$$

(b) The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60 \text{ m}^3/\text{s}}{0.16212 \text{ m}^3/\text{kg}} = 0.0308 \text{ kg/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{?0 (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor becomes,

$$\dot{W}_{\text{a,in}} = (0.0308 \text{ kg/s})(292.26 - 236.97) \text{ kJ/kg} = \mathbf{1.70 \text{ kW}}$$

8-136 EES Problem 8-135 is reconsidered. The problem is to be solved by considering the kinetic energy and by assuming an inlet-to-exit area ratio of 1.5 for the compressor when the compressor exit pipe inside diameter is 2 cm.

Analysis The problem is solved using EES, and the solution is given below.

"Input Data from diagram window"

{P[1] = 120 "kPa"

P[2] = 1000 "kPa"

Vol_dot_1 = 0.3 "m^3/min"

Eta_c = 0.80 "Compressor adiabatic efficiency"

A_ratio = 1.5

d_2 = 2/100 "m"}

"System: Control volume containing the compressor, see the diagram window.

Property Relation: Use the real fluid properties for R134a.

Process: Steady-state, steady-flow, adiabatic process."

Fluid\$='R134a'

"Property Data for state 1"

T[1]=temperature(Fluid\$,P=P[1],x=1)"Real fluid equ. at the sat. vapor state"

h[1]=enthalpy(Fluid\$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

s[1]=entropy(Fluid\$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

v[1]=volume(Fluid\$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

"Property Data for state 2"

s_s[1]=s[1]; T_s[1]=T[1] "needed for plot"

s_s[2]=s[1] "for the ideal, isentropic process across the compressor"

h_s[2]=ENTHALPY(Fluid\$, P=P[2], s=s_s[2])"Enthalpy 2 at the isentropic state 2s and pressure P[2]"

T_s[2]=Temperature(Fluid\$, P=P[2], s=s_s[2])"Temperature of ideal state - needed only for plot."

"Steady-state, steady-flow conservation of mass"

m_dot_1 = m_dot_2

m_dot_1 = Vol_dot_1/(v[1]*60)

Vol_dot_1/v[1]=Vol_dot_2/v[2]

Vel[2]=Vol_dot_2/(A[2]*60)

A[2] = pi*(d_2)^2/4

A_ratio*Vel[1]/v[1] = Vel[2]/v[2] "Mass flow rate: = A*Vel/v, A_ratio = A[1]/A[2]"

A_ratio=A[1]/A[2]

"Steady-state, steady-flow conservation of energy, adiabatic compressor, see diagram window"

m_dot_1*(h[1]+(Vel[1])^2/(2*1000)) + W_dot_c = m_dot_2*(h[2]+(Vel[2])^2/(2*1000))

"Definition of the compressor adiabatic efficiency, Eta_c=W_isen/W_act"

Eta_c = (h_s[2]-h[1])/(h[2]-h[1])

"Knowing h[2], the other properties at state 2 can be found."

v[2]=volume(Fluid\$, P=P[2], h=h[2])"v[2] is found at the actual state 2, knowing P and h."

T[2]=temperature(Fluid\$, P=P[2],h=h[2])"Real fluid equ. for T at the known outlet h and P."

s[2]=entropy(Fluid\$, P=P[2], h=h[2]) "Real fluid equ. at the known outlet h and P."

T_exit=T[2]

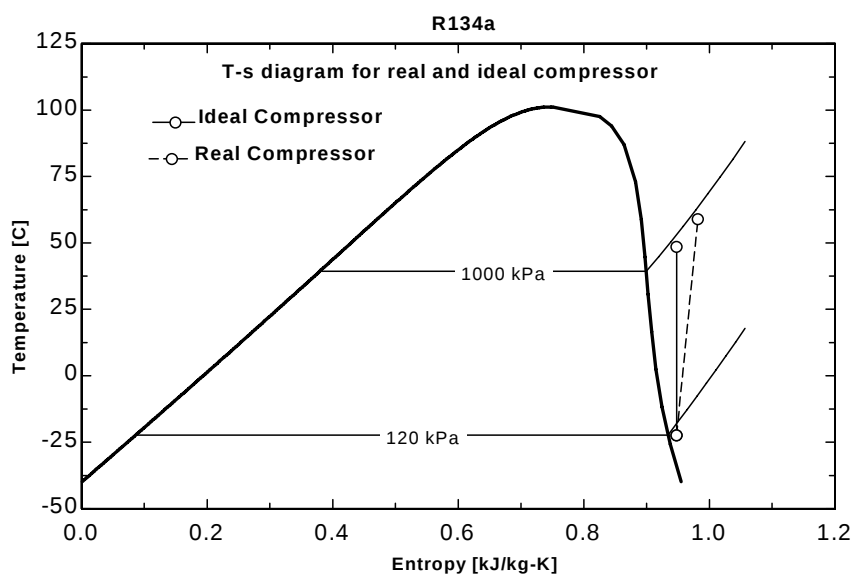
"Neglecting the kinetic energies, the work is:"

m_dot_1*h[1] + W_dot_c_noke = m_dot_2*h[2]

SOLUTION

$A[1]=0.0004712 \text{ [m}^2\text{]}$
 $A[2]=0.0003142 \text{ [m}^2\text{]}$
 $A_{\text{ratio}}=1.5$
 $d_2=0.02 \text{ [m]}$
 $\text{Eta}_c=0.8$
 $\text{Fluid}=\text{'R134a'}$
 $h[1]=237 \text{ [kJ/kg]}$
 $h[2]=292.3 \text{ [kJ/kg]}$
 $h_s[2]=281.2 \text{ [kJ/kg]}$
 $m_{\text{dot}}[1]=0.03084 \text{ [kg/s]}$
 $m_{\text{dot}}[2]=0.03084 \text{ [kg/s]}$
 $P[1]=120.0 \text{ [kPa]}$
 $P[2]=1000.0 \text{ [kPa]}$
 $s[1]=0.9478 \text{ [kJ/kg-K]}$
 $s[2]=0.9816 \text{ [kJ/kg-K]}$

$s_s[1]=0.9478 \text{ [kJ/kg-K]}$
 $s_s[2]=0.9478 \text{ [kJ/kg-K]}$
 $T[1]=-22.32 \text{ [C]}$
 $T[2]=58.94 \text{ [C]}$
 $T_{\text{exit}}=58.94 \text{ [C]}$
 $T_s[1]=-22.32 \text{ [C]}$
 $T_s[2]=48.58 \text{ [C]}$
 $\text{Vol}_{\text{dot}}[1]=0.3 \text{ [m}^3\text{/min]}$
 $\text{Vol}_{\text{dot}}[2]=0.04244 \text{ [m}^3\text{/min]}$
 $v[1]=0.1621 \text{ [m}^3\text{/kg]}$
 $v[2]=0.02294 \text{ [m}^3\text{/kg]}$
 $\text{Vel}[1]=10.61 \text{ [m/s]}$
 $\text{Vel}[2]=2.252 \text{ [m/s]}$
 $W_{\text{dot}}[c]=1.704 \text{ [kW]}$
 $W_{\text{dot}}[c]_{\text{noke}}=1.706 \text{ [kW]}$



8-137 Air enters an adiabatic compressor with an isentropic efficiency of 84% at a specified state, and leaves at a specified temperature. The exit pressure of air and the power input to the compressor are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 The device is adiabatic and thus heat transfer is negligible. 4 Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-1)

Analysis (a) From the air table (Table A-21),

$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg}, \quad P_{r1} = 1.2311$$

$$T_2 = 530 \text{ K} \longrightarrow h_{2a} = 533.98 \text{ kJ/kg}$$

From the isentropic efficiency relation $\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1}$,

$$\begin{aligned} h_{2s} &= h_1 + \eta_c (h_{2a} - h_1) \\ &= 290.16 + (0.84)(533.98 - 290.16) = 495.0 \text{ kJ/kg} \longrightarrow P_{r2} = 7.951 \end{aligned}$$

Then from the isentropic relation ,

$$\frac{P_2}{P_1} = \frac{P_{r2}}{P_{r1}} \longrightarrow P_2 = \left(\frac{P_{r2}}{P_{r1}} \right) P_1 = \left(\frac{7.951}{1.2311} \right) (100 \text{ kPa}) = \mathbf{646 \text{ kPa}}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

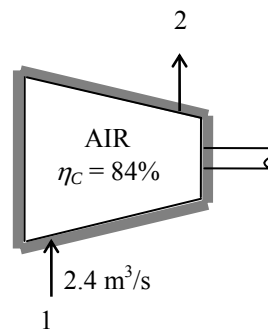
$$\begin{aligned} \underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} &= \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0 \\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{W}_{\text{a,in}} + \dot{m}h_1 &= \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0) \\ \dot{W}_{\text{a,in}} &= \dot{m}(h_2 - h_1) \end{aligned}$$

where

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100 \text{ kPa})(2.4 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 2.884 \text{ kg/s}$$

Then the power input to the compressor is determined to be

$$\dot{W}_{\text{a,in}} = (2.884 \text{ kg/s})(533.98 - 290.16) \text{ kJ/kg} = \mathbf{703 \text{ kW}}$$



8-138 Air is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor and the exit temperature of air for the isentropic case are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Analysis (a) From the air table (Table A-21),

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}, \quad P_{r_1} = 1.386$$

$$T_2 = 550 \text{ K} \longrightarrow h_{2a} = 554.74 \text{ kJ/kg}$$

From the isentropic relation,

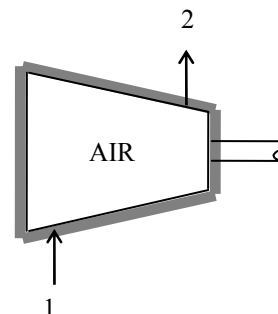
$$P_{r_2} = \left(\frac{P_2}{P_1} \right) P_{r_1} = \left(\frac{600 \text{ kPa}}{95 \text{ kPa}} \right) (1.386) = 8.754 \longrightarrow h_{2s} = 508.72 \text{ kJ/kg}$$

Then the isentropic efficiency becomes

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{508.72 - 300.19}{554.74 - 300.19} = 0.819 = \mathbf{81.9\%}$$

(b) If the process were isentropic, the exit temperature would be

$$h_{2s} = 508.72 \text{ kJ/kg} \longrightarrow T_{2s} = \mathbf{505.5 \text{ K}}$$



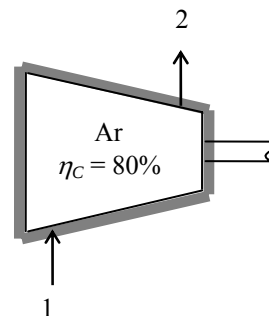
8-139E Argon enters an adiabatic compressor with an isentropic efficiency of 80% at a specified state, and leaves at a specified pressure. The exit temperature of argon and the work input to the compressor are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Potential energy changes are negligible. 3 The device is adiabatic and thus heat transfer is negligible. 4 Argon is an ideal gas with constant specific heats.

Properties The specific heat ratio of argon is $k = 1.667$. The constant pressure specific heat of argon is $c_p = 0.1253 \text{ Btu/lbm} \cdot \text{R}$ (Table A-2E).

Analysis (a) The isentropic exit temperature T_{2s} is determined from

$$T_{2s} = T_1 \left(\frac{P_{2s}}{P_1} \right)^{(k-1)/k} = (550 \text{ R}) \left(\frac{200 \text{ psia}}{20 \text{ psia}} \right)^{0.667/1.667} = 1381.9 \text{ R}$$



The actual kinetic energy change during this process is

$$\Delta ke_a = \frac{V_2^2 - V_1^2}{2} = \frac{(240 \text{ ft/s})^2 - (60 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = 1.08 \text{ Btu/lbm}$$

The effect of kinetic energy on isentropic efficiency is very small. Therefore, we can take the kinetic energy changes for the actual and isentropic cases to be same in efficiency calculations. From the isentropic efficiency relation, including the effect of kinetic energy,

$$\eta_c = \frac{w_s}{w_a} = \frac{(h_{2s} - h_1) + \Delta ke}{(h_{2a} - h_1) + \Delta ke} = \frac{c_p(T_{2s} - T_1) + \Delta ke_s}{c_p(T_{2a} - T_1) + \Delta ke_a} \longrightarrow 0.8 = \frac{0.1253(1381.9 - 550) + 1.08}{0.1253(T_{2a} - 550) + 1.08}$$

It yields

$$T_{2a} = \mathbf{1592 \text{ R}}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{ } \neq 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{a,\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \equiv \Delta p e \cong 0)$$

$$\dot{W}_{a,\text{in}} = \dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right) \longrightarrow w_{a,\text{in}} = h_2 - h_1 + \Delta ke$$

Substituting, the work input to the compressor is determined to be

$$w_{a,\text{in}} = (0.1253 \text{ Btu/lbm} \cdot \text{R})(1592 - 550) \text{ R} + 1.08 \text{ Btu/lbm} = \mathbf{131.6 \text{ Btu/lbm}}$$

8-140E Air is accelerated in a 90% efficient adiabatic nozzle from low velocity to a specified velocity. The exit temperature and pressure of the air are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Analysis From the air table (Table A-21E),

$$T_1 = 1480 \text{ R} \longrightarrow h_1 = 363.89 \text{ Btu/lbm}, \quad P_{r_1} = 53.04$$

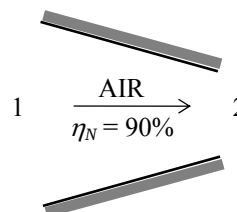
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\phi 0} \text{ (steady)}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta p e \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2}$$



Substituting, the exit temperature of air is determined to be

$$h_2 = 363.89 \text{ kJ/kg} - \frac{(800 \text{ ft/s})^2 - 0}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = 351.11 \text{ Btu/lbm}$$

From the air table we read

$$T_{2a} = 1431.3 \text{ R}$$

From the isentropic efficiency relation

$$\eta_N = \frac{h_{2a} - h_1}{h_{2s} - h_1} \longrightarrow$$

$$h_{2s} = h_1 + (h_{2a} - h_1)/\eta_N = 363.89 + (351.11 - 363.89)/(0.90) = 349.69 \text{ Btu/lbm} \longrightarrow P_{r_2} = 46.04$$

Then the exit pressure is determined from the isentropic relation to be

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}} \right) P_1 = \left(\frac{46.04}{53.04} \right) (60 \text{ psia}) = 52.1 \text{ psia}$$

8-141E EES Problem 8-140E is reconsidered. The effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on the exit temperature and pressure of the air is to be investigated, and the results are to be plotted.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

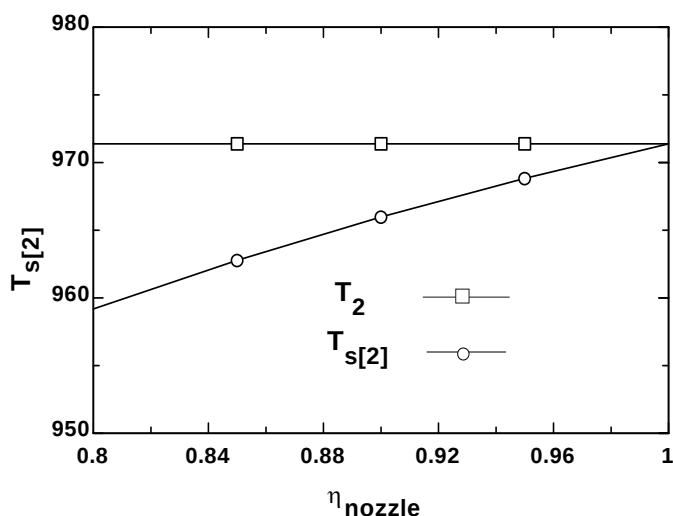
"Knowns:"

WorkFluid\$ = 'Air'
 $P[1] = 60$ [psia]
 $T[1] = 1020$ [F]
 $Vel[2] = 800$ [ft/s]
 $Vel[1] = 0$ [ft/s]
 $\eta_{\text{nozzle}} = 0.9$

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential energy, no heat transfer:"

$h[1] = \text{enthalpy}(\text{WorkFluid}\$, T=T[1])$
 $s[1] = \text{entropy}(\text{WorkFluid}\$, P=P[1], T=T[1])$
 $T_{s[1]} = T[1]$
 $s[2] = s[1]$
 $s_{s[2]} = s[1]$
 $h_{s[2]} = \text{enthalpy}(\text{WorkFluid}\$, T=T_{s[2]})$
 $T_{s[2]} = \text{temperature}(\text{WorkFluid}\$, P=P[2], s=s_{s[2]})$
 $\eta_{\text{nozzle}} = ke[2]/ke_{s[2]}$
 $ke[1] = Vel[1]^2/2$
 $ke[2] = Vel[2]^2/2$
 $h[1] + ke[1] \cdot \text{convert}(\text{ft}^2/\text{s}^2, \text{Btu/lbm}) = h[2] + ke[2] \cdot \text{convert}(\text{ft}^2/\text{s}^2, \text{Btu/lbm})$
 $h[1] + ke[1] \cdot \text{convert}(\text{ft}^2/\text{s}^2, \text{Btu/lbm}) = h_{s[2]} + ke_{s[2]} \cdot \text{convert}(\text{ft}^2/\text{s}^2, \text{Btu/lbm})$
 $T[2] = \text{temperature}(\text{WorkFluid}\$, h=h[2])$
 $P_2_answer = P[2]$
 $T_2_answer = T[2]$

η_{nozzle}	P_2 [psia]	T_2 [F]	$T_{s,2}$ [F]
0.8	51.09	971.4	959.2
0.85	51.58	971.4	962.8
0.9	52.03	971.4	966
0.95	52.42	971.4	968.8
1	52.79	971.4	971.4



8-142 Air is expanded in an adiabatic nozzle with an isentropic efficiency of 0.96. The air velocity at the exit is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ and $k = 1.4$ (Table A-2a).

Analysis For the isentropic process of an ideal gas, the exit temperature is determined from

$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (180 + 273 \text{ K}) \left(\frac{100 \text{ kPa}}{300 \text{ kPa}} \right)^{0.4/1.4} = 331.0 \text{ K}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

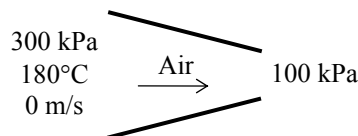
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$h_1 - h_2 = \frac{V_2^2 - V_1^2}{2}$$

$$c_p (T_1 - T_2) = \frac{V_2^2 - V_1^2}{2} = \Delta \text{ke}$$



The kinetic energy change for the isentropic case is

$$\Delta \text{ke}_s = c_p (T_1 - T_{2s}) = (1.005 \text{ kJ/kg}\cdot\text{K})(453 - 331)\text{K} = 122.6 \text{ kJ/kg}$$

The kinetic energy change for the actual process is

$$\Delta \text{ke}_a = \eta_N \Delta \text{ke}_s = (0.96)(122.6 \text{ kJ/kg}) = 117.7 \text{ kJ/kg}$$

Substituting into the energy balance and solving for the exit velocity gives

$$V_2 = (2\Delta \text{ke}_a)^{0.5} = \left[2(117.7 \text{ kJ/kg}) \left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}} \right) \right]^{0.5} = \mathbf{485 \text{ m/s}}$$

8-143E Air is decelerated in an adiabatic diffuser with an isentropic efficiency of 0.82. The air velocity at the exit is to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $k = 1.4$ (Table A-2Ea).

Analysis For the isentropic process of an ideal gas, the exit temperature is determined from

$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (30 + 460 \text{ R}) \left(\frac{20 \text{ psia}}{13 \text{ psia}} \right)^{0.4/1.4} = 554.2 \text{ R}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\approx 0 \text{ (steady)}} = 0$$

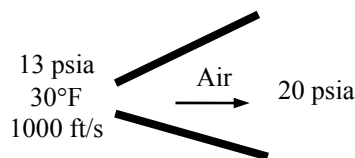
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$h_1 - h_2 = \frac{V_2^2 - V_1^2}{2}$$

$$c_p (T_1 - T_2) = \frac{V_2^2 - V_1^2}{2} = \Delta \text{ke}$$



The kinetic energy change for the isentropic case is

$$\Delta \text{ke}_s = c_p (T_{2s} - T_1) = (0.240 \text{ Btu/lbm} \cdot \text{R})(554.2 - 490) \text{ R} = 15.41 \text{ Btu/lbm}$$

The kinetic energy change for the actual process is

$$\Delta \text{ke}_a = \eta_N \Delta \text{ke}_s = (0.82)(15.41 \text{ Btu/lbm}) = 12.63 \text{ Btu/lbm}$$

Substituting into the energy balance and solving for the exit velocity gives

$$V_2 = (V_1^2 - 2\Delta \text{ke}_a)^{0.5} = \left[(1000 \text{ ft/s})^2 - 2(12.63 \text{ Btu/lbm}) \left(\frac{25,037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right) \right]^{0.5} = \mathbf{606 \text{ ft/s}}$$

Entropy Balance

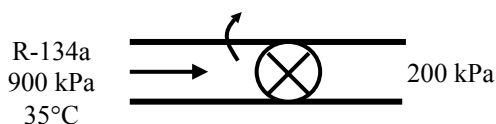
8-144 Heat is lost from Refrigerant-134a as it is throttled. The exit temperature of the refrigerant and the entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible.

Analysis The properties of the refrigerant at the inlet of the device are (Table A-13)

$$\left. \begin{array}{l} P_1 = 900 \text{ kPa} \\ T_1 = 35^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 100.86 \text{ kJ/kg} \\ s_1 = 0.37135 \text{ kJ/kg}\cdot\text{K} \end{array}$$

The enthalpy of the refrigerant at the exit of the device is



$$h_2 = h_1 - q_{\text{out}} = 100.86 - 0.8 = 100.06 \text{ kJ/kg}$$

Now, the properties at the exit state may be obtained from the R-134a tables

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ h_2 = 100.06 \text{ kJ/kg} \end{array} \right\} \begin{array}{l} T_2 = -10.09^\circ\text{C} \\ s_2 = 0.38884 \text{ kJ/kg}\cdot\text{K} \end{array}$$

The entropy generation associated with this process may be obtained by adding the entropy change of R-134a as it flows in the device and the entropy change of the surroundings.

$$\Delta s_{\text{R-134a}} = s_2 - s_1 = 0.38884 - 0.37135 = 0.01749 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta s_{\text{surr}} = \frac{q_{\text{out}}}{T_{\text{surr}}} = \frac{0.8 \text{ kJ/kg}}{(25 + 273) \text{ K}} = 0.002685 \text{ kJ/kg}\cdot\text{K}$$

$$s_{\text{gen}} = \Delta s_{\text{total}} = \Delta s_{\text{R-134a}} + \Delta s_{\text{surr}} = 0.01749 + 0.002685 = \mathbf{0.0202 \text{ kJ/kg}\cdot\text{K}}$$

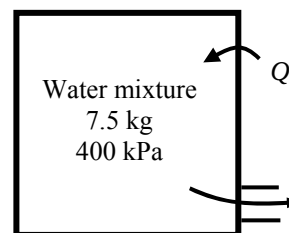
8-145 Liquid water is withdrawn from a rigid tank that initially contains saturated water mixture until no liquid is left in the tank. The quality of steam in the tank at the initial state, the amount of mass that has escaped, and the entropy generation during this process are to be determined.

Assumptions 1 Kinetic and potential energy changes are zero. 2 There are no work interactions.

Analysis (a) The properties of the steam in the tank at the final state and the properties of exiting steam are (Tables A-4 through A-6)

$$\left. \begin{array}{l} P_2 = 400 \text{ kPa} \\ x_2 = 1 \text{ (sat. vap.)} \end{array} \right\} \begin{array}{l} u_2 = 2553.1 \text{ kJ/kg} \\ v_2 = 0.46242 \text{ m}^3/\text{kg} \\ s_2 = 6.8955 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left. \begin{array}{l} P_{2e} = 400 \text{ kPa} \\ x_e = 0 \text{ (sat. liq.)} \end{array} \right\} \begin{array}{l} h_e = 604.66 \text{ kJ/kg} \\ s_e = 1.7765 \text{ kJ/kg}\cdot\text{K} \end{array}$$



The relations for the volume of the tank and the final mass in the tank are

$$\begin{aligned} V &= m_1 v_1 = (7.5 \text{ kg})v_1 \\ m_2 &= \frac{V}{v_2} = \frac{(7.5 \text{ kg})v_1}{0.46242 \text{ m}^3/\text{kg}} = 16.219v_1 \end{aligned}$$

The mass, energy, and entropy balances may be written as

$$m_e = m_1 - m_2$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$\frac{Q_{\text{in}}}{T_{\text{source}}} - m_e s_e + S_{\text{gen}} = m_2 s_2 - m_1 s_1$$

Substituting,

$$m_e = 7.5 - 16.219v_1 \quad (1)$$

$$5 - (7.5 - 16.219v_1)(604.66) = 16.219v_1(2553.1) - 7.5u_1 \quad (2)$$

$$\frac{5}{500 + 273} - (7.5 - 16.219v_1)(1.7765) + S_{\text{gen}} = 16.219v_1(6.8955) - 7.5s_1 \quad (3)$$

Eq. (2) may be solved by a trial-error approach by trying different qualities at the inlet state. Or, we can use EES to solve the equations to find

$$x_1 = \mathbf{0.8666}$$

Other properties at the initial state are

$$\left. \begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.8666 \end{array} \right\} \begin{array}{l} u_1 = 2293.2 \text{ kJ/kg} \\ v_1 = 0.40089 \text{ m}^3/\text{kg} \\ s_1 = 6.2129 \text{ kJ/kg}\cdot\text{K} \end{array}$$

Substituting into Eqs (1) and (3),

$$(b) \quad m_e = 7.5 - 16.219(0.40089) = \mathbf{0.998 \text{ kg}}$$

(c)

$$\frac{5}{500 + 273} - [7.5 - 16.219(0.40089)](1.7765) + S_{\text{gen}} = 16.219(0.40089)(6.8955) - 7.5(6.2129)$$

$$S_{\text{gen}} = \mathbf{0.00553 \text{ kJ/K}}$$

8-146 Each member of a family of four take a 5-min shower every day. The amount of entropy generated by this family per year is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The kinetic and potential energies are negligible. 3 Heat losses from the pipes and the mixing section are negligible and thus $\dot{Q} \cong 0$. 4 Showers operate at maximum flow conditions during the entire shower. 5 Each member of the household takes a 5-min shower every day. 6 Water is an incompressible substance with constant properties at room temperature. 7 The efficiency of the electric water heater is 100%.

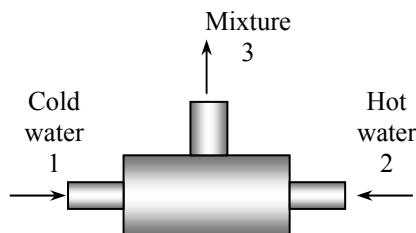
Properties The density and specific heat of water at room temperature are $\rho = 1 \text{ kg/L}$ and $c = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis The mass flow rate of water at the shower head is

$$\dot{m} = \rho \dot{V} = (1 \text{ kg/L})(12 \text{ L/min}) = 12 \text{ kg/min}$$

The mass balance for the mixing chamber can be expressed in the rate form as

$$\begin{aligned} \dot{m}_{\text{in}} - \dot{m}_{\text{out}} &= \Delta \dot{m}_{\text{system}} \stackrel{\text{no (steady)}}{=} 0 \\ \dot{m}_{\text{in}} &= \dot{m}_{\text{out}} \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 \end{aligned}$$



where the subscript 1 denotes the cold water stream, 2 the hot water stream, and 3 the mixture.

The rate of entropy generation during this process can be determined by applying the rate form of the entropy balance on a system that includes the electric water heater and the mixing chamber (the T-elbow). Noting that there is no entropy transfer associated with work transfer (electricity) and there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\begin{aligned} \underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} &= \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \stackrel{\text{no (steady)}}{=} 0 \\ \dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} &= 0 \quad (\text{since } Q = 0 \text{ and work is entropy free}) \\ \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 \end{aligned}$$

Noting from mass balance that $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$ and $s_2 = s_1$ since hot water enters the system at the same temperature as the cold water, the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - (\dot{m}_1 + \dot{m}_2) s_1 = \dot{m}_3 (s_3 - s_1) = \dot{m}_3 c_p \ln \frac{T_3}{T_1} \\ &= (12 \text{ kg/min})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{42 + 273}{15 + 273} = 4.495 \text{ kJ/min} \cdot \text{K} \end{aligned}$$

Noting that 4 people take a 5-min shower every day, the amount of entropy generated per year is

$$\begin{aligned} S_{\text{gen}} &= (\dot{S}_{\text{gen}}) \Delta t (\text{No. of people}) (\text{No. of days}) \\ &= (4.495 \text{ kJ/min} \cdot \text{K}) (5 \text{ min/person} \cdot \text{day}) (4 \text{ persons}) (365 \text{ days/year}) \\ &= \mathbf{32,814 \text{ kJ/K}} \quad (\text{per year}) \end{aligned}$$

Discussion The value above represents the entropy generated within the water heater and the T-elbow in the absence of any heat losses. It does not include the entropy generated as the shower water at 42°C is discarded or cooled to the outdoor temperature. Also, an entropy balance on the mixing chamber alone (hot water entering at 55°C instead of 15°C) will exclude the entropy generated within the water heater.

8-147 Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam and the rate of entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The enthalpy and entropy of vaporization of water at 60°C are $h_{fg} = 2357.7$ kJ/kg and $s_{fg} = 7.0769$ kJ/kg.K (Table A-4). The specific heat of water at room temperature is $c_p = 4.18$ kJ/kg.°C (Table A-3).

Analysis (a) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the heat transfer rate to the cooling water in the condenser becomes

$$\begin{aligned} \dot{Q} &= [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cooling water}} \\ &= (75 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(27^\circ\text{C} - 18^\circ\text{C}) = 2822 \text{ kJ/s} \end{aligned}$$

The rate of condensation of steam is determined to be

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{2822 \text{ kJ/s}}{2357.7 \text{ kJ/kg}} = \mathbf{1.20 \text{ kg/s}}$$

(b) The rate of entropy generation within the condenser during this process can be determined by applying the rate form of the entropy balance on the entire condenser. Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\approx 0 \text{ (steady)}}{=}$$

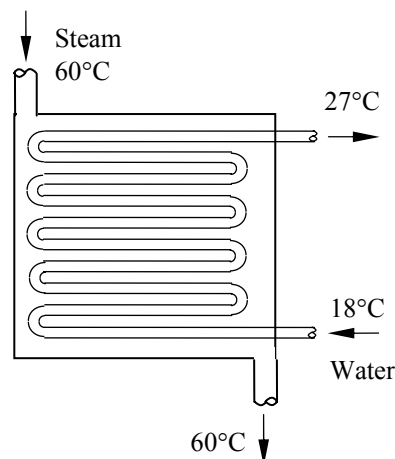
$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{steam}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{steam}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_{\text{water}} c_p \ln \frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}} c_p \ln \frac{T_2}{T_1} - \dot{m}_{\text{steam}} s_{fg} \\ &= (75 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{27 + 273}{18 + 273} - (1.20 \text{ kg/s})(7.0769 \text{ kJ/kg} \cdot \text{K}) \\ &= \mathbf{1.06 \text{ kW/K}} \end{aligned}$$



8-148 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg·°C, respectively.

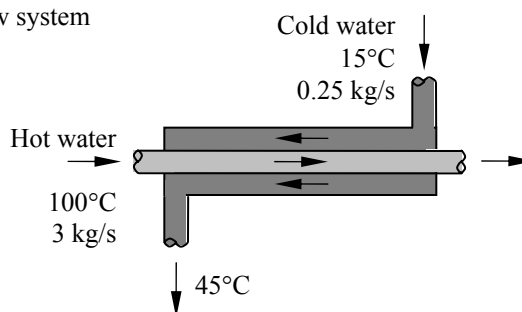
Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q}_{\text{in}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cold water}} = (0.25 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(45^\circ\text{C} - 15^\circ\text{C}) = \mathbf{31.35 \text{ kW}}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{hot water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 100^\circ\text{C} - \frac{31.35 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg} \cdot ^\circ\text{C})} = 97.5^\circ\text{C}$$

(b) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{cold}} s_1 + \dot{m}_{\text{hot}} s_3 - \dot{m}_{\text{cold}} s_2 - \dot{m}_{\text{hot}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{cold}}(s_2 - s_1) + \dot{m}_{\text{hot}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_{\text{cold}} c_p \ln \frac{T_2}{T_1} + \dot{m}_{\text{hot}} c_p \ln \frac{T_4}{T_3} \\ &= (0.25 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{45 + 273}{15 + 273} + (3 \text{ kg/s})(4.19 \text{ kJ/kg} \cdot \text{K}) \ln \frac{97.5 + 273}{100 + 273} \\ &= \mathbf{0.0190 \text{ kW/K}} \end{aligned}$$

8-149 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer, the outlet temperature of the air, and the rate of entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg·°C, respectively. The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1).

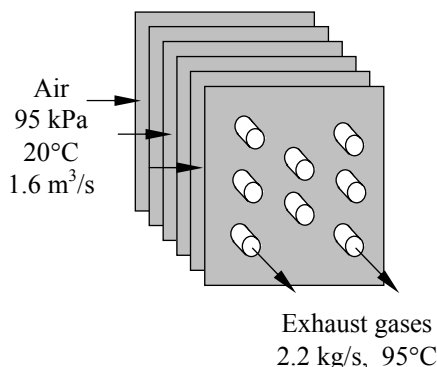
Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas}} = (2.2 \text{ kg/s})(1.1 \text{ kJ/kg} \cdot ^\circ\text{C})(180^\circ\text{C} - 95^\circ\text{C}) = \mathbf{205.7 \text{ kW}}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95 \text{ kPa})(1.6 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 1.808 \text{ kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}c_p} = 20^\circ\text{C} + \frac{205.7 \text{ kW}}{(1.808 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{133.2^\circ\text{C}}$$

The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\text{no (steady)}}{=} 0$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } \dot{Q} = 0)$$

$$\dot{m}_{\text{exhaust}} s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_{\text{exhaust}} s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}}(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3)$$

Then the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_{\text{exhaust}} c_p \ln \frac{T_2}{T_1} + \dot{m}_{\text{air}} c_p \ln \frac{T_4}{T_3} \\ &= (2.2 \text{ kg/s})(1.1 \text{ kJ/kg} \cdot \text{K}) \ln \frac{95 + 273}{180 + 273} + (1.808 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot \text{K}) \ln \frac{133.2 + 273}{20 + 273} \\ &= \mathbf{0.091 \text{ kW/K}} \end{aligned}$$

8-150 Water is heated by hot oil in a heat exchanger. The outlet temperature of the oil and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg·°C, respectively.

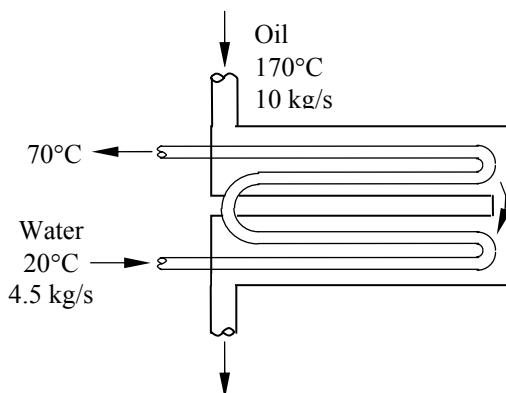
Analysis (a) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(70^\circ\text{C} - 20^\circ\text{C}) = 940.5 \text{ kW}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} \rightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 170^\circ\text{C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg} \cdot ^\circ\text{C})} = 129.1^\circ\text{C}$$

(b) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{m}_1s_1 + \dot{m}_3s_3 - \dot{m}_2s_2 - \dot{m}_4s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } \dot{Q} = 0)$$

$$\dot{m}_{\text{water}}s_1 + \dot{m}_{\text{oil}}s_3 - \dot{m}_{\text{water}}s_2 - \dot{m}_{\text{oil}}s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{oil}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_{\text{water}}c_p \ln \frac{T_2}{T_1} + \dot{m}_{\text{oil}}c_p \ln \frac{T_4}{T_3} \\ &= (4.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{70 + 273}{20 + 273} + (10 \text{ kg/s})(2.3 \text{ kJ/kg} \cdot \text{K}) \ln \frac{129.1 + 273}{170 + 273} \\ &= \mathbf{0.736 \text{ kW/K}} \end{aligned}$$

8-151E Refrigerant-134a is expanded adiabatically from a specified state to another. The entropy generation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible.

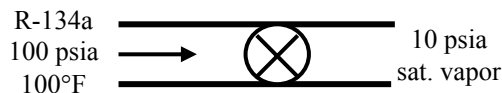
Analysis The rate of entropy generation within the expansion device during this process can be determined by applying the rate form of the entropy balance on the system. Noting that the system is adiabatic and thus there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\approx 0 \text{ (steady)}}{=}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

$$s_{\text{gen}} = s_2 - s_1$$



The properties of the refrigerant at the inlet and exit states are (Tables A-11E through A-13E)

$$\left. \begin{array}{l} P_1 = 100 \text{ psia} \\ T_1 = 100^\circ\text{F} \end{array} \right\} s_1 = 0.22900 \text{ Btu/lbm} \cdot \text{R}$$

$$\left. \begin{array}{l} P_2 = 10 \text{ psia} \\ x_2 = 1 \end{array} \right\} s_2 = 0.22948 \text{ Btu/lbm} \cdot \text{R}$$

Substituting,

$$s_{\text{gen}} = s_2 - s_1 = 0.22948 - 0.22900 = \mathbf{0.00048 \text{ Btu/lbm} \cdot \text{R}}$$

8-152 In an ice-making plant, water is frozen by evaporating saturated R-134a liquid. The rate of entropy generation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible.

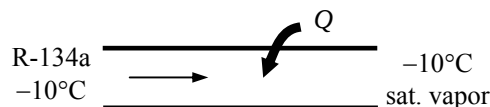
Analysis We take the control volume formed by the R-134a evaporator with a single inlet and single exit as the system. The rate of entropy generation within this evaporator during this process can be determined by applying the rate form of the entropy balance on the system. The entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \quad \text{0 (steady)}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{\text{in}}}{T_w} + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_R (s_2 - s_1) - \frac{\dot{Q}_{\text{in}}}{T_w}$$

$$\dot{S}_{\text{gen}} = \dot{m}_R s_{fg} - \frac{\dot{Q}_{\text{in}}}{T_w}$$



The properties of the refrigerant are (Table A-11)

$$h_{fg} @ -10^\circ\text{C} = 205.96 \text{ kJ/kg}$$

$$s_{fg} @ -10^\circ\text{C} = 0.78263 \text{ kJ/kg} \cdot \text{K}$$

The rate of that must be removed from the water in order to freeze it at a rate of 4000 kg/h is

$$\dot{Q}_{\text{in}} = \dot{m}_w h_{if} = (4000 / 3600 \text{ kg/s})(333.7 \text{ kJ/kg}) = 370.8 \text{ kW}$$

where the heat of fusion of water at 1 atm is 333.7 kJ/kg. The mass flow rate of R-134a is

$$\dot{m}_R = \frac{\dot{Q}_{\text{in}}}{h_{fg}} = \frac{370.8 \text{ kJ/s}}{205.96 \text{ kJ/kg}} = 1.800 \text{ kg/s}$$

Substituting,

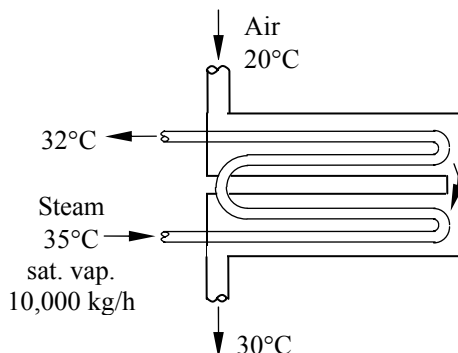
$$\dot{S}_{\text{gen}} = \dot{m}_R s_{fg} - \frac{\dot{Q}_{\text{in}}}{T_w} = (1.800 \text{ kg/s})(0.78263 \text{ kJ/kg} \cdot \text{K}) - \frac{370.8 \text{ kW}}{273 \text{ K}} = \mathbf{0.0505 \text{ kW/K}}$$

8-153 Air is heated by steam in a heat exchanger. The rate of entropy generation associated with this process is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Air is an ideal gas with constant specific heats.

Properties The specific heat of air at room temperature is $c_p = 1.005 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-2).

Analysis The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:



$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\phi=0 \text{ (steady)}}{=} 0$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} (s_2 - s_1) + \dot{m}_{\text{air}} (s_4 - s_3)$$

The properties of the steam at the inlet and exit states are

$$\left. \begin{array}{l} T_1 = 35^\circ\text{C} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} h_1 = 2564.6 \text{ kJ/kg} \\ s_1 = 8.3517 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{Table A-4})$$

$$\left. \begin{array}{l} T_2 = 32^\circ\text{C} \\ x_2 = 0 \end{array} \right\} \begin{array}{l} h_2 = 134.10 \text{ kJ/kg} \\ s_2 = 0.4641 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{Table A-4})$$

From an energy balance, the heat given up by steam is equal to the heat picked up by the air. Then,

$$\dot{Q} = \dot{m}_{\text{water}} (h_1 - h_2) = (10,000 / 3600 \text{ kg/s})(2564.6 - 134.10) \text{ kJ/kg} = 6751 \text{ kW}$$

$$\dot{m}_{\text{air}} = \frac{\dot{Q}}{c_p (T_4 - T_3)} = \frac{6751 \text{ kW}}{(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})(30 - 20)^\circ\text{C}} = 671.7 \text{ kg/s}$$

Substituting into the entropy balance relation,

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_{\text{water}} (s_2 - s_1) + \dot{m}_{\text{air}} (s_4 - s_3) \\ &= \dot{m}_{\text{water}} (s_2 - s_1) + \dot{m}_{\text{air}} c_p \ln \frac{T_4}{T_3} \\ &= (10,000 / 3600 \text{ kg/s})(0.4641 - 8.3517) \text{ kJ/kg} \cdot \text{K} + (671.7 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot \text{K}) \ln \frac{303 \text{ K}}{293 \text{ K}} \\ &= \mathbf{0.745 \text{ kW/K}} \end{aligned}$$

Note that the pressure of air remains unchanged as it flows in the heat exchanger. This is why the pressure term is not included in the entropy change expression of air.

8-154 Oxygen is cooled as it flows in an insulated pipe. The rate of entropy generation in the pipe is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The pipe is well-insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies are negligible. 4 Oxygen is an ideal gas with constant specific heats.

Properties The properties of oxygen at room temperature are $R = 0.2598 \text{ kJ/kg}\cdot\text{K}$, $c_p = 0.918 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis The rate of entropy generation in the pipe is determined by applying the rate form of the entropy balance on the pipe:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\text{0 (steady)}}{=} 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

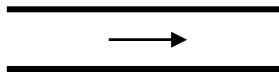
$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

Oxygen

240 kPa

20°C

70 m/s



200 kPa

18°C

The specific volume of oxygen at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.2598 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})}{240 \text{ kPa}} = 0.3172 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{\pi D^2 V_1}{4 \nu_1} = \frac{\pi (0.12 \text{ m})^2 (70 \text{ m/s})}{4 (0.3172 \text{ m}^3/\text{kg})} = 2.496 \text{ kg/s}$$

Substituting into the entropy balance relation,

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}(s_2 - s_1) \\ &= \dot{m} \left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) \\ &= (2.496 \text{ kg/s}) \left[(0.918 \text{ kJ/kg} \cdot \text{K}) \ln \frac{291 \text{ K}}{293 \text{ K}} - (0.2598 \text{ kJ/kg} \cdot \text{K}) \ln \frac{200 \text{ kPa}}{240 \text{ kPa}} \right] \\ &= \mathbf{0.1025 \text{ kW/K}} \end{aligned}$$

8-155 Nitrogen is compressed by an adiabatic compressor. The entropy generation for this process is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The compressor is well-insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies are negligible. 4 Nitrogen is an ideal gas with constant specific heats.

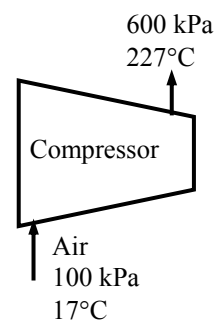
Properties The specific heat of nitrogen at the average temperature of $(17+227)/2=122^\circ\text{C} = 395\text{ K}$ is $c_p = 1.044\text{ kJ/kg}\cdot\text{K}$ (Table A-2b).

Analysis The rate of entropy generation in the pipe is determined by applying the rate form of the entropy balance on the compressor:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta\dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \quad \text{?0 (steady)}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$



Substituting per unit mass of the oxygen,

$$\begin{aligned} s_{\text{gen}} &= s_2 - s_1 \\ &= c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \\ &= (1.044\text{ kJ/kg}\cdot\text{K}) \ln \frac{(227+273)\text{ K}}{(17+273)\text{ K}} - (0.2968\text{ kJ/kg}\cdot\text{K}) \ln \frac{600\text{ kPa}}{100\text{ kPa}} \\ &= \mathbf{0.0369\text{ kJ/kg}\cdot\text{K}} \end{aligned}$$

8-156E Steam is condensed by cooling water in a condenser. The rate of heat transfer and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy and entropy of vaporization of water at 120°F are 1025.2 Btu/lbm and $s_{fg} = 1.7686$ Btu/lbm.R (Table A-4E).

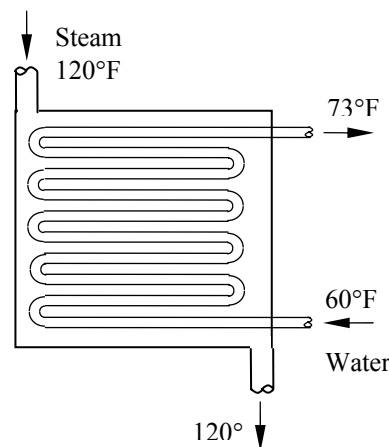
Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\approx 0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (92 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73^\circ\text{F} - 60^\circ\text{F}) = \mathbf{1196 \text{ Btu/s}}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1196 \text{ Btu/s}}{1025.2 \text{ Btu/lbm}} = 1.167 \text{ lbm/s}$$

(b) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{gen}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{system}}_{\text{Rate of change of entropy}} \overset{\approx 0 \text{ (steady)}}{=}$$

$$\dot{m}_1s_1 + \dot{m}_3s_3 - \dot{m}_2s_2 - \dot{m}_4s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}}s_1 + \dot{m}_{\text{steam}}s_3 - \dot{m}_{\text{water}}s_2 - \dot{m}_{\text{steam}}s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{\text{water}}c_p \ln \frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}}c_p \ln \frac{T_2}{T_1} - \dot{m}_{\text{steam}}s_{fg}$$

$$= (92 \text{ lbm/s})(1.0 \text{ Btu/lbm.R}) \ln \frac{73 + 460}{60 + 460} - (1.167 \text{ lbm/s})(17686 \text{ Btu/lbm.R})$$

$$= \mathbf{0.209 \text{ Btu/s.R}}$$

8-157 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year and the annual reduction in the rate of entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The properties of the milk are constant.

Properties The average density and specific heat of milk can be taken to be $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1 \text{ kg/L}$ and $c_{p, \text{milk}} = 3.79 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-3).

Analysis The mass flow rate of the milk is

$$\dot{m}_{\text{milk}} = \rho \dot{V}_{\text{milk}} = (1 \text{ kg/L})(12 \text{ L/s}) = 12 \text{ kg/s} = 43,200 \text{ kg/h}$$

Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{milk}} c_p (T_2 - T_1)$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{\text{current}} = [\dot{m} c_p (T_{\text{pasteurization}} - T_{\text{refrigeration}})]_{\text{milk}} = (12 \text{ kg/s})(3.79 \text{ kJ/kg} \cdot ^\circ\text{C})(72 - 4)^\circ\text{C} = 3093 \text{ kJ/s}$$

The proposed regenerator has an effectiveness of $\varepsilon = 0.82$, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{\text{saved}} = \varepsilon \dot{Q}_{\text{current}} = (0.82)(3093 \text{ kJ/s}) = 2536 \text{ kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{\text{boiler}} = 0.82$, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{\text{saved}}}{\eta_{\text{boiler}}} = \frac{(2536 \text{ kJ/s})}{(0.82)} \frac{(1 \text{ therm})}{(105,500 \text{ kJ})} = 0.02931 \text{ therm/s}$$

Noting that 1 year = 365 × 24 = 8760 h and unit cost of natural gas is \$0.52/therm, the annual fuel and money savings will be

$$\text{Fuel Saved} = (0.02931 \text{ therms/s})(8760 \times 3600 \text{ s}) = \mathbf{924,450 \text{ therms/yr}}$$

$$\text{Money saved} = (\text{Fuel saved})(\text{Unit cost of fuel}) = (924,450 \text{ therm/yr})(\$1.04/\text{therm}) = \mathbf{\$961,400/\text{yr}}$$

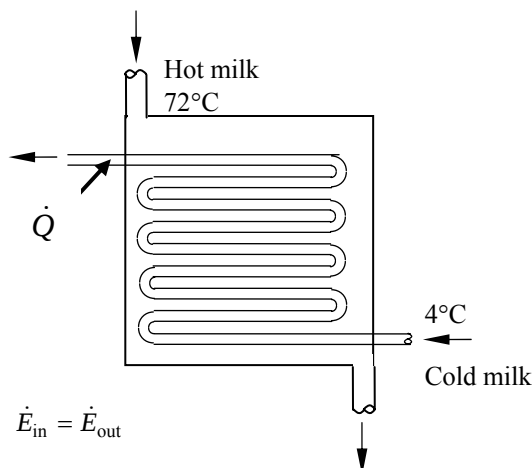
The rate of entropy generation during this process is determined by applying the rate form of the entropy balance on an *extended system* that includes the regenerator and the immediate surroundings so that the boundary temperature is the surroundings temperature, which we take to be the cold water temperature of 18°C:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \stackrel{\text{no (steady)}}{=} 0 \rightarrow \dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$$

Disregarding entropy transfer associated with fuel flow, the only significant difference between the two cases is the reduction in the entropy transfer to water due to the reduction in heat transfer to water, and is determined to be

$$\dot{S}_{\text{gen, reduction}} = \dot{S}_{\text{out, reduction}} = \frac{\dot{Q}_{\text{out, reduction}}}{T_{\text{surr}}} = \frac{\dot{Q}_{\text{saved}}}{T_{\text{surr}}} = \frac{2536 \text{ kJ/s}}{18 + 273} = 8.715 \text{ kW/K}$$

$$S_{\text{gen, reduction}} = \dot{S}_{\text{gen, reduction}} \Delta t = (8.715 \text{ kJ/s} \cdot \text{K})(8760 \times 3600 \text{ s/year}) = \mathbf{2.75 \times 10^8 \text{ kJ/K (per year)}}$$



8-158 Stainless steel ball bearings leaving the oven at a uniform temperature of 900°C at a rate of 1400 /min are exposed to air and are cooled to 850°C before they are dropped into the water for quenching. The rate of heat transfer from the ball to the air and the rate of entropy generation due to this heat transfer are to be determined.

Assumptions 1 The thermal properties of the bearing balls are constant. 2 The kinetic and potential energy changes of the balls are negligible. 3 The balls are at a uniform temperature at the end of the process

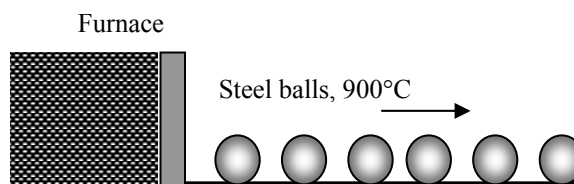
Properties The density and specific heat of the ball bearings are given to be $\rho = 8085 \text{ kg/m}^3$ and $c_p = 0.480 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis (a) We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8085 \text{ kg/m}^3) \frac{\pi (0.012 \text{ m})^3}{6} = 0.007315 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.007315 \text{ kg})(0.480 \text{ kJ/kg} \cdot ^\circ\text{C})(900 - 850)^\circ\text{C} = 0.1756 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{out (per ball)}} = (1400 \text{ balls/min}) \times (0.1756 \text{ kJ/ball}) = \mathbf{245.8 \text{ kJ/min} = 4.10 \text{ kW}}$$

Therefore, heat is lost to the air at a rate of 4.10 kW.

(b) We again take a single bearing ball as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the ball and its immediate surroundings so that the boundary temperature of the extended system is at 30°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\text{Net entropy transfer by heat and mass}} + \underbrace{S_{\text{gen}}}_{\text{Entropy generation}} = \underbrace{\Delta S_{\text{system}}}_{\text{Change in entropy}}$$

$$-\frac{Q_{\text{out}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (0.007315 \text{ kg})(0.480 \text{ kJ/kg} \cdot \text{K}) \ln \frac{850 + 273}{900 + 273} = -0.0001530 \text{ kJ/K}$$

Substituting,

$$S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}} = \frac{0.1756 \text{ kJ}}{303 \text{ K}} - 0.0001530 \text{ kJ/K} = 0.0004265 \text{ kJ/K (per ball)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}} \dot{n}_{\text{ball}} = (0.0004265 \text{ kJ/K} \cdot \text{ball})(1400 \text{ balls/min}) = 0.597 \text{ kJ/min} \cdot \text{K} = \mathbf{0.00995 \text{ kW/K}}$$

8-159 An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked and the amount of entropy generation associated with this heat transfer process are to be determined.

Assumptions 1 The egg is spherical in shape with a radius of $r_0 = 2.75$ cm. 2 The thermal properties of the egg are constant. 3 Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. 4 There are no changes in kinetic and potential energies.

Properties The density and specific heat of the egg are given to be $\rho = 1020$ kg/m³ and $c_p = 3.32$ kJ/kg·°C.

Analysis We take the egg as the system. This is a closed system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

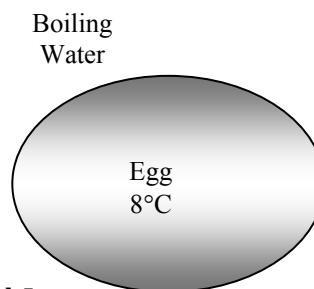
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{egg}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020 \text{ kg/m}^3) \frac{\pi (0.055 \text{ m})^3}{6} = 0.0889 \text{ kg}$$

$$Q_{\text{in}} = mc_p(T_2 - T_1) = (0.0889 \text{ kg})(3.32 \text{ kJ/kg} \cdot ^\circ\text{C})(70 - 8)^\circ\text{C} = \mathbf{18.3 \text{ kJ}}$$



We again take a single egg as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the egg and its immediate surroundings so that the boundary temperature of the extended system is at 97°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (0.0889 \text{ kg})(3.32 \text{ kJ/kg} \cdot \text{K}) \ln \frac{70 + 273}{8 + 273} = 0.0588 \text{ kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{18.3 \text{ kJ}}{370 \text{ K}} + 0.0588 \text{ kJ/K} = \mathbf{0.00934 \text{ kJ/K}} \quad (\text{per egg})$$

8-160 Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven and the rate of entropy generation associated with this heat transfer process are to be determined.

Assumptions 1 The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

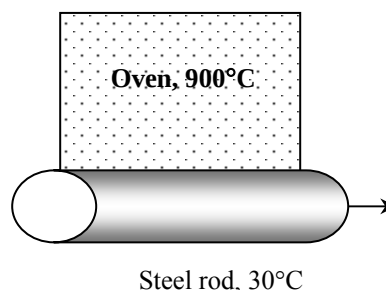
Properties The density and specific heat of the steel rods are given to be $\rho = 7833 \text{ kg/m}^3$ and $c_p = 0.465 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis (a) Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho V = \rho LA = \rho L(\pi D^2 / 4) \\ = (7833 \text{ kg/m}^3)(3 \text{ m})[\pi(0.1 \text{ m})^2 / 4] = 184.6 \text{ kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\ Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6 \text{ kg})(0.465 \text{ kJ/kg} \cdot ^\circ\text{C})(700 - 30)^\circ\text{C} = 57,512 \text{ kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}} / \Delta t = (57,512 \text{ kJ}) / (1 \text{ min}) = 57,512 \text{ kJ/min} = \mathbf{958.5 \text{ kW}}$$

(b) We again take the 3-m long section of the rod as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the rod and its immediate surroundings so that the boundary temperature of the extended system is at 900°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}} \\ \frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (184.6 \text{ kg})(0.465 \text{ kJ/kg} \cdot \text{K}) \ln \frac{700 + 273}{30 + 273} = 100.1 \text{ kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{57,512 \text{ kJ}}{900 + 273 \text{ K}} + 100.1 \text{ kJ/K} = 51.1 \text{ kJ/K}$$

Noting that this much entropy is generated in 1 min, the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = \frac{S_{\text{gen}}}{\Delta t} = \frac{51.1 \text{ kJ/K}}{1 \text{ min}} = 51.1 \text{ kJ/min} \cdot \text{K} = \mathbf{0.85 \text{ kW/K}}$$

8-161 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of entropy generation within the wall is to be determined.

Assumptions Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values.

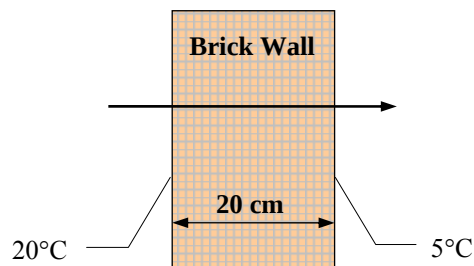
Analysis We take the wall to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for the wall simplifies to

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\text{Steady}}{=} 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,wall}} = 0$$

$$\frac{1890 \text{ W}}{293 \text{ K}} - \frac{1890 \text{ W}}{278 \text{ K}} + \dot{S}_{\text{gen,wall}} = 0$$

$$\dot{S}_{\text{gen,wall}} = \mathbf{0.348 \text{ W/K}}$$



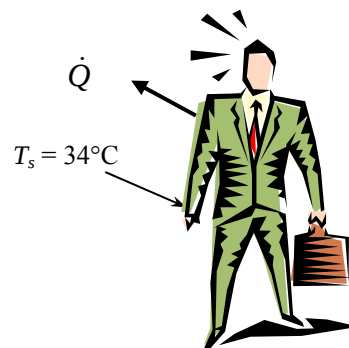
Therefore, the rate of entropy generation in the wall is 0.348 W/K.

8-162 A person is standing in a room at a specified temperature. The rate of entropy transfer from the body with heat is to be determined.

Assumptions Steady operating conditions exist.

Analysis Noting that Q/T represents entropy transfer with heat, the rate of entropy transfer from the body of the person accompanying heat transfer is

$$\dot{S}_{\text{transfer}} = \frac{\dot{Q}}{T} = \frac{336 \text{ W}}{307 \text{ K}} = \mathbf{1.094 \text{ W/K}}$$



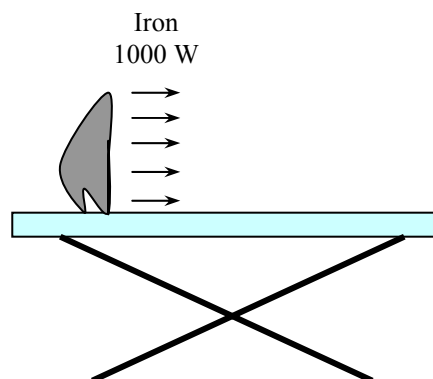
8-163 A 1000-W iron is left on the iron board with its base exposed to the air at 20°C. The rate of entropy generation is to be determined in steady operation.

Assumptions Steady operating conditions exist.

Analysis We take the iron to be the system, which is a closed system. Considering that the iron experiences no change in its properties in steady operation, including its entropy, the rate form of the entropy balance for the iron simplifies to

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\text{st}}{=} 0$$

$$-\frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,iron}} = 0$$



Therefore,

$$\dot{S}_{\text{gen,iron}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} = \frac{1000 \text{ W}}{673 \text{ K}} = \mathbf{1.486 \text{ W/K}}$$

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the iron and its immediate surroundings so that the boundary temperature of the extended system is at 20°C at all times. It gives

$$\dot{S}_{\text{gen,total}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} = \frac{\dot{Q}}{T_{\text{surr}}} = \frac{1000 \text{ W}}{293 \text{ K}} = \mathbf{3.413 \text{ W/K}}$$

Discussion Note that only about one-third of the entropy generation occurs within the iron. The rest occurs in the air surrounding the iron as the temperature drops from 400°C to 20°C without serving any useful purpose.

8-164E A cylinder contains saturated liquid water at a specified pressure. Heat is transferred to liquid from a source and some liquid evaporates. The total entropy generation during this process is to be determined.

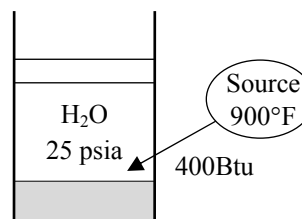
Assumptions 1 No heat loss occurs from the water to the surroundings during the process. 2 The pressure inside the cylinder and thus the water temperature remains constant during the process. 3 No irreversibilities occur within the cylinder during the process.

Analysis The pressure of the steam is maintained constant. Therefore, the temperature of the steam remains constant also at

$$T = T_{\text{sat}@25 \text{ psia}} = 240.03^\circ\text{F} = 700 \text{ R} \quad (\text{Table A-5E})$$

Taking the contents of the cylinder as the system and noting that the temperature of water remains constant, the entropy change of the system during this isothermal, internally reversible process becomes

$$\Delta S_{\text{system}} = \frac{Q_{\text{sys,in}}}{T_{\text{sys}}} = \frac{400 \text{ Btu}}{700 \text{ R}} = 0.572 \text{ Btu/R}$$



Similarly, the entropy change of the heat source is determined from

$$\Delta S_{\text{source}} = -\frac{Q_{\text{source,out}}}{T_{\text{source}}} = -\frac{400 \text{ Btu}}{900 + 460\text{R}} = -0.294 \text{ Btu/R}$$

Now consider a combined system that includes the cylinder and the source. Noting that no heat or mass crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen,total}} = \Delta S_{\text{water}} + \Delta S_{\text{source}}$$

Therefore, the total entropy generated during this process is

$$S_{\text{gen,total}} = \Delta S_{\text{water}} + \Delta S_{\text{source}} = 0.572 - 0.294 = \mathbf{0.278 \text{ Btu/R}}$$

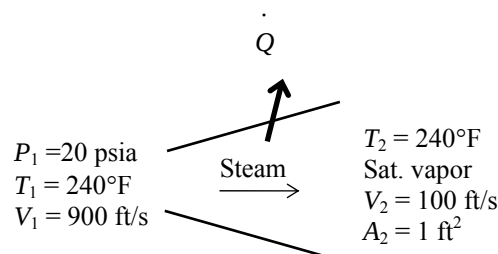
Discussion The entropy generation in this case is entirely due to the irreversible heat transfer through a finite temperature difference. We could also determine the total entropy generation by writing an energy balance on an extended system that includes the system and its immediate surroundings so that part of the boundary of the extended system, where heat transfer occurs, is at the source temperature.

8-165E Steam is decelerated in a diffuser from a velocity of 900 ft/s to 100 ft/s. The mass flow rate of steam and the rate of entropy generation are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

Properties The properties of steam at the inlet and the exit of the diffuser are (Tables A-4E through A-6E)

$$\begin{aligned} P_1 = 20 \text{ psia} & \left\{ \begin{array}{l} h_1 = 1162.3 \text{ Btu/lbm} \\ s_1 = 1.7406 \text{ Btu/lbm} \cdot \text{R} \end{array} \right. \\ T_1 = 240^\circ\text{F} & \\ T_2 = 240^\circ\text{F} & \left\{ \begin{array}{l} h_2 = 1160.5 \text{ Btu/lbm} \\ s_2 = 1.7141 \text{ Btu/lbm} \cdot \text{R} \end{array} \right. \\ \text{sat. vapor} & \left\{ \begin{array}{l} v_2 = 16.316 \text{ ft}^3/\text{lbm} \end{array} \right. \end{aligned}$$



Analysis (a) The mass flow rate of the steam can be determined from its definition to be

$$\dot{m} = \frac{1}{v_2} A_2 V_2 = \frac{1}{16.316 \text{ ft}^3/\text{lbm}} (1 \text{ ft}^2) (100 \text{ ft/s}) = \mathbf{6.129 \text{ lbm/s}}$$

(b) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) - \dot{Q}_{\text{out}} = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = -\dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting, the rate of heat loss from the diffuser is determined to be

$$\dot{Q}_{\text{out}} = -(6.129 \text{ lbm/s}) \left(1160.5 - 1162.3 + \frac{(100 \text{ ft/s})^2 - (900 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) \right) = 108.42 \text{ Btu/s}$$

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the diffuser and its immediate surroundings so that the boundary temperature of the extended system is 77°F at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \stackrel{\text{no}}{=} 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the total rate of entropy generation during this process becomes

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} = (6.129 \text{ lbm/s})(1.7141 - 1.7406) \text{ Btu/lbm} \cdot \text{R} + \frac{108.42 \text{ Btu/s}}{537 \text{ R}} = \mathbf{0.0395 \text{ Btu/s} \cdot \text{R}}$$

8-166 Steam expands in a turbine from a specified state to another specified state. The rate of entropy generation during this process is to be determined.

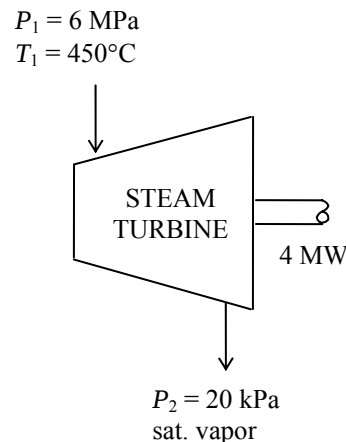
Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\begin{aligned} P_1 = 6 \text{ MPa} & \left\{ \begin{array}{l} h_1 = 3302.9 \text{ kJ/kg} \\ T_1 = 450^\circ\text{C} \end{array} \right. & \left\{ \begin{array}{l} s_1 = 6.7219 \text{ kJ/kg} \cdot \text{K} \\ h_2 = 2608.9 \text{ kJ/kg} \end{array} \right. \\ P_2 = 20 \text{ kPa} & \left\{ \begin{array}{l} h_2 = 2608.9 \text{ kJ/kg} \\ \text{sat. vapor} \end{array} \right. & \left\{ \begin{array}{l} s_2 = 7.9073 \text{ kJ/kg} \cdot \text{K} \end{array} \right. \end{aligned}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\begin{aligned} \underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} &= \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0 \\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}h_1 &= \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \dot{m}h_2 \\ \dot{Q}_{\text{out}} &= \dot{m}(h_1 - h_2) - \dot{W}_{\text{out}} \end{aligned}$$



Substituting,

$$\dot{Q}_{\text{out}} = (25,000/3600 \text{ kg/s})(3302.9 - 2608.9) \text{ kJ/kg} - 4000 \text{ kJ/s} = 819.3 \text{ kJ/s}$$

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the turbine and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

$$\begin{aligned} \underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} &= \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \stackrel{\text{no}}{=} 0 \\ \dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} &= 0 \end{aligned}$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} = (25,000/3600 \text{ kg/s})(7.9073 - 6.7219) \text{ kJ/kg} \cdot \text{K} + \frac{819.3 \text{ kW}}{298 \text{ K}} = \mathbf{11.0 \text{ kW/K}}$$

8-167 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water stream and the rate of entropy generation are to be determined.

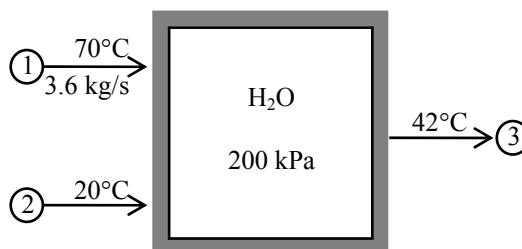
Assumptions 1 Steady operating conditions exist. 2 The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies of fluid streams are negligible.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.21^\circ\text{C}$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus from Table A-4,

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 70^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 \cong h_{f@70^\circ\text{C}} = 293.07 \text{ kJ/kg} \\ s_1 \cong s_{f@70^\circ\text{C}} = 0.9551 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 20^\circ\text{C} \end{array} \right\} \begin{array}{l} h_2 \cong h_{f@20^\circ\text{C}} = 83.91 \text{ kJ/kg} \\ s_2 \cong s_{f@20^\circ\text{C}} = 0.2965 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_3 = 200 \text{ kPa} \\ T_3 = 42^\circ\text{C} \end{array} \right\} \begin{array}{l} h_3 \cong h_{f@42^\circ\text{C}} = 175.90 \text{ kJ/kg} \\ s_3 \cong s_{f@42^\circ\text{C}} = 0.5990 \text{ kJ/kg} \cdot \text{K} \end{array}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\text{net}} \stackrel{\text{no (steady)}}{=} 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\text{net}} \stackrel{\text{no (steady)}}{=}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two relations gives $\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2} \dot{m}_1 = \frac{(293.07 - 175.90) \text{ kJ/kg}}{(175.90 - 83.91) \text{ kJ/kg}} (3.6 \text{ kg/s}) = \mathbf{4.586 \text{ kg/s}}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 3.6 + 4.586 = 8.186 \text{ kg/s}$

(b) Noting that the mixing chamber is adiabatic and thus there is no heat transfer to the surroundings, the entropy balance of the steady-flow system (the mixing chamber) can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\text{net}} \stackrel{\text{no}}{=}}_{\text{Rate of change of entropy}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0$$

Substituting, the total rate of entropy generation during this process becomes

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 \\ &= (8.186 \text{ kg/s})(0.5990 \text{ kJ/kg} \cdot \text{K}) - (4.586 \text{ kg/s})(0.2965 \text{ kJ/kg} \cdot \text{K}) - (3.6 \text{ kg/s})(0.9551 \text{ kJ/kg} \cdot \text{K}) \\ &= \mathbf{0.1054 \text{ kW/K}} \end{aligned}$$

8-168 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of entropy generation are to be determined.

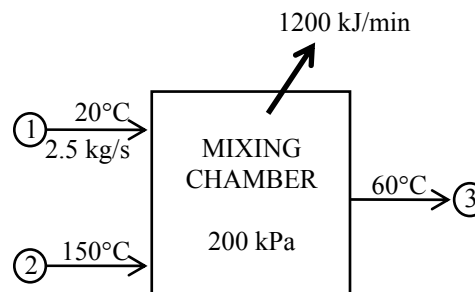
Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.21^\circ\text{C}$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 20^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 \cong h_{f@20^\circ\text{C}} = 83.91 \text{ kJ/kg} \\ s_1 \cong s_{f@20^\circ\text{C}} = 0.2965 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 150^\circ\text{C} \end{array} \right\} \begin{array}{l} h_2 = 2769.1 \text{ kJ/kg} \\ s_2 = 7.2810 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_3 = 200 \text{ kPa} \\ T_3 = 60^\circ\text{C} \end{array} \right\} \begin{array}{l} h_3 \cong h_{f@60^\circ\text{C}} = 251.18 \text{ kJ/kg} \\ s_3 \cong s_{f@60^\circ\text{C}} = 0.8313 \text{ kJ/kg} \cdot \text{K} \end{array}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

$$\text{Mass balance: } \dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\approx 0 \text{ (steady)}}{=} 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\approx 0 \text{ (steady)}}{=}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3 = \dot{m}_1 (h_1 - h_3) + \dot{m}_2 (h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1 (h_1 - h_3)}{h_2 - h_3} = \frac{(1200/60 \text{ kJ/s}) - (2.5 \text{ kg/s})(83.91 - 251.18) \text{ kJ/kg}}{(2769.1 - 251.18) \text{ kJ/kg}} = \mathbf{0.166 \text{ kg/s}}$$

$$\text{Also, } \dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.166 = 2.666 \text{ kg/s}$$

(b) The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}} \overset{\approx 0}{=}}_{\text{Rate of change of entropy}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} \\ &= (2.666 \text{ kg/s})(0.8313 \text{ kJ/kg} \cdot \text{K}) - (0.166 \text{ kg/s})(7.2810 \text{ kJ/kg} \cdot \text{K}) \\ &\quad - (2.5 \text{ kg/s})(0.2965 \text{ kJ/kg} \cdot \text{K}) + \frac{(1200/60 \text{ kJ/s})}{298 \text{ K}} \\ &= \mathbf{0.333 \text{ kW/K}} \end{aligned}$$

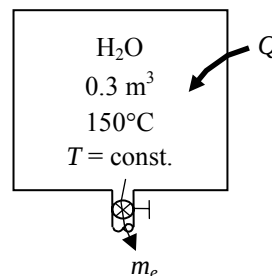
8-169 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer and the entropy generation during this process are to be determined.

Assumptions 1 This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$T_1 = 150^\circ\text{C} \quad \left\{ \begin{array}{l} \nu_1 = \nu_{f@150^\circ\text{C}} = 0.001091 \text{ m}^3/\text{kg} \\ u_1 = u_{f@150^\circ\text{C}} = 631.66 \text{ kJ/kg} \\ s_1 = s_{f@150^\circ\text{C}} = 1.8418 \text{ kJ/kg} \cdot \text{K} \end{array} \right.$$

$$T_e = 150^\circ\text{C} \quad \left\{ \begin{array}{l} h_e = h_{f@150^\circ\text{C}} = 632.18 \text{ kJ/kg} \\ s_e = s_{f@150^\circ\text{C}} = 1.8418 \text{ kJ/kg} \cdot \text{K} \end{array} \right.$$



Analysis (a) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u , respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{\nu}{\nu_1} = \frac{0.3 \text{ m}^3}{0.001091 \text{ m}^3/\text{kg}} = 275.10 \text{ kg}$$

$$m_2 = \frac{1}{2} m_1 = \frac{1}{2} (275.10 \text{ kg}) = 137.55 \text{ kg} = m_e$$

Now we determine the final internal energy and entropy,

$$\nu_2 = \frac{\nu}{m_2} = \frac{0.3 \text{ m}^3}{137.75 \text{ kg}} = 0.002181 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{\nu_2 - \nu_f}{\nu_{fg}} = \frac{0.002181 - 0.001091}{0.39248 - 0.001091} = 0.002786$$

$$T_2 = 150^\circ\text{C} \quad \left\{ \begin{array}{l} u_2 = u_f + x_2 u_{fg} = 631.66 + (0.002786)(1927.4) = 637.03 \text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 1.8418 + (0.002786)(4.9953) = 1.8557 \text{ kJ/kg} \cdot \text{K} \end{array} \right.$$

The heat transfer during this process is determined by substituting these values into the energy balance equation,

$$\begin{aligned}
 Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\
 &= (137.55 \text{ kg})(632.18 \text{ kJ/kg}) + (137.55 \text{ kg})(637.03 \text{ kJ/kg}) - (275.10 \text{ kg})(631.66 \text{ kJ/kg}) \\
 &= \mathbf{810 \text{ kJ}}
 \end{aligned}$$

(b) The total entropy generation is determined by considering a combined system that includes the tank and the heat source. Noting that no heat crosses the boundaries of this combined system and no mass enters, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}} \longrightarrow -m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} + \Delta S_{\text{source}}$$

Therefore, the total entropy generated during this process is

$$\begin{aligned}
 S_{\text{gen}} &= m_e s_e + \Delta S_{\text{tank}} + \Delta S_{\text{source}} = m_e s_e + (m_2 s_2 - m_1 s_1) - \frac{Q_{\text{source, out}}}{T_{\text{source}}} \\
 &= (137.55 \text{ kg})(1.8418 \text{ kJ/kg} \cdot \text{K}) + (137.55 \text{ kg})(1.8557 \text{ kJ/kg} \cdot \text{K}) \\
 &\quad - (275.10 \text{ kg})(1.8418 \text{ kJ/kg} \cdot \text{K}) - \frac{810 \text{ kJ}}{423 \text{ K}} \\
 &= \mathbf{0.202 \text{ kJ/K}}
 \end{aligned}$$

8-170E An unknown mass of iron is dropped into water in an insulated tank while being stirred by a 200-W paddle wheel. Thermal equilibrium is established after 10 min. The mass of the iron block and the entropy generated during this process are to be determined.

Assumptions 1 Both the water and the iron block are incompressible substances with constant specific heats at room temperature. 2 The system is stationary and thus the kinetic and potential energy changes are zero. 3 The system is well-insulated and thus there is no heat transfer.

Properties The specific heats of water and the iron block at room temperature are $c_{p, \text{water}} = 1.00 \text{ Btu/lbm} \cdot ^\circ\text{F}$ and $c_{p, \text{iron}} = 0.107 \text{ Btu/lbm} \cdot ^\circ\text{F}$ (Table A-3E). The density of water at room temperature is 62.1 lbm/ft^3 .

Analysis (a) We take the entire contents of the tank, water + iron block, as the system. This is a closed system since no mass crosses the system boundary during the process. The energy balance on the system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{pw, in}} = \Delta U$$

or,
$$W_{\text{pw, in}} = \Delta U_{\text{iron}} + \Delta U_{\text{water}}$$

$$W_{\text{pw, in}} = [mc(T_2 - T_1)]_{\text{iron}} + [mc(T_2 - T_1)]_{\text{water}}$$

where

$$m_{\text{water}} = \rho V = (62.1 \text{ lbm/ft}^3)(0.8 \text{ ft}^3) = 49.7 \text{ lbm}$$

$$W_{\text{pw}} = \dot{W}_{\text{pw}} \Delta t = (0.2 \text{ kJ/s})(10 \times 60 \text{ s}) \left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}} \right) = 113.7 \text{ Btu}$$

Using specific heat values for iron and liquid water and substituting,

$$113.7 \text{ Btu} = m_{\text{iron}}(0.107 \text{ Btu/lbm} \cdot ^\circ\text{F})(75 - 185)^\circ\text{F} + (49.7 \text{ lbm})(1.00 \text{ Btu/lbm} \cdot ^\circ\text{F})(75 - 70)^\circ\text{F}$$

$$m_{\text{iron}} = \mathbf{11.4 \text{ lbm}}$$

(b) Again we take the iron + water in the tank to be the system. Noting that no heat or mass crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen, total}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}}$$

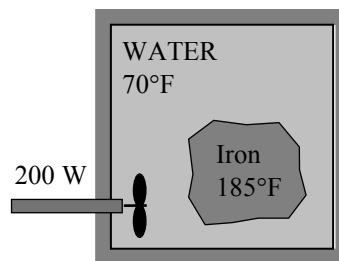
where

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln \left(\frac{T_2}{T_1} \right) = (11.4 \text{ lbm})(0.107 \text{ Btu/lbm} \cdot \text{R}) \ln \left(\frac{535 \text{ R}}{645 \text{ R}} \right) = -0.228 \text{ Btu/R}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln \left(\frac{T_2}{T_1} \right) = (49.6 \text{ lbm})(1.0 \text{ Btu/lbm} \cdot \text{R}) \ln \left(\frac{535 \text{ R}}{530 \text{ R}} \right) = 0.466 \text{ Btu/R}$$

Therefore, the entropy generated during this process is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}} = -0.228 + 0.466 = \mathbf{0.238 \text{ Btu/R}}$$



8-171E Air is compressed steadily by a compressor. The mass flow rate of air through the compressor and the rate of entropy generation are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is 0.06855 Btu/lbm·R (Table A-1E). The inlet and exit enthalpies of air are (Table A-21E)

$$\left. \begin{array}{l} T_1 = 520 \text{ R} \\ P_1 = 15 \text{ psia} \end{array} \right\} \begin{array}{l} h_1 = 124.27 \text{ Btu/lbm} \\ s_1^\circ = 0.59173 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left. \begin{array}{l} T_2 = 1080 \text{ R} \\ P_2 = 150 \text{ psia} \end{array} \right\} \begin{array}{l} h_2 = 260.97 \text{ Btu/lbm} \\ s_2^\circ = 0.76964 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

Analysis (a) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \Delta \text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting, the mass flow rate is determined to be

$$\text{Thus,} \quad (400 \text{ hp}) \left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}} \right) - \frac{1500 \text{ Btu}}{60 \text{ s}} = \dot{m} \left(260.97 - 124.27 + \frac{(350 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) \right)$$

It yields $\dot{m} = 1.852 \text{ lbm/s}$

(b) Again we take the compressor to be the system. Noting that no heat or mass crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} \stackrel{\text{no}}{=} 0$$

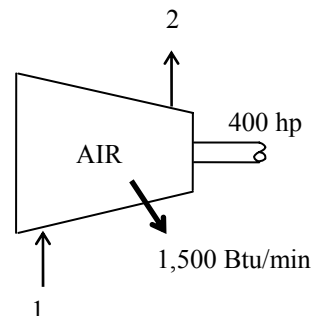
$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}}$$

where

$$\begin{aligned} \Delta \dot{S}_{\text{air}} &= \dot{m}(s_2 - s_1) = \dot{m} \left(s_2^\circ - s_1^\circ - R \ln \frac{P_2}{P_1} \right) \\ &= (1.852 \text{ lbm/s}) \left(0.76964 - 0.59173 - (0.06855 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{150 \text{ psia}}{15 \text{ psia}} \right) = 0.0372 \text{ Btu/s} \cdot \text{R} \end{aligned}$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} = 0.0372 \text{ Btu/s} \cdot \text{R} + \frac{1500/60 \text{ Btu/s}}{520 \text{ R}} = \mathbf{0.0853 \text{ Btu/s} \cdot \text{R}}$$

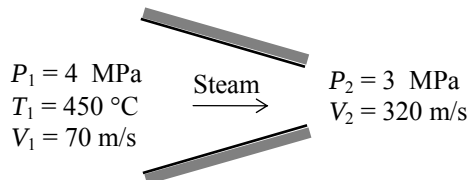


8-172 Steam is accelerated in a nozzle from a velocity of 70 m/s to 320 m/s. The exit temperature and the rate of entropy generation are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Potential energy changes are negligible. 3 There are no work interactions. 4 The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6),

$$\left. \begin{array}{l} P_1 = 4 \text{ MPa} \\ T_1 = 450^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 3331.2 \text{ kJ/kg} \\ s_1 = 6.9386 \text{ kJ/kg} \cdot \text{K} \\ \nu_1 = 0.08004 \text{ m}^3/\text{kg} \end{array}$$



Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,,

$$\text{or,} \quad h_2 = 3331.2 \text{ kJ/kg} - \frac{(320 \text{ m/s})^2 - (70 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 3282.4 \text{ kJ/kg}$$

Thus,

$$\left. \begin{array}{l} P_2 = 3 \text{ MPa} \\ h_{2a} = 3282.4 \text{ kJ/kg} \end{array} \right\} \begin{array}{l} T_2 = \mathbf{422.3^\circ\text{C}} \\ s_2 = 6.9976 \text{ kJ/kg} \cdot \text{K} \end{array}$$

The mass flow rate of steam is

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 = \frac{1}{0.08004 \text{ m}^3/\text{kg}} (7 \times 10^{-4} \text{ m}^2) (70 \text{ m/s}) = 0.6122 \text{ kg/s}$$

(b) Again we take the nozzle to be the system. Noting that no heat crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \stackrel{\text{no}}{=} 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) = (0.6122 \text{ kg/s})(6.9976 - 6.9386) \text{ kJ/kg} \cdot \text{K} = \mathbf{0.0361 \text{ kW/K}}$$

Review Problems

8-173E The source and sink temperatures and the thermal efficiency of a heat engine are given. The entropy change of the two reservoirs is to be calculated and it is to be determined if this engine satisfies the increase of entropy principle.

Assumptions The heat engine operates steadily.

Analysis According to the first law and the definition of the thermal efficiency,

$$Q_L = (1 - \eta)Q_H = (1 - 0.4)(1 \text{ Btu}) = 0.6 \text{ Btu}$$

when the thermal efficiency is 40%. The entropy change of everything involved in this process is then

$$\begin{aligned} \Delta S_{\text{total}} &= \Delta S_H + \Delta S_L \\ &= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-1 \text{ Btu}}{1300 \text{ R}} + \frac{0.6 \text{ Btu}}{500 \text{ R}} = \mathbf{0.000431 \text{ Btu/R}} \end{aligned}$$

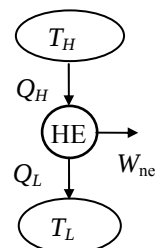
Since the entropy of everything has increased, this engine is possible. When the thermal efficiency of the engine is 70%,

$$Q_L = (1 - \eta)Q_H = (1 - 0.7)(1 \text{ Btu}) = 0.3 \text{ Btu}$$

The total entropy change is then

$$\begin{aligned} \Delta S_{\text{total}} &= \Delta S_H + \Delta S_L \\ &= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-1 \text{ Btu}}{1300 \text{ R}} + \frac{0.3 \text{ Btu}}{500 \text{ R}} = \mathbf{-0.000169 \text{ Btu/R}} \end{aligned}$$

which is a decrease in the entropy of everything involved with this engine. Therefore, this engine is now impossible.



8-174 The source and sink temperatures and the COP of a refrigerator are given. The total entropy change of the two reservoirs is to be calculated and it is to be determined if this refrigerator satisfies the second law.

Assumptions The refrigerator operates steadily.

Analysis Combining the first law and the definition of the coefficient of performance produces

$$Q_H = Q_L \left(1 + \frac{1}{\text{COP}_R} \right) = (1 \text{ kJ}) \left(1 + \frac{1}{4} \right) = 1.25 \text{ kJ}$$

when $\text{COP} = 4$. The entropy change of everything is then

$$\begin{aligned} \Delta S_{\text{total}} &= \Delta S_H + \Delta S_L \\ &= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{1.25 \text{ kJ}}{303 \text{ K}} + \frac{-1 \text{ kJ}}{253 \text{ K}} = \mathbf{0.000173 \text{ kJ/K}} \end{aligned}$$

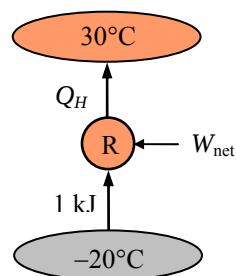
Since the entropy increases, a refrigerator with $\text{COP} = 4$ is possible. When the coefficient of performance is increased to 6,

$$Q_H = Q_L \left(1 + \frac{1}{\text{COP}_R} \right) = (1 \text{ kJ}) \left(1 + \frac{1}{6} \right) = 1.167 \text{ kJ}$$

and the net entropy change is

$$\begin{aligned} \Delta S_{\text{total}} &= \Delta S_H + \Delta S_L \\ &= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{1.167 \text{ kJ}}{303 \text{ K}} + \frac{-1 \text{ kJ}}{253 \text{ K}} = \mathbf{-0.000101 \text{ kJ/K}} \end{aligned}$$

and the refrigerator can no longer be possible.



8-175 The operating conditions and thermal reservoir temperatures of a heat pump are given. It is to be determined if the increase of entropy principle is satisfied.

Assumptions The heat pump operates steadily.

Analysis Applying the first law to the cyclic heat pump gives

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 25 \text{ kW} - 5 \text{ kW} = 20 \text{ kW}$$

According to the definition of the entropy, the rate at which the entropy of the high-temperature reservoir increases is

$$\Delta \dot{S}_H = \frac{\dot{Q}_H}{T_H} = \frac{25 \text{ kW}}{300 \text{ K}} = 0.0833 \text{ kW/K}$$

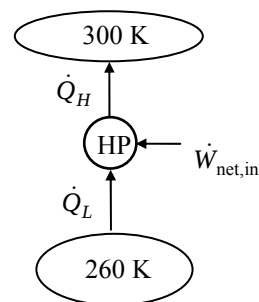
Similarly, the rate at which the entropy of the low-temperature reservoir decreases is

$$\Delta \dot{S}_L = \frac{\dot{Q}_L}{T_L} = \frac{-20 \text{ kW}}{260 \text{ K}} = -0.0769 \text{ kW/K}$$

The rate at which the entropy of everything changes is then

$$\Delta \dot{S}_{\text{total}} = \Delta \dot{S}_H + \Delta \dot{S}_L = 0.0833 - 0.0769 = 0.0064 \text{ kW/K}$$

which is positive and therefore it **satisfies** the increase in entropy principle.



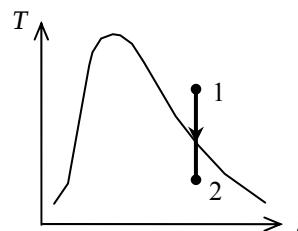
8-176 Steam is expanded adiabatically in a closed system. The minimum internal energy that can be achieved during this process is to be determined.

Analysis The entropy at the initial state is

$$\left. \begin{array}{l} P_1 = 1500 \text{ kPa} \\ T_1 = 320^\circ\text{C} \end{array} \right\} s_1 = 6.9957 \text{ kJ/kg} \cdot \text{K} \quad (\text{from Table A - 6 or from EES})$$

The internal energy will be minimum if the process is isentropic. Then,

$$\left. \begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 = 6.9957 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \begin{aligned} x_2 &= \frac{s_2 - s_f}{s_{fg}} = \frac{6.9957 - 1.3028}{6.0562} = 0.9400 \\ u_2 &= u_f + x_2 u_{fg} = 417.40 + (0.9400)(2088.2) = \mathbf{2380.3 \text{ kJ/kg}} \end{aligned}$$



8-177E Water is expanded in an isothermal, reversible process. It is to be determined if the process is possible.

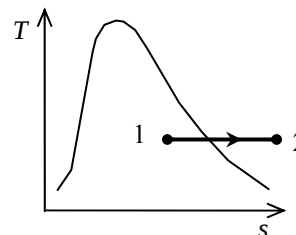
Analysis The entropies at the initial and final states are (Tables A-5E and A-6E)

$$\left. \begin{array}{l} P_1 = 30 \text{ psia} \\ x_1 = 0.70 \end{array} \right\} \begin{array}{l} T_1 = 250.3^\circ\text{F} \\ s_1 = s_f + x_1 s_{fg} = 0.36821 + (0.70)(1.33132) = 1.3001 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left. \begin{array}{l} P_2 = 10 \text{ psia} \\ T_2 = T_1 = 250.3^\circ\text{F} \end{array} \right\} s_2 = 1.8277 \text{ Btu/lbm} \cdot \text{R}$$

The heat transfer during this isothermal, reversible process is the area under the process line:

$$\begin{aligned} q &= T(s_2 - s_1) \\ &= (250 + 460 \text{ K})(1.8277 - 1.3001) \text{ Btu/lbm} \cdot \text{R} \\ &= 374.6 \text{ Btu/lbm} \end{aligned}$$



The total entropy change (i.e., entropy generation) is the sum of the entropy changes of water and the reservoir:

$$\begin{aligned} \Delta s_{\text{total}} &= \Delta s_{\text{water}} + \Delta s_{\text{R}} \\ &= s_2 - s_1 + \frac{-q}{T_{\text{R}}} \\ &= (1.8277 - 1.3001) \text{ Btu/lbm} \cdot \text{R} + \frac{-374.6 \text{ Btu/lbm}}{(300 + 460) \text{ R}} \\ &= \mathbf{0.0347 \text{ Btu/lbm} \cdot \text{R}} \end{aligned}$$

Note that the sign of heat transfer is with respect to the reservoir.

8-178E Air is compressed adiabatically in a closed system. It is to be determined if this process is possible.

Assumptions 1 Changes in the kinetic and potential energies are negligible. 4 Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$, $c_p = 0.240 \text{ Btu/lbm} \cdot \text{R}$ (Table A-2Ea).

Analysis The specific volume of air at the initial state is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(560 \text{ R})}{16 \text{ psia}} = 12.96 \text{ ft}^3/\text{lbm}$$

The volume at the final state will be minimum if the process is isentropic. The specific volume for this case is determined from the isentropic relation of an ideal gas to be

$$\nu_{2,\min} = \nu_1 \left(\frac{P_1}{P_2} \right)^{1/k} = (12.96 \text{ ft}^3/\text{lbm}) \left(\frac{16 \text{ psia}}{100 \text{ psia}} \right)^{1/1.4} = 3.500 \text{ ft}^3/\text{lbm}$$

and the minimum volume is

$$\nu_2 = m \nu_2 = (2 \text{ lbm})(3.500 \text{ ft}^3/\text{lbm}) = \mathbf{7.00 \text{ ft}^3}$$

which is greater than the proposed volume $4 \text{ ft}^3/\text{lbm}$. Hence, it is not possible to compress this air into $4 \text{ ft}^3/\text{lbm}$.

8-179 Oxygen is expanded adiabatically in a piston-cylinder device. The maximum volume is to be determined.

Assumptions 1 Changes in the kinetic and potential energies are negligible. 4 Oxygen is an ideal gas with constant specific heats.

Properties The gas constant of oxygen is $R = 0.2598 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$. The specific heat ratio at the room temperature is $k = 1.395$ (Table A-2a).

Analysis The volume of oxygen at the initial state is

$$\nu_1 = \frac{mRT_1}{P_1} = \frac{(3 \text{ kg})(0.2598 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(373 + 273 \text{ K})}{950 \text{ kPa}} = 0.5300 \text{ m}^3$$

The volume at the final state will be maximum if the process is isentropic. The volume for this case is determined from the isentropic relation of an ideal gas to be

$$\nu_{2,\max} = \nu_1 \left(\frac{P_1}{P_2} \right)^{1/k} = (0.5300 \text{ m}^3) \left(\frac{950 \text{ kPa}}{100 \text{ kPa}} \right)^{1/1.395} = \mathbf{2.66 \text{ m}^3}$$

8-180E A solid block is heated with saturated water vapor. The final temperature of the block and water, and the entropy changes of the block, water, and the entire system are to be determined.

Assumptions 1 The system is stationary and thus the kinetic and potential energy changes are zero. 2 There are no work interactions involved. 3 There is no heat transfer between the system and the surroundings.

Analysis (a) As the block is heated, some of the water vapor will be condensed. We will assume (will be checked later) that the water is a mixture of liquid and vapor at the end of the process. Based upon this assumption, the final temperature of the water and solid block is **212°F** (The saturation temperature at 14.7 psia). The heat picked up by the block is

$$Q_{\text{block}} = mc(T_2 - T_1) = (100 \text{ lbm})(0.5 \text{ Btu/lbm} \cdot \text{R})(212 - 70)\text{R} = 7100 \text{ Btu}$$

The water properties at the initial state are

$$\left. \begin{array}{l} P_1 = 14.7 \text{ psia} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} T_1 = 212^\circ\text{F} \\ h_1 = 1150.3 \text{ Btu/lbm} \\ s_1 = 1.7566 \text{ Btu/lbm} \cdot \text{R} \end{array} \quad (\text{Table A-5E})$$

The heat released by the water is equal to the heat picked up by the block. Also noting that the pressure of water remains constant, the enthalpy of water at the end of the heat exchange process is determined from

$$h_2 = h_1 - \frac{Q_{\text{water}}}{m_w} = 1150.3 \text{ Btu/lbm} - \frac{7100 \text{ Btu}}{10 \text{ lbm}} = 440.3 \text{ Btu/lbm}$$

The state of water at the final state is saturated mixture. Thus, our initial assumption was correct. The properties of water at the final state are

$$\left. \begin{array}{l} P_2 = 14.7 \text{ psia} \\ h_2 = 440.3 \text{ Btu/lbm} \end{array} \right\} \begin{array}{l} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{440.3 - 180.16}{970.12} = 0.2682 \\ s_2 = s_f + x_2 s_{fg} = 0.31215 + (0.2682)(1.44441) = 0.69947 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

The entropy change of the water is then

$$\Delta S_{\text{water}} = m_w(s_2 - s_1) = (10 \text{ lbm})(0.69947 - 1.7566) \text{ Btu/lbm} = \mathbf{-10.57 \text{ Btu/R}}$$

(b) The entropy change of the block is

$$\Delta S_{\text{block}} = mc \ln \frac{T_2}{T_1} = (100 \text{ lbm})(0.5 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{(212 + 460)\text{R}}{(70 + 460)\text{R}} = \mathbf{11.87 \text{ Btu/R}}$$

(c) The total entropy change is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{block}} = -10.57 + 11.87 = \mathbf{1.30 \text{ Btu/R}}$$

The positive result for the total entropy change (i.e., entropy generation) indicates that this process is possible.

8-181 Air is compressed in a piston-cylinder device. It is to be determined if this process is possible.

Assumptions 1 Changes in the kinetic and potential energies are negligible. 4 Air is an ideal gas with constant specific heats. 3 The compression process is reversible.

Properties The properties of air at room temperature are $R = 0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$, $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ (Table A-2a).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

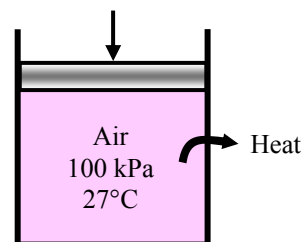
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$W_{b,\text{in}} - Q_{\text{out}} = mc_p(T_2 - T_1)$$

$$W_{b,\text{in}} - Q_{\text{out}} = 0 \quad (\text{since } T_2 = T_1)$$

$$Q_{\text{out}} = W_{b,\text{in}}$$



The work input for this isothermal, reversible process is

$$w_{\text{in}} = RT \ln \frac{P_2}{P_1} = (0.287 \text{ kJ/kg}\cdot\text{K})(300 \text{ K}) \ln \frac{250 \text{ kPa}}{100 \text{ kPa}} = 78.89 \text{ kJ/kg}$$

That is,

$$q_{\text{out}} = w_{\text{in}} = 78.89 \text{ kJ/kg}$$

The entropy change of air during this isothermal process is

$$\Delta s_{\text{air}} = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} = -R \ln \frac{P_2}{P_1} = -(0.287 \text{ kJ/kg}\cdot\text{K}) \ln \frac{250 \text{ kPa}}{100 \text{ kPa}} = -0.2630 \text{ kJ/kg}\cdot\text{K}$$

The entropy change of the reservoir is

$$\Delta s_{\text{R}} = \frac{q_{\text{R}}}{T_{\text{R}}} = \frac{78.89 \text{ kJ/kg}}{300 \text{ K}} = 0.2630 \text{ kJ/kg}\cdot\text{K}$$

Note that the sign of heat transfer is taken with respect to the reservoir. The total entropy change (i.e., entropy generation) is the sum of the entropy changes of air and the reservoir:

$$\Delta s_{\text{total}} = \Delta s_{\text{air}} + \Delta s_{\text{R}} = -0.2630 + 0.2630 = \mathbf{0 \text{ kJ/kg}\cdot\text{K}}$$

Not only this process is possible but also completely reversible.

8-182 A paddle wheel does work on the water contained in a rigid tank. For a zero entropy change of water, the final pressure in the tank, the amount of heat transfer between the tank and the surroundings, and the entropy generation during the process are to be determined.

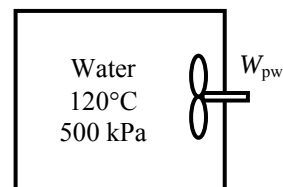
Assumptions The tank is stationary and the kinetic and potential energy changes are negligible.

Analysis (a) Using saturated liquid properties for the compressed liquid at the initial state (Table A-4)

$$\left. \begin{array}{l} T_1 = 120^\circ\text{C} \\ x_1 = 0 \text{ (sat. liq.)} \end{array} \right\} \begin{array}{l} u_1 = 503.60 \text{ kJ/kg} \\ s_1 = 1.5279 \text{ kJ/kg}\cdot\text{K} \end{array}$$

The entropy change of water is zero, and thus at the final state we have

$$\left. \begin{array}{l} T_2 = 95^\circ\text{C} \\ s_2 = s_1 = 1.5279 \text{ kJ/kg}\cdot\text{K} \end{array} \right\} \begin{array}{l} P_2 = \mathbf{84.6 \text{ kPa}} \\ u_2 = 492.63 \text{ kJ/kg} \end{array}$$



(b) The heat transfer can be determined from an energy balance on the tank

$$Q_{\text{out}} = W_{\text{pw, in}} - m(u_2 - u_1) = 22 \text{ kJ} - (1.5 \text{ kg})(492.63 - 503.60) \text{ kJ/kg} = \mathbf{38.5 \text{ kJ}}$$

(c) Since the entropy change of water is zero, the entropy generation is only due to the entropy increase of the surroundings, which is determined from

$$S_{\text{gen}} = \Delta S_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{38.5 \text{ kJ}}{(15 + 273) \text{ K}} = \mathbf{0.134 \text{ kJ/K}}$$

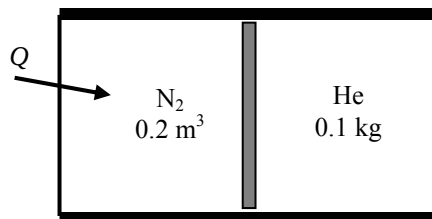
8-183 A horizontal cylinder is separated into two compartments by a piston, one side containing nitrogen and the other side containing helium. Heat is added to the nitrogen side. The final temperature of the helium, the final volume of the nitrogen, the heat transferred to the nitrogen, and the entropy generation during this process are to be determined.

Assumptions 1 Kinetic and potential energy changes are negligible. 2 Nitrogen and helium are ideal gases with constant specific heats at room temperature. 3 The piston is adiabatic and frictionless.

Properties The properties of nitrogen at room temperature are $R = 0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$, $c_p = 1.039 \text{ kJ/kg}\cdot\text{K}$, $c_v = 0.743 \text{ kJ/kg}\cdot\text{K}$, $k = 1.4$. The properties for helium are $R = 2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$, $c_p = 5.1926 \text{ kJ/kg}\cdot\text{K}$, $c_v = 3.1156 \text{ kJ/kg}\cdot\text{K}$, $k = 1.667$ (Table A-2).

Analysis (a) Helium undergoes an isentropic compression process, and thus the final helium temperature is determined from

$$T_{\text{He},2} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (20 + 273) \text{ K} \left(\frac{120 \text{ kPa}}{95 \text{ kPa}} \right)^{(1.667-1)/1.667} = \mathbf{321.7 \text{ K}}$$



(b) The initial and final volumes of the helium are

$$V_{\text{He},1} = \frac{mRT_1}{P_1} = \frac{(0.1 \text{ kg})(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})}{95 \text{ kPa}} = 0.6406 \text{ m}^3$$

$$V_{\text{He},2} = \frac{mRT_2}{P_2} = \frac{(0.1 \text{ kg})(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(321.7 \text{ K})}{120 \text{ kPa}} = 0.5568 \text{ m}^3$$

Then, the final volume of nitrogen becomes

$$V_{\text{N}_2,2} = V_{\text{N}_2,1} + V_{\text{He},1} - V_{\text{He},2} = 0.2 + 0.6406 - 0.5568 = \mathbf{0.2838 \text{ m}^3}$$

(c) The mass and final temperature of nitrogen are

$$m_{\text{N}_2} = \frac{P_1 V_1}{RT_1} = \frac{(95 \text{ kPa})(0.2 \text{ m}^3)}{(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})} = 0.2185 \text{ kg}$$

$$T_{\text{N}_2,2} = \frac{P_2 V_2}{mR} = \frac{(120 \text{ kPa})(0.2838 \text{ m}^3)}{(0.2185 \text{ kg})(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 525.1 \text{ K}$$

The heat transferred to the nitrogen is determined from an energy balance

$$\begin{aligned} Q_{\text{in}} &= \Delta U_{\text{N}_2} + \Delta U_{\text{He}} \\ &= [mc_v(T_2 - T_1)]_{\text{N}_2} + [mc_v(T_2 - T_1)]_{\text{He}} \\ &= (0.2185 \text{ kg})(0.743 \text{ kJ/kg}\cdot\text{K})(525.1 - 293) + (0.1 \text{ kg})(3.1156 \text{ kJ/kg}\cdot\text{K})(321.7 - 293) \\ &= \mathbf{46.6 \text{ kJ}} \end{aligned}$$

(d) Noting that helium undergoes an isentropic process, the entropy generation is determined to be

$$\begin{aligned} S_{\text{gen}} &= \Delta S_{\text{N}_2} + \Delta S_{\text{surr}} = m_{\text{N}_2} \left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) + \frac{-Q_{\text{in}}}{T_R} \\ &= (0.2185 \text{ kg}) \left[(1.039 \text{ kJ/kg}\cdot\text{K}) \ln \frac{525.1 \text{ K}}{293 \text{ K}} - (0.2968 \text{ kJ/kg}\cdot\text{K}) \ln \frac{120 \text{ kPa}}{95 \text{ kPa}} \right] + \frac{-46.6 \text{ kJ}}{(500 + 273) \text{ K}} \\ &= \mathbf{0.057 \text{ kJ/K}} \end{aligned}$$

8-184 An electric resistance heater is doing work on carbon dioxide contained in a rigid tank. The final temperature in the tank, the amount of heat transfer, and the entropy generation are to be determined.

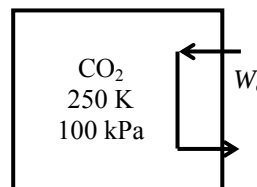
Assumptions 1 Kinetic and potential energy changes are negligible. 2 Carbon dioxide is ideal gas with constant specific heats at room temperature.

Properties The properties of CO₂ at an anticipated average temperature of 350 K are $R = 0.1889$ kPa·m³/kg·K, $c_p = 0.895$ kJ/kg·K, $c_v = 0.706$ kJ/kg·K (Table A-2b).

Analysis (a) The mass and the final temperature of CO₂ may be determined from ideal gas equation

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(0.8 \text{ m}^3)}{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(250 \text{ K})} = 1.694 \text{ kg}$$

$$T_2 = \frac{P_2 V}{mR} = \frac{(175 \text{ kPa})(0.8 \text{ m}^3)}{(1.694 \text{ kg})(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})} = \mathbf{437.5 \text{ K}}$$



(b) The amount of heat transfer may be determined from an energy balance on the system

$$Q_{\text{out}} = \dot{E}_{e,\text{in}} \Delta t - mc_v(T_2 - T_1)$$

$$= (0.5 \text{ kW})(40 \times 60 \text{ s}) - (1.694 \text{ kg})(0.706 \text{ kJ/kg} \cdot \text{K})(437.5 - 250) \text{ K} = \mathbf{975.8 \text{ kJ}}$$

(c) The entropy generation associated with this process may be obtained by calculating total entropy change, which is the sum of the entropy changes of CO₂ and the surroundings

$$S_{\text{gen}} = \Delta S_{\text{CO}_2} + \Delta S_{\text{surr}} = m \left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

$$= (1.694 \text{ kg}) \left[(0.895 \text{ kJ/kg} \cdot \text{K}) \ln \frac{437.5 \text{ K}}{250 \text{ K}} - (0.1889 \text{ kJ/kg} \cdot \text{K}) \ln \frac{175 \text{ kPa}}{100 \text{ kPa}} \right] + \frac{975.8 \text{ kJ}}{300 \text{ K}}$$

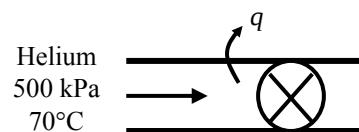
$$= \mathbf{3.92 \text{ kJ/K}}$$

8-185 Heat is lost from the helium as it is throttled in a throttling valve. The exit pressure and temperature of helium and the entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible. 3 Helium is an ideal gas with constant specific heats.

Properties The properties of helium are $R = 2.0769$ kPa.m³/kg.K, $c_p = 5.1926$ kJ/kg.K (Table A-2a).

Analysis (a) The final temperature of helium may be determined from an energy balance on the control volume



$$q_{\text{out}} = c_p (T_1 - T_2) \longrightarrow T_2 = T_1 - \frac{q_{\text{out}}}{c_p} = 70^\circ\text{C} - \frac{2.5 \text{ kJ/kg}}{5.1926 \text{ kJ/kg}\cdot^\circ\text{C}} = 342.5 \text{ K} = \mathbf{69.5^\circ\text{C}}$$

The final pressure may be determined from the relation for the entropy change of helium

$$\Delta s_{\text{He}} = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$$

$$0.25 \text{ kJ/kg}\cdot\text{K} = (5.1926 \text{ kJ/kg}\cdot\text{K}) \ln \frac{342.5 \text{ K}}{343 \text{ K}} - (2.0769 \text{ kJ/kg}\cdot\text{K}) \ln \frac{P_2}{500 \text{ kPa}}$$

$$P_2 = \mathbf{441.7 \text{ kPa}}$$

(b) The entropy generation associated with this process may be obtained by adding the entropy change of helium as it flows in the valve and the entropy change of the surroundings

$$s_{\text{gen}} = \Delta s_{\text{He}} + \Delta s_{\text{surr}} = \Delta s_{\text{He}} + \frac{q_{\text{out}}}{T_{\text{surr}}} = 0.25 \text{ kJ/kg}\cdot\text{K} + \frac{2.5 \text{ kJ/kg}}{(25 + 273) \text{ K}} = \mathbf{0.258 \text{ kJ/kg}\cdot\text{K}}$$

8-186 Refrigerant-134a is compressed in a compressor. The rate of heat loss from the compressor, the exit temperature of R-134a, and the rate of entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible.

Analysis (a) The properties of R-134a at the inlet of the compressor are (Table A-12)

$$\left. \begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} \nu_1 = 0.09987 \text{ m}^3/\text{kg} \\ h_1 = 244.46 \text{ kJ/kg} \\ s_1 = 0.93773 \text{ kJ/kg}\cdot\text{K} \end{array}$$

The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.03 \text{ m}^3/\text{s}}{0.09987 \text{ m}^3/\text{kg}} = 0.3004 \text{ kg/s}$$

Given the entropy increase of the surroundings, the heat lost from the compressor is

$$\Delta \dot{S}_{\text{surr}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}} \longrightarrow \dot{Q}_{\text{out}} = T_{\text{surr}} \Delta \dot{S}_{\text{surr}} = (20 + 273 \text{ K})(0.008 \text{ kW/K}) = \mathbf{2.344 \text{ kW}}$$

(b) An energy balance on the compressor gives

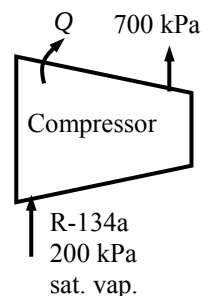
$$\begin{aligned} \dot{W}_{\text{in}} - \dot{Q}_{\text{out}} &= \dot{m}(h_2 - h_1) \\ 10 \text{ kW} - 2.344 \text{ kW} &= (0.3004 \text{ kg/s})(h_2 - 244.46) \text{ kJ/kg} \longrightarrow h_2 = 269.94 \text{ kJ/kg} \end{aligned}$$

The exit state is now fixed. Then,

$$\left. \begin{array}{l} P_2 = 700 \text{ kPa} \\ h_2 = 269.94 \text{ kJ/kg} \end{array} \right\} \begin{array}{l} T_2 = \mathbf{31.5^\circ\text{C}} \\ s_2 = 0.93620 \text{ kJ/kg}\cdot\text{K} \end{array}$$

(c) The entropy generation associated with this process may be obtained by adding the entropy change of R-134a as it flows in the compressor and the entropy change of the surroundings

$$\begin{aligned} \dot{S}_{\text{gen}} &= \Delta \dot{S}_{\text{R}} + \Delta \dot{S}_{\text{surr}} = \dot{m}(s_2 - s_1) + \Delta \dot{S}_{\text{surr}} \\ &= (0.3004 \text{ kg/s})(0.93620 - 0.93773) \text{ kJ/kg}\cdot\text{K} + 0.008 \text{ kW/K} \\ &= \mathbf{0.00754 \text{ kJ/K}} \end{aligned}$$

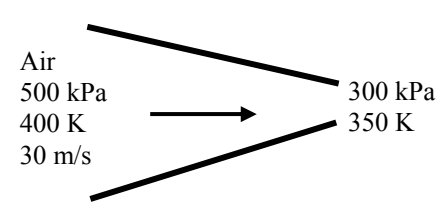


8-187 Air flows in an adiabatic nozzle. The isentropic efficiency, the exit velocity, and the entropy generation are to be determined.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg}\cdot\text{K}$ (Table A-1).

Assumptions 1 Steady operating conditions exist. 2 Potential energy changes are negligible.

Analysis (a) (b) Using variable specific heats, the properties can be determined from air table as follows

$$\begin{aligned}
 T_1 = 400 \text{ K} &\longrightarrow h_1 = 400.98 \text{ kJ/kg} \\
 &\quad s_1^0 = 1.99194 \text{ kJ/kg}\cdot\text{K} \\
 &\quad P_{r1} = 3.806 \\
 T_2 = 350 \text{ K} &\longrightarrow h_2 = 350.49 \text{ kJ/kg} \\
 &\quad s_2^0 = 1.85708 \text{ kJ/kg}\cdot\text{K} \\
 P_{r2} = \frac{P_2}{P_1} P_{r1} &= \frac{300 \text{ kPa}}{500 \text{ kPa}} (3.806) = 2.2836 \longrightarrow h_{2s} = 346.31 \text{ kJ/kg}
 \end{aligned}$$


Energy balances on the control volume for the actual and isentropic processes give

$$\begin{aligned}
 h_1 + \frac{V_1^2}{2} &= h_2 + \frac{V_2^2}{2} \\
 400.98 \text{ kJ/kg} + \frac{(30 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) &= 350.49 \text{ kJ/kg} + \frac{V_2^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \\
 V_2 &= \mathbf{319.1 \text{ m/s}}
 \end{aligned}$$

$$\begin{aligned}
 h_1 + \frac{V_1^2}{2} &= h_{2s} + \frac{V_{2s}^2}{2} \\
 400.98 \text{ kJ/kg} + \frac{(30 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) &= 346.31 \text{ kJ/kg} + \frac{V_{2s}^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \\
 V_{2s} &= 331.8 \text{ m/s}
 \end{aligned}$$

The isentropic efficiency is determined from its definition,

$$\eta_N = \frac{V_2^2}{V_{2s}^2} = \frac{(319.1 \text{ m/s})^2}{(331.8 \text{ m/s})^2} = \mathbf{0.925}$$

(c) Since the nozzle is adiabatic, the entropy generation is equal to the entropy increase of the air as it flows in the nozzle

$$\begin{aligned}
 s_{\text{gen}} &= \Delta s_{\text{air}} = s_2^0 - s_1^0 - R \ln \frac{P_2}{P_1} \\
 &= (1.85708 - 1.99194) \text{ kJ/kg}\cdot\text{K} - (0.287 \text{ kJ/kg}\cdot\text{K}) \ln \frac{300 \text{ kPa}}{500 \text{ kPa}} = \mathbf{0.0118 \text{ kJ/kg}\cdot\text{K}}
 \end{aligned}$$

8-188 It is to be shown that the difference between the steady-flow and boundary works is the flow energy.

Analysis The total differential of flow energy Pv can be expressed as

$$d(Pv) = Pd v + v dP = \delta w_b - \delta w_{flow} = \delta(w_b - w_{flow})$$

Therefore, the difference between the reversible steady-flow work and the reversible boundary work is the flow energy.

8-189 An insulated rigid tank is connected to a piston-cylinder device with zero clearance that is maintained at constant pressure. A valve is opened, and some steam in the tank is allowed to flow into the cylinder. The final temperatures in the tank and the cylinder are to be determined.

Assumptions 1 Both the tank and cylinder are well-insulated and thus heat transfer is negligible. 2 The water that remains in the tank underwent a reversible adiabatic process. 3 The thermal energy stored in the tank and cylinder themselves is negligible. 4 The system is stationary and thus kinetic and potential energy changes are negligible.

Analysis (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\begin{aligned}
 P_1 = 500 \text{ kPa} \quad \left. \begin{array}{l} \nu_1 = \nu_{g@500 \text{ kPa}} = 0.37483 \text{ m}^3/\text{kg} \\ u_1 = u_{g@500 \text{ kPa}} = 2560.7 \text{ kJ/kg} \\ s_1 = s_{g@500 \text{ kPa}} = 6.8207 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \text{sat.vapor} \\
 \\
 P_2 = 150 \text{ kPa} \quad \left. \begin{array}{l} s_2 = s_1 \\ (\text{sat.mixture}) \end{array} \right\} \quad \begin{aligned} T_{2,A} &= T_{\text{sat}@150 \text{ kPa}} = \mathbf{111.35^\circ\text{C}} \\ x_{2,A} &= \frac{s_{2,A} - s_f}{s_{fg}} = \frac{6.8207 - 1.4337}{5.7894} = 0.9305 \\ \nu_{2,A} &= \nu_f + x_{2,A} \nu_{fg} = 0.001053 + (0.9305)(1.1594 - 0.001053) = 1.0789 \text{ m}^3/\text{kg} \\ u_{2,A} &= u_f + x_{2,A} u_{fg} = 466.97 + (0.9305)(2052.3 \text{ kJ/kg}) = 2376.6 \text{ kJ/kg} \end{aligned}
 \end{aligned}$$

The initial and the final masses in tank A are

$$m_{1,A} = \frac{\nu_A}{\nu_{1,A}} = \frac{0.4 \text{ m}^3}{0.37483 \text{ m}^3/\text{kg}} = 1.067 \text{ kg} \quad \text{and} \quad m_{2,A} = \frac{\nu_A}{\nu_{2,A}} = \frac{0.4 \text{ m}^3}{1.0789 \text{ m}^3/\text{kg}} = 0.371 \text{ kg}$$

Thus,

$$m_{2,B} = m_{1,A} - m_{2,A} = 1.067 - 0.371 = 0.696 \text{ kg}$$

(b) The boundary work done during this process is

$$W_{b,\text{out}} = \int_1^2 P d\nu = P_B (\nu_{2,B} - 0) = P_B m_{2,B} \nu_{2,B}$$

Taking the contents of both the tank and the cylinder to be the system, the energy balance for this closed system can be expressed as

$$\begin{aligned}
 \underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} &= \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \\
 -W_{b,\text{out}} = \Delta U &= (\Delta U)_A + (\Delta U)_B
 \end{aligned}$$

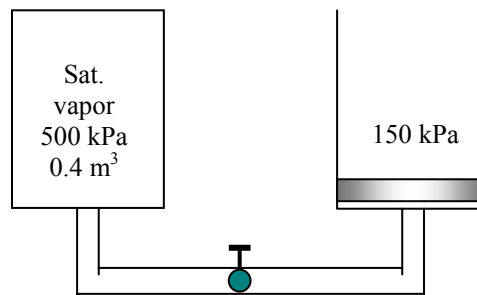
$$\begin{aligned}
 W_{b,\text{out}} + (\Delta U)_A + (\Delta U)_B &= 0 \\
 \text{or,} \quad P_B m_{2,B} \nu_{2,B} + (m_2 u_2 - m_1 u_1)_A + (m_2 u_2)_B &= 0 \\
 m_{2,B} h_{2,B} + (m_2 u_2 - m_1 u_1)_A &= 0
 \end{aligned}$$

Thus,

$$h_{2,B} = \frac{(m_1 u_1 - m_2 u_2)_A}{m_{2,B}} = \frac{(1.067)(2560.7) - (0.371)(2376.6)}{0.696} = 2658.8 \text{ kJ/kg}$$

At 150 kPa, $h_f = 467.13$ and $h_g = 2693.1$ kJ/kg. Thus at the final state, the cylinder will contain a saturated liquid-vapor mixture since $h_f < h_2 < h_g$. Therefore,

$$T_{2,B} = T_{\text{sat}@150 \text{ kPa}} = \mathbf{111.35^\circ\text{C}}$$



8-190 Carbon dioxide is compressed in a reversible isothermal process using a steady-flow device. The work required and the heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Changes in the kinetic and potential energies are negligible. 3 CO₂ is an ideal gas with constant specific heats.

Properties The gas constant of CO₂ is $R = 0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$ (Table A-2a).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

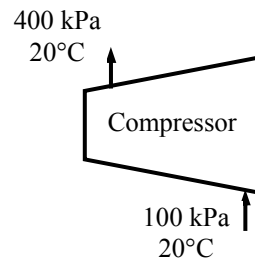
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} \quad (\text{since } T_2 = T_1)$$



The work input for this isothermal, reversible process is

$$w_{\text{in}} = RT \ln \frac{P_2}{P_1} = (0.1889 \text{ kJ/kg} \cdot \text{K})(293 \text{ K}) \ln \frac{400 \text{ kPa}}{100 \text{ kPa}} = \mathbf{76.7 \text{ kJ/kg}}$$

From the energy balance equation,

$$q_{\text{out}} = w_{\text{in}} = \mathbf{76.7 \text{ kJ/kg}}$$

8-191 Carbon dioxide is compressed in an isentropic process using a steady-flow device. The work required and the heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Changes in the kinetic and potential energies are negligible. 3 CO₂ is an ideal gas with constant specific heats.

Properties The gas constant of CO₂ is $R = 0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$. Other properties at room temperature are $c_p = 0.846 \text{ kJ/kg}\cdot\text{K}$ and $k = 1.289$ (Table A-2a).

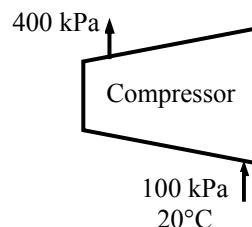
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



The temperature at the compressor exit for the isentropic process of an ideal gas is

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (293 \text{ K}) \left(\frac{400 \text{ kPa}}{100 \text{ kPa}} \right)^{0.289/1.289} = 399.8 \text{ K}$$

Substituting,

$$w_{\text{in}} = c_p(T_2 - T_1) = (0.846 \text{ kJ/kg}\cdot\text{K})(399.8 - 293)\text{K} = \mathbf{90.4 \text{ kJ}}$$

The work input increases from 76.7 kJ/kg to 90.4 kJ/kg when the process is executed isentropically instead of isothermally. Since the process is isentropic (i.e., reversible, adiabatic), the heat transfer is zero.

8-192 R-134a is compressed in an isentropic compressor. The work required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic). **3** Kinetic and potential energy changes are negligible.

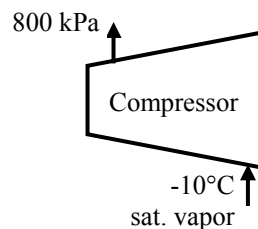
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

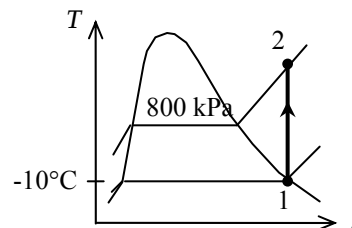


The inlet state properties are

$$\left. \begin{array}{l} T_1 = -10^\circ\text{C} \\ x_1 = 1 \end{array} \right\} \begin{array}{l} h_1 = 244.51 \text{ kJ/kg} \\ s_1 = 0.93766 \text{ kJ/kg} \cdot \text{K} \end{array} \quad (\text{Table A-11})$$

For this isentropic process, the final state enthalpy is

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ s_2 = s_1 = 0.93766 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} h_2 = 273.22 \text{ kJ/kg} \quad (\text{Table A-13})$$



Substituting,

$$w_{\text{in}} = h_2 - h_1 = (273.22 - 244.51) \text{ kJ/kg} = \mathbf{28.7 \text{ kJ/kg}}$$

8-193 Refrigerant-134a is expanded adiabatically in a capillary tube. The rate of entropy generation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible.

Analysis The rate of entropy generation within the expansion device during this process can be determined by applying the rate form of the entropy balance on the system. Noting that the system is adiabatic and thus there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\begin{array}{c}
 \underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{\text{gen}}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\text{Rate of change of entropy}} \overset{\text{0 (steady)}}{=} 0 \\
 \dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \\
 \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \\
 s_{\text{gen}} = s_2 - s_1
 \end{array}
 \quad
 \begin{array}{c}
 \text{R-134a} \\
 50^\circ\text{C} \\
 \text{sat. liq.}
 \end{array}
 \xrightarrow{\text{Capillary tube}}
 \begin{array}{c}
 -12^\circ\text{C}
 \end{array}$$

It may be easily shown with an energy balance that the enthalpy remains constant during the throttling process. The properties of the refrigerant at the inlet and exit states are (Tables A-11 through A-13)

$$\left. \begin{array}{l} T_1 = 50^\circ\text{C} \\ x_1 = 0 \end{array} \right\} \begin{array}{l} h_1 = 123.50 \text{ kJ/kg} \cdot \text{K} \\ s_1 = 0.44193 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} T_2 = -12^\circ\text{C} \\ h_2 = h_1 = 123.50 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \begin{array}{l} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{123.50 - 35.92}{207.38} = 0.4223 \\ s_2 = s_f + x_2 s_{fg} = 0.14504 + (0.4223)(0.79406) = 0.48038 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Substituting,

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) = (0.2 \text{ kg/s})(0.48038 - 0.44193) \text{ kJ/kg} \cdot \text{K} = \mathbf{0.00769 \text{ kW/K}}$$

8-194 Steam is expanded in an adiabatic turbine. Six percent of the inlet steam is bled for feedwater heating. The isentropic efficiencies for two stages of the turbine are given. The power produced by the turbine and the overall efficiency of the turbine are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 The turbine is well-insulated, and there is no heat transfer from the turbine.

Analysis There is one inlet and two exits. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

$$w_{\text{out}} = h_1 - 0.06h_2 - 0.94h_3$$

$$w_{\text{out}} = (h_1 - h_2) + 0.94(h_2 - h_3)$$

The isentropic and actual enthalpies at three states are determined using steam tables as follows:

$$\left. \begin{array}{l} P_1 = 4 \text{ MPa} \\ T_1 = 350^\circ\text{C} \end{array} \right\} \begin{array}{l} h_1 = 3093.3 \text{ kJ/kg} \\ s_1 = 6.5843 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ s_2 = s_1 = 6.5843 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \begin{array}{l} x_{2s} = 0.9832 \\ h_{2s} = 2734.0 \text{ kJ/kg} \end{array}$$

$$\eta_{T,1} = \frac{h_1 - h_2}{h_1 - h_{2s}} \longrightarrow h_2 = h_1 - \eta_{T,1}(h_1 - h_{2s}) = 3093.3 - (0.97)(3093.3 - 2734.0) = 2744.8 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 800 \text{ kPa} \\ h_2 = 2744.8 \text{ kJ/kg} \end{array} \right\} \begin{array}{l} x_2 = 0.9885 \\ s_2 = 6.6086 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_3 = 30 \text{ kPa} \\ s_3 = s_2 = 6.6086 \text{ kJ/kg} \cdot \text{K} \end{array} \right\} \begin{array}{l} x_{3s} = 0.8302 \\ h_{3s} = 2227.9 \text{ kJ/kg} \end{array}$$

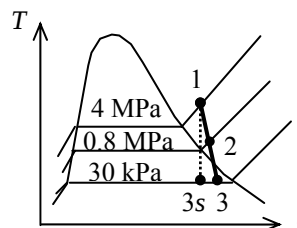
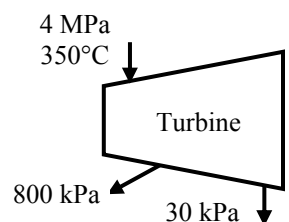
$$\eta_{T,2} = \frac{h_2 - h_3}{h_2 - h_{3s}} \longrightarrow h_3 = h_2 - \eta_{T,2}(h_2 - h_{3s}) = 2744.8 - (0.95)(2744.8 - 2227.9) = 2253.7 \text{ kJ/kg}$$

Substituting,

$$\begin{aligned} w_{\text{out}} &= (h_1 - h_2) + 0.94(h_2 - h_3) \\ &= (3093.3 - 2744.8) + 0.94(2744.8 - 2253.7) \\ &= \mathbf{810.1 \text{ kJ/kg - inlet}} \end{aligned}$$

The overall isentropic efficiency of the turbine is

$$\eta_T = \frac{(h_1 - h_2) + 0.94(h_2 - h_3)}{(h_1 - h_{2s}) + 0.94(h_2 - h_{3s})} = \frac{(3093.3 - 2744.8) + 0.94(2744.8 - 2253.7)}{(3093.3 - 2734.0) + 0.94(2744.8 - 2227.9)} = \frac{810.1}{845.2} = \mathbf{0.958 = 95.8\%}$$



8-195 Air is compressed steadily by a compressor from a specified state to a specified pressure. The minimum power input required is to be determined for the cases of adiabatic and isothermal operation.

Assumptions 1 This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats. **4** The process is reversible since the work input to the compressor will be minimum when the compression process is reversible.

Properties The gas constant of air is $R = 0.287 \text{ kJ/kg} \cdot \text{K}$ (Table A-1).

Analysis (a) For the adiabatic case, the process will be reversible and adiabatic (i.e., isentropic), thus the isentropic relations are applicable.

$$T_1 = 290 \text{ K} \longrightarrow P_{r_1} = 1.2311 \text{ and } h_1 = 290.16 \text{ kJ/kg}$$

and

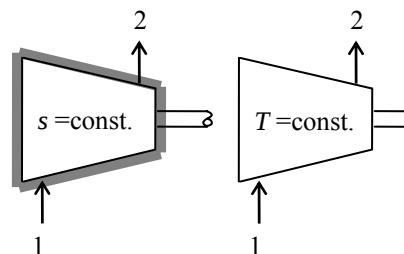
$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{700 \text{ kPa}}{100 \text{ kPa}} (1.2311) = 8.6177 \rightarrow \begin{matrix} T_2 = 503.3 \text{ K} \\ h_2 = 506.45 \text{ kJ/kg} \end{matrix}$$

The energy balance for the compressor, which is a steady-flow system, can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \overset{\phi_0 \text{ (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \rightarrow \dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$



Substituting, the power input to the compressor is determined to be

$$\dot{W}_{\text{in}} = (5/60 \text{ kg/s})(506.45 - 290.16) \text{ kJ/kg} = \mathbf{18.0 \text{ kW}}$$

(b) In the case of the reversible isothermal process, the steady-flow energy balance becomes

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \rightarrow \dot{W}_{\text{in}} + \dot{m}h_1 - \dot{Q}_{\text{out}} = \dot{m}h_2 \rightarrow \dot{W}_{\text{in}} = \dot{Q}_{\text{out}} + \dot{m}(h_2 - h_1) \overset{\phi_0}{=} \dot{Q}_{\text{out}}$$

since $h = h(T)$ for ideal gases, and thus the enthalpy change in this case is zero. Also, for a reversible isothermal process,

$$\dot{Q}_{\text{out}} = \dot{m}T(s_1 - s_2) = -\dot{m}T(s_2 - s_1)$$

where

$$s_2 - s_1 = (s_2^\circ - s_1^\circ) \overset{\phi_0}{=} -R \ln \frac{P_2}{P_1} = -R \ln \frac{700}{100} = -(0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{700 \text{ kPa}}{100 \text{ kPa}} = -0.5585 \text{ kJ/kg} \cdot \text{K}$$

Substituting, the power input for the reversible isothermal case becomes

$$\dot{W}_{\text{in}} = -(5/60 \text{ kg/s})(290 \text{ K})(-0.5585 \text{ kJ/kg} \cdot \text{K}) = \mathbf{13.5 \text{ kW}}$$

8-196 Refrigerant-134a is compressed by a 0.7-kW adiabatic compressor from a specified state to another specified state. The isentropic efficiency, the volume flow rate at the inlet, and the maximum flow rate at the compressor inlet are to be determined.

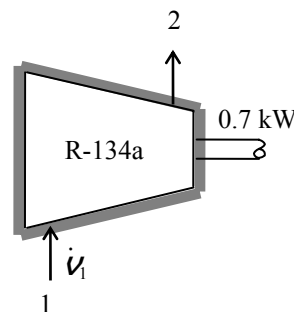
Assumptions 1 This is a steady-flow process since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 The device is adiabatic and thus heat transfer is negligible.

Properties From the R-134a tables (Tables A-11 through A-13)

$$\left. \begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10^\circ\text{C} \end{array} \right\} \begin{array}{l} v_1 = 0.14605 \text{ m}^3/\text{kg} \\ h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.9724 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 700 \text{ kPa} \\ T_2 = 50^\circ\text{C} \end{array} \right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_2 = 700 \text{ kPa} \\ s_{2s} = s_1 \end{array} \right\} h_{2s} = 281.16 \text{ kJ/kg}$$



Analysis (a) The isentropic efficiency is determined from its definition,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{281.16 - 246.36}{288.53 - 246.36} = 0.825 = \mathbf{82.5\%}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\phi 0} (\text{steady})}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{a,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \equiv \Delta \text{ke} \equiv \Delta \text{pe} \equiv 0)$$

$$\dot{W}_{a,\text{in}} = \dot{m}(h_2 - h_1)$$

Then the mass and volume flow rates of the refrigerant are determined to be

$$\dot{m} = \frac{\dot{W}_{a,\text{in}}}{h_{2a} - h_1} = \frac{0.7 \text{ kJ/s}}{(288.53 - 246.36) \text{ kJ/kg}} = 0.0166 \text{ kg/s}$$

$$\dot{V}_1 = \dot{m}v_1 = (0.0166 \text{ kg/s})(0.14605 \text{ m}^3/\text{kg}) = 0.00242 \text{ m}^3/\text{s} = \mathbf{145 \text{ L/min}}$$

(c) The volume flow rate will be a maximum when the process is isentropic, and it is determined similarly from the steady-flow energy equation applied to the isentropic process. It gives

$$\dot{m}_{\text{max}} = \frac{\dot{W}_{s,\text{in}}}{h_{2s} - h_1} = \frac{0.7 \text{ kJ/s}}{(281.16 - 246.36) \text{ kJ/kg}} = 0.0201 \text{ kg/s}$$

$$\dot{V}_{1,\text{max}} = \dot{m}_{\text{max}}v_1 = (0.0201 \text{ kg/s})(0.14605 \text{ m}^3/\text{kg}) = 0.00294 \text{ m}^3/\text{s} = \mathbf{176 \text{ L/min}}$$

Discussion Note that the raising the isentropic efficiency of the compressor to 100% would increase the volumetric flow rate by more than 20%.

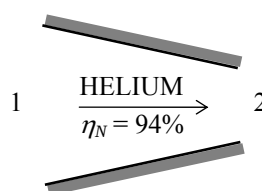
8-197E Helium is accelerated by a 94% efficient nozzle from a low velocity to 1000 ft/s. The pressure and temperature at the nozzle inlet are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time. 2 Helium is an ideal gas with constant specific heats. 3 Potential energy changes are negligible. 4 The device is adiabatic and thus heat transfer is negligible.

Properties The specific heat ratio of helium is $k = 1.667$. The constant pressure specific heat of helium is 1.25 Btu/lbm·R (Table A-2E).

Analysis We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\phi_0} (\text{steady})}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$


$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta p e \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,\text{avg}}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Solving for T_1 and substituting,

$$T_1 = T_{2a} + \frac{V_{2s}^2 - V_1^2}{2C_p} \stackrel{\phi_0}{=} 180^\circ\text{F} + \frac{(1000 \text{ ft/s})^2}{2(1.25 \text{ Btu/lbm} \cdot \text{R})} \left(\frac{1 \text{ Btu/lbm}}{25,037 \text{ ft}^2/\text{s}^2} \right) = \mathbf{196.0^\circ\text{F} = 656 \text{ R}}$$

From the isentropic efficiency relation,

$$\eta_N = \frac{h_{2a} - h_1}{h_{2s} - h_1} = \frac{c_p(T_{2a} - T_1)}{c_p(T_{2s} - T_1)}$$

or,

$$T_{2s} = T_1 + (T_{2a} - T_1)/\eta_N = 656 + (640 - 656)/(0.94) = 639 \text{ R}$$

From the isentropic relation, $\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1} \right)^{(k-1)/k}$

$$P_1 = P_2 \left(\frac{T_1}{T_{2s}} \right)^{k/(k-1)} = (14 \text{ psia}) \left(\frac{656 \text{ R}}{639 \text{ R}} \right)^{1.667/0.667} = \mathbf{14.9 \text{ psia}}$$

8-198 ... 8-201 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 9

MECHANISMS OF HEAT TRANSFER

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Heat Transfer Mechanisms

9-1C The house with the lower rate of heat transfer through the walls will be more energy efficient. Heat conduction is proportional to thermal conductivity (which is 0.72 W/m.°C for brick and 0.17 W/m.°C for wood, Table 9-1) and inversely proportional to thickness. The wood house is more energy efficient since the wood wall is twice as thick but it has about one-fourth the conductivity of brick wall.

9-2C The thermal conductivity of a material is the rate of heat transfer through a unit thickness of the material per unit area and per unit temperature difference. The thermal conductivity of a material is a measure of how fast heat will be conducted in that material.

9-3C The mechanisms of heat transfer are conduction, convection and radiation. Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion, and it involves combined effects of conduction and fluid motion. Radiation is energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules.

9-4C In solids, conduction is due to the combination of the vibrations of the molecules in a lattice and the energy transport by free electrons. In gases and liquids, it is due to the collisions of the molecules during their random motion.

9-5C The parameters that effect the rate of heat conduction through a windowless wall are the geometry and surface area of wall, its thickness, the material of the wall, and the temperature difference across the wall.

9-6C Conduction is expressed by Fourier's law of conduction as $\dot{Q}_{\text{cond}} = -kA \frac{dT}{dx}$ where dT/dx is the temperature gradient, k is the thermal conductivity, and A is the area which is normal to the direction of heat transfer.

Convection is expressed by Newton's law of cooling as $\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty)$ where h is the convection heat transfer coefficient, A_s is the surface area through which convection heat transfer takes place, T_s is the surface temperature and T_∞ is the temperature of the fluid sufficiently far from the surface.

Radiation is expressed by Stefan-Boltzman law as $\dot{Q}_{\text{rad}} = \epsilon \sigma A_s(T_s^4 - T_{\text{surr}}^4)$ where ϵ is the emissivity of surface, A_s is the surface area, T_s is the surface temperature, T_{surr} is the average surrounding surface temperature and $\sigma = 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4$ is the Stefan-Boltzman constant.

9-7C Convection involves fluid motion, conduction does not. In a solid we can have only conduction.

9-8C No. It is purely by radiation.

9-9C In forced convection the fluid is forced to move by external means such as a fan, pump, or the wind. The fluid motion in natural convection is due to buoyancy effects only.

9-10C Emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody at the same temperature. Absorptivity is the fraction of radiation incident on a surface that is absorbed by the surface. The Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature and wavelength.

9-11C A blackbody is an idealized body which emits the maximum amount of radiation at a given temperature and which absorbs all the radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

9-12C No. Such a definition will imply that doubling the thickness will double the heat transfer rate. The equivalent but "more correct" unit of thermal conductivity is $\text{W}\cdot\text{m}/\text{m}^2\cdot^\circ\text{C}$ that indicates product of heat transfer rate and thickness per unit surface area per unit temperature difference.

9-13C In a typical house, heat loss through the wall with glass window will be larger since the glass is much thinner than a wall, and its thermal conductivity is higher than the average conductivity of a wall.

9-14C Diamond is a better heat conductor.

9-15C The rate of heat transfer through both walls can be expressed as

$$\dot{Q}_{\text{wood}} = k_{\text{wood}} A \frac{T_1 - T_2}{L_{\text{wood}}} = (0.16 \text{ W/m}\cdot^\circ\text{C}) A \frac{T_1 - T_2}{0.1 \text{ m}} = 1.6 A (T_1 - T_2)$$

$$\dot{Q}_{\text{brick}} = k_{\text{brick}} A \frac{T_1 - T_2}{L_{\text{brick}}} = (0.72 \text{ W/m}\cdot^\circ\text{C}) A \frac{T_1 - T_2}{0.25 \text{ m}} = 2.88 A (T_1 - T_2)$$

Therefore, heat transfer through the brick wall will be larger despite its higher thickness.

9-16C The thermal conductivity of gases is proportional to the square root of absolute temperature. The thermal conductivity of most liquids, however, decreases with increasing temperature, with water being a notable exception.

9-17C Superinsulations are obtained by using layers of highly reflective sheets separated by glass fibers in an evacuated space. Radiation heat transfer between two surfaces is inversely proportional to the number of sheets used and thus heat loss by radiation will be very low by using this highly reflective sheets. At the same time, evacuating the space between the layers forms a vacuum under 0.000001 atm pressure which minimize conduction or convection through the air space between the layers.

9-18C Most ordinary insulations are obtained by mixing fibers, powders, or flakes of insulating materials with air. Heat transfer through such insulations is by conduction through the solid material, and conduction or convection through the air space as well as radiation. Such systems are characterized by apparent thermal conductivity instead of the ordinary thermal conductivity in order to incorporate these convection and radiation effects.

9-19C The thermal conductivity of an alloy of two metals will most likely be less than the thermal conductivities of both metals.

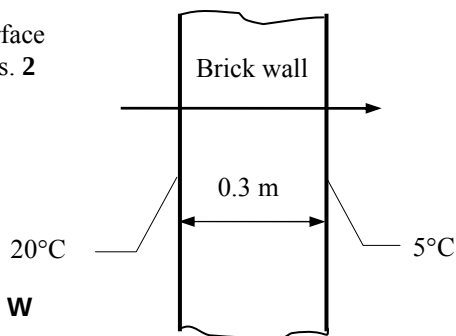
9-20 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. 2 Thermal properties of the wall are constant.

Properties The thermal conductivity of the wall is given to be $k = 0.69 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis Under steady conditions, the rate of heat transfer through the wall is

$$\dot{Q}_{\text{cond}} = kA \frac{\Delta T}{L} = (0.69 \text{ W/m} \cdot ^\circ\text{C})(4 \times 7 \text{ m}^2) \frac{(20 - 5)^\circ\text{C}}{0.3 \text{ m}} = \mathbf{966 \text{ W}}$$



9-21 The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat transfer through the glass in 5 h is to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. 2 Thermal properties of the glass are constant.

Properties The thermal conductivity of the glass is given to be $k = 0.78 \text{ W/m} \cdot ^\circ\text{C}$.

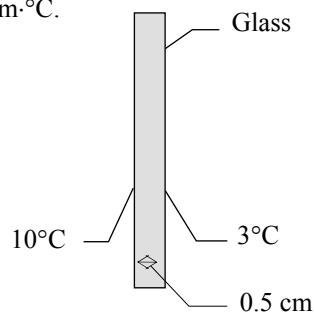
Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is

$$\dot{Q}_{\text{cond}} = kA \frac{\Delta T}{L} = (0.78 \text{ W/m} \cdot ^\circ\text{C})(2 \times 2 \text{ m}^2) \frac{(10 - 3)^\circ\text{C}}{0.005 \text{ m}} = 4368 \text{ W}$$

Then the amount of heat transfer over a period of 5 h becomes

$$Q = \dot{Q}_{\text{cond}} \Delta t = (4.368 \text{ kJ/s})(5 \times 3600 \text{ s}) = \mathbf{78,620 \text{ kJ}}$$

If the thickness of the glass doubled to 1 cm, then the amount of heat transfer will go down by half to **39,310 kJ**.



9-22 EES Prob. 9-21 is reconsidered. The amount of heat loss through the glass as a function of the window glass thickness is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

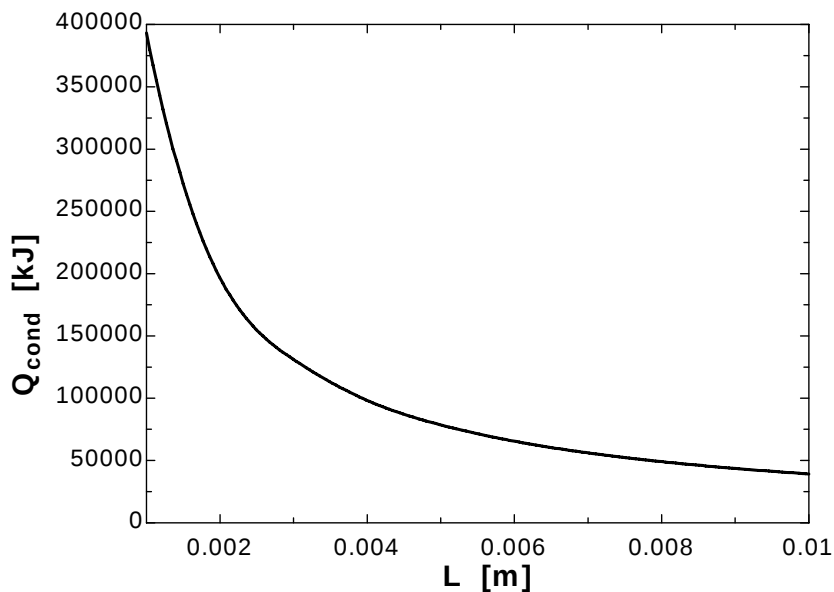
"GIVEN"

L=0.005 [m]
 A=2*2 [m^2]
 T_1=10 [C]
 T_2=3 [C]
 k=0.78 [W/m-C]
 time=5*3600 [s]

"ANALYSIS"

$Q_{\text{dot_cond}} = k \cdot A \cdot (T_1 - T_2) / L$
 $Q_{\text{cond}} = Q_{\text{dot_cond}} \cdot \text{time} \cdot \text{Convert}(\text{J}, \text{kJ})$

L [m]	Q _{cond} [kJ]
0.001	393120
0.002	196560
0.003	131040
0.004	98280
0.005	78624
0.006	65520
0.007	56160
0.008	49140
0.009	43680
0.01	39312



9-23 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values. 2 Thermal properties of the aluminum pan are constant.

Properties The thermal conductivity of the aluminum is given to be $k = 237 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The heat transfer area is

$$A = \pi r^2 = \pi (0.075 \text{ m})^2 = 0.0177 \text{ m}^2$$

Under steady conditions, the rate of heat transfer through the bottom of the pan by conduction is

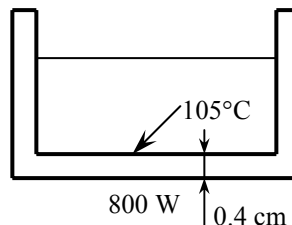
$$\dot{Q} = kA \frac{\Delta T}{L} = kA \frac{T_2 - T_1}{L}$$

Substituting,

$$800 \text{ W} = (237 \text{ W/m}\cdot^\circ\text{C})(0.0177 \text{ m}^2) \frac{T_2 - 105^\circ\text{C}}{0.004 \text{ m}}$$

which gives

$$T_2 = 105.76^\circ\text{C}$$



9-24E The inner and outer surface temperatures of the wall of an electrically heated home during a winter night are measured. The rate of heat loss through the wall that night and its cost are to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values during the entire night. 2 Thermal properties of the wall are constant.

Properties The thermal conductivity of the brick wall is given to be $k = 0.42 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$.

Analysis (a) Noting that the heat transfer through the wall is by conduction and the surface area of the wall is $A = 20 \text{ ft} \times 10 \text{ ft} = 200 \text{ ft}^2$, the steady rate of heat transfer through the wall can be determined from

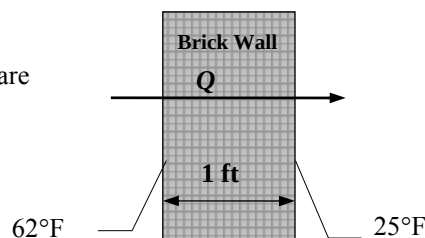
$$\dot{Q} = kA \frac{T_1 - T_2}{L} = (0.42 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(200 \text{ ft}^2) \frac{(62 - 25)^\circ\text{F}}{1 \text{ ft}} = 3108 \text{ Btu/h}$$

or 0.911 kW since $1 \text{ kW} = 3412 \text{ Btu/h}$.

(b) The amount of heat lost during an 8 hour period and its cost are

$$Q = \dot{Q}\Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$$

$$\begin{aligned} \text{Cost} &= (\text{Amount of energy})(\text{Unit cost of energy}) \\ &= (7.288 \text{ kWh})(\$0.07/\text{kWh}) \\ &= \$0.51 \end{aligned}$$



Therefore, the cost of the heat loss through the wall to the home owner that night is \$0.51.

9-25 The thermal conductivity of a material is to be determined by ensuring one-dimensional heat conduction, and by measuring temperatures when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time.

2 Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples. **3** The apparatus possesses thermal symmetry.

Analysis The electrical power consumed by the heater and converted to heat is

$$\dot{W}_e = VI = (110 \text{ V})(0.6 \text{ A}) = 66 \text{ W}$$

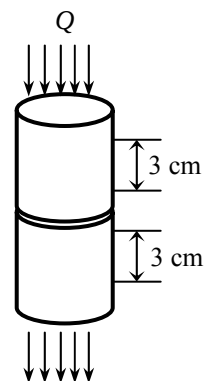
The rate of heat flow through each sample is

$$\dot{Q} = \frac{\dot{W}_e}{2} = \frac{66 \text{ W}}{2} = 33 \text{ W}$$

Then the thermal conductivity of the sample becomes

$$A = \frac{\pi D^2}{4} = \frac{\pi (0.04 \text{ m})^2}{4} = 0.001257 \text{ m}^2$$

$$\dot{Q} = kA \frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(33 \text{ W})(0.03 \text{ m})}{(0.001257 \text{ m}^2)(10^\circ\text{C})} = \mathbf{78.8 \text{ W/m}\cdot^\circ\text{C}}$$



9-26 The thermal conductivity of a material is to be determined by ensuring one-dimensional heat conduction, and by measuring temperatures when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time.

2 Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples. **3** The apparatus possesses thermal symmetry.

Analysis For each sample we have

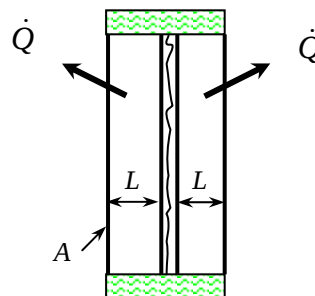
$$\dot{Q} = 25 / 2 = 12.5 \text{ W}$$

$$A = (0.1 \text{ m})(0.1 \text{ m}) = 0.01 \text{ m}^2$$

$$\Delta T = 82 - 74 = 8^\circ\text{C}$$

Then the thermal conductivity of the material becomes

$$\dot{Q} = kA \frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(12.5 \text{ W})(0.005 \text{ m})}{(0.01 \text{ m}^2)(8^\circ\text{C})} = \mathbf{0.781 \text{ W/m}\cdot^\circ\text{C}}$$



9-27 The thermal conductivity of a material is to be determined by ensuring one-dimensional heat conduction, and by measuring temperatures when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples. **3** The apparatus possesses thermal symmetry.

Analysis For each sample we have

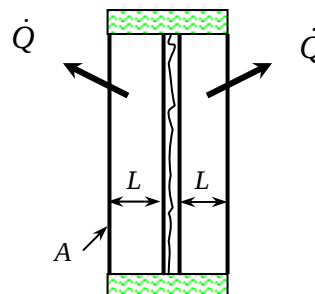
$$\dot{Q} = 20 / 2 = 10 \text{ W}$$

$$A = (0.1 \text{ m})(0.1 \text{ m}) = 0.01 \text{ m}^2$$

$$\Delta T = 82 - 74 = 8^\circ\text{C}$$

Then the thermal conductivity of the material becomes

$$\dot{Q} = kA \frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(10 \text{ W})(0.005 \text{ m})}{(0.01 \text{ m}^2)(8^\circ\text{C})} = \mathbf{0.625 \text{ W/m} \cdot ^\circ\text{C}}$$

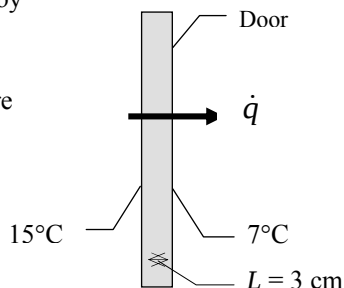


9-28 The thermal conductivity of a refrigerator door is to be determined by measuring the surface temperatures and heat flux when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist when measurements are taken. **2** Heat transfer through the door is one dimensional since the thickness of the door is small relative to other dimensions.

Analysis The thermal conductivity of the door material is determined directly from Fourier's relation to be

$$\dot{q} = k \frac{\Delta T}{L} \longrightarrow k = \frac{\dot{q}L}{\Delta T} = \frac{(25 \text{ W/m}^2)(0.03 \text{ m})}{(15 - 7)^\circ\text{C}} = \mathbf{0.09375 \text{ W/m} \cdot ^\circ\text{C}}$$



9-29 The rate of radiation heat transfer between a person and the surrounding surfaces at specified temperatures is to be determined in summer and in winter.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by convection is not considered. 3 The person is completely surrounded by the interior surfaces of the room. 4 The surrounding surfaces are at a uniform temperature.

Properties The emissivity of a person is given to be $\varepsilon = 0.95$

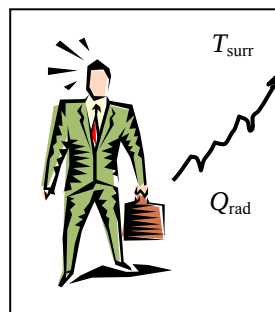
Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are:

(a) Summer: $T_{\text{surr}} = 23 + 273 = 296$

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.95)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1.6 \text{ m}^2)[(32 + 273)^4 - (296 \text{ K})^4] \text{ K}^4 \\ &= \mathbf{84.2 \text{ W}}\end{aligned}$$

(b) Winter: $T_{\text{surr}} = 12 + 273 = 285 \text{ K}$

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.95)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1.6 \text{ m}^2)[(32 + 273)^4 - (285 \text{ K})^4] \text{ K}^4 \\ &= \mathbf{177.2 \text{ W}}\end{aligned}$$



Discussion Note that the radiation heat transfer from the person more than doubles in winter.

9-30 EES Prob. 9-29 is reconsidered. The rate of radiation heat transfer in winter as a function of the temperature of the inner surface of the room is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

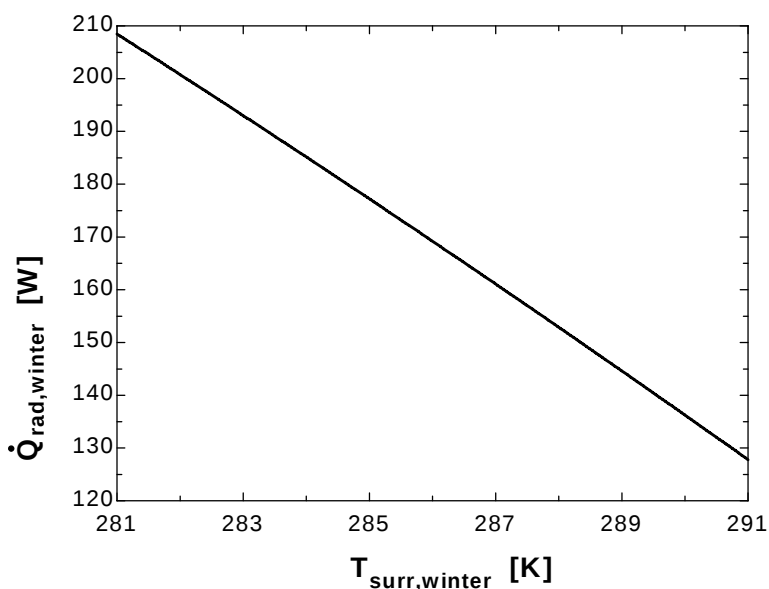
"GIVEN"

$T_{\infty} = (20 + 273) \text{ [K]}$
 $T_{\text{surr, winter}} = (12 + 273) \text{ [K]}$
 $T_{\text{surr, summer}} = (23 + 273) \text{ [K]}$
 $A = 1.6 \text{ [m}^2\text{]}$
 $\epsilon = 0.95$
 $T_s = (32 + 273) \text{ [K]}$

"ANALYSIS"

$\sigma = 5.67 \times 10^{-8} \text{ [W/m}^2\text{-K}^4\text{]}$ "Stefan-Boltzman constant"
 $\dot{Q}_{\text{rad, summer}} = \epsilon \sigma A (T_s^4 - T_{\text{surr, summer}}^4)$
 $\dot{Q}_{\text{rad, winter}} = \epsilon \sigma A (T_s^4 - T_{\text{surr, winter}}^4)$

$T_{\text{surr, winter}} \text{ [K]}$	$\dot{Q}_{\text{rad, winter}} \text{ [W]}$
281	208.5
282	200.8
283	193
284	185.1
285	177.2
286	169.2
287	161.1
288	152.9
289	144.6
290	136.2
291	127.8



9-31 A person is standing in a room at a specified temperature. The rate of heat transfer between a person and the surrounding air by convection is to be determined.

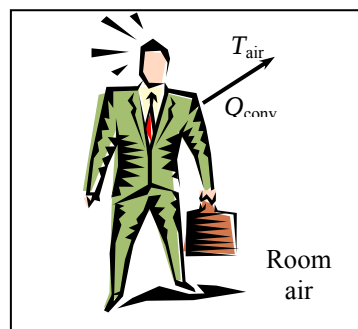
Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is not considered. 3 The environment is at a uniform temperature.

Analysis The heat transfer surface area of the person is

$$A_s = \pi DL = \pi(0.3 \text{ m})(1.70 \text{ m}) = 1.602 \text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA_s \Delta T = (20 \text{ W/m}^2 \cdot ^\circ\text{C})(1.602 \text{ m}^2)(34 - 18)^\circ\text{C} = \mathbf{513 \text{ W}}$$

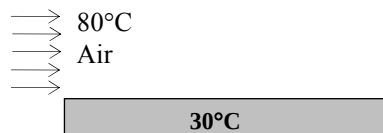


9-32 Hot air is blown over a flat surface at a specified temperature. The rate of heat transfer from the air to the plate is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is not considered. 3 The convection heat transfer coefficient is constant and uniform over the surface.

Analysis Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA_s \Delta T = (55 \text{ W/m}^2 \cdot ^\circ\text{C})(2 \times 4 \text{ m}^2)(80 - 30)^\circ\text{C} = \mathbf{22,000 \text{ W}}$$



9-33 EES Prob. 9-32 is reconsidered. The rate of heat transfer as a function of the heat transfer coefficient is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$T_{\text{infinity}}=80 \text{ [C]}$$

$$A=2*4 \text{ [m}^2\text{]}$$

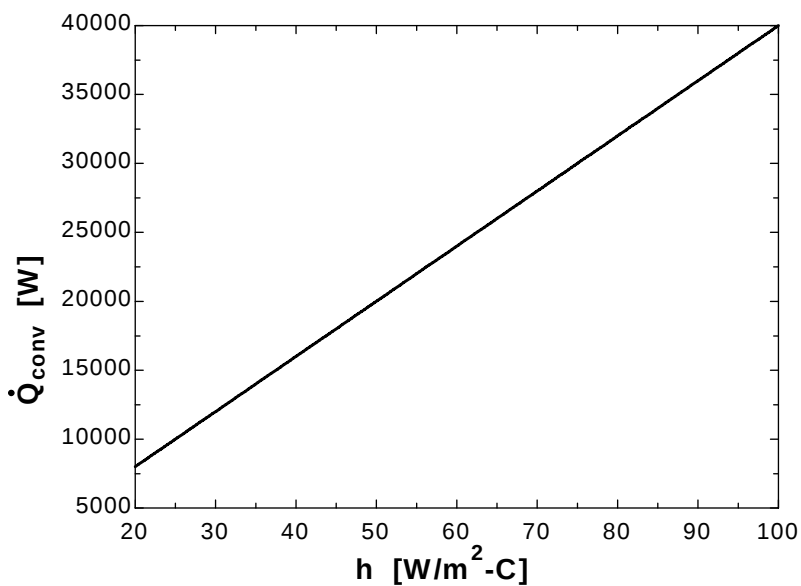
$$T_s=30 \text{ [C]}$$

$$h=55 \text{ [W/m}^2\text{-C]}$$

"ANALYSIS"

$$\dot{Q}_{\text{conv}}=h*A*(T_{\text{infinity}}-T_s)$$

$h \text{ [W/m}^2\text{-C]}$	$\dot{Q}_{\text{conv}} \text{ [W]}$
20	8000
30	12000
40	16000
50	20000
60	24000
70	28000
80	32000
90	36000
100	40000



9-34 The heat generated in the circuitry on the surface of a 3-W silicon chip is conducted to the ceramic substrate. The temperature difference across the chip in steady operation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Thermal properties of the chip are constant.

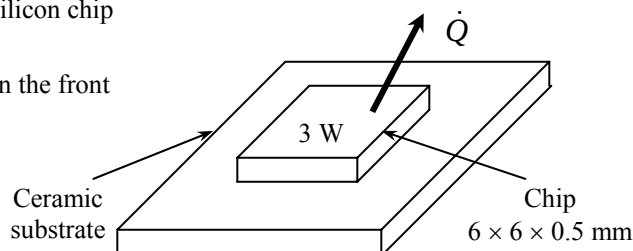
Properties The thermal conductivity of the silicon chip is given to be $k = 130 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The temperature difference between the front and back surfaces of the chip is

$$A = (0.006 \text{ m})(0.006 \text{ m}) = 0.000036 \text{ m}^2$$

$$\dot{Q} = kA \frac{\Delta T}{L}$$

$$\Delta T = \frac{\dot{Q}L}{kA} = \frac{(3 \text{ W})(0.0005 \text{ m})}{(130 \text{ W/m}\cdot^\circ\text{C})(0.000036 \text{ m}^2)} = \mathbf{0.32^\circ\text{C}}$$



9-35 An electric resistance heating element is immersed in water initially at 20°C . The time it will take for this heater to raise the water temperature to 80°C as well as the convection heat transfer coefficients at the beginning and at the end of the heating process are to be determined.

Assumptions 1 Steady operating conditions exist and thus the rate of heat loss from the wire equals the rate of heat generation in the wire as a result of resistance heating. 2 Thermal properties of water are constant. 3 Heat losses from the water in the tank are negligible.

Properties The specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-15).

Analysis When steady operating conditions are reached, we have $\dot{Q} = \dot{E}_{\text{generated}} = 800 \text{ W}$. This is also equal to the rate of heat gain by water. Noting that this is the only mechanism of energy transfer, the time it takes to raise the water temperature from 20°C to 80°C is determined to be

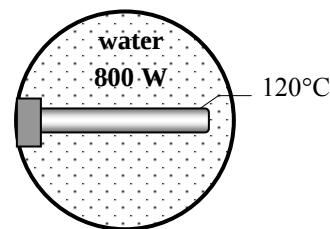
$$Q_{\text{in}} = mc(T_2 - T_1)$$

$$\dot{Q}_{\text{in}} \Delta t = mc(T_2 - T_1)$$

$$\Delta t = \frac{mc(T_2 - T_1)}{\dot{Q}_{\text{in}}} = \frac{(75 \text{ kg})(4180 \text{ J/kg}\cdot^\circ\text{C})(80 - 20)^\circ\text{C}}{800 \text{ J/s}} = 23,510 \text{ s} = \mathbf{6.53 \text{ h}}$$

The surface area of the wire is

$$A_s = \pi DL = \pi(0.005 \text{ m})(0.4 \text{ m}) = 0.00628 \text{ m}^2$$



The Newton's law of cooling for convection heat transfer is expressed as $\dot{Q} = hA_s(T_s - T_\infty)$. Disregarding any heat transfer by radiation and thus assuming all the heat loss from the wire to occur by convection, the convection heat transfer coefficients at the beginning and at the end of the process are determined to be

$$h_1 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 1})} = \frac{800 \text{ W}}{(0.00628 \text{ m}^2)(120 - 20)^\circ\text{C}} = \mathbf{1274 \text{ W/m}^2\cdot^\circ\text{C}}$$

$$h_2 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 2})} = \frac{800 \text{ W}}{(0.00628 \text{ m}^2)(120 - 80)^\circ\text{C}} = \mathbf{3185 \text{ W/m}^2\cdot^\circ\text{C}}$$

Discussion Note that a larger heat transfer coefficient is needed to dissipate heat through a smaller temperature difference for a specified heat transfer rate.

9-36 A hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/m²·°C. The rate of heat loss from the pipe by convection is to be determined.

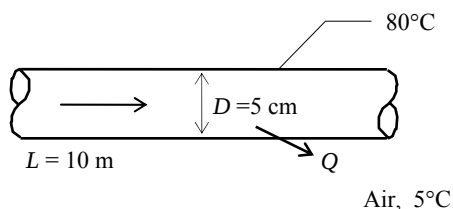
Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is not considered. 3 The convection heat transfer coefficient is constant and uniform over the surface.

Analysis The heat transfer surface area is

$$A_s = \pi DL = \pi (0.05 \text{ m})(10 \text{ m}) = 1.571 \text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA_s \Delta T = (25 \text{ W/m}^2 \cdot ^\circ\text{C})(1.571 \text{ m}^2)(80 - 5)^\circ\text{C} = \mathbf{2945 \text{ W}}$$



9-37 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. 2 Heat transfer through the shell is one-dimensional. 3 Thermal properties of the iron shell are constant. 4 The inner surface of the shell is at the same temperature as the iced water, 0°C.

Properties The thermal conductivity of iron is $k = 80.2 \text{ W/m} \cdot ^\circ\text{C}$ (Table A-24). The heat of fusion of water is given to be 333.7 kJ/kg.

Analysis This spherical shell can be approximated as a plate of thickness 0.4 cm and area

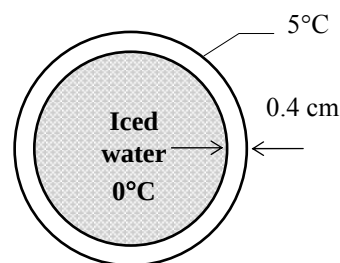
$$A = \pi D^2 = \pi (0.2 \text{ m})^2 = 0.126 \text{ m}^2$$

Then the rate of heat transfer through the shell by conduction is

$$\dot{Q}_{\text{cond}} = kA \frac{\Delta T}{L} = (80.2 \text{ W/m} \cdot ^\circ\text{C})(0.126 \text{ m}^2) \frac{(5 - 0)^\circ\text{C}}{0.004 \text{ m}} = 12,632 \text{ W}$$

Considering that it takes 333.7 kJ of energy to melt 1 kg of ice at 0°C, the rate at which ice melts in the container can be determined from

$$\dot{m}_{\text{ice}} = \frac{\dot{Q}}{h_{if}} = \frac{12.632 \text{ kJ/s}}{333.7 \text{ kJ/kg}} = \mathbf{0.038 \text{ kg/s}}$$



Discussion We should point out that this result is slightly in error for approximating a curved wall as a plain wall. The error in this case is very small because of the large diameter to thickness ratio. For better accuracy, we could use the inner surface area ($D = 19.2 \text{ cm}$) or the mean surface area ($D = 19.6 \text{ cm}$) in the calculations.

9-38 EES Prob. 9-37 is reconsidered. The rate at which ice melts as a function of the container thickness is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$D=0.2 \text{ [m]}$$

$$L=0.4 \text{ [cm]}$$

$$T_1=0 \text{ [C]}$$

$$T_2=5 \text{ [C]}$$

"PROPERTIES"

$$h_{if}=333.7 \text{ [kJ/kg]}$$

$$k=k_{\text{'Iron', 25}}$$

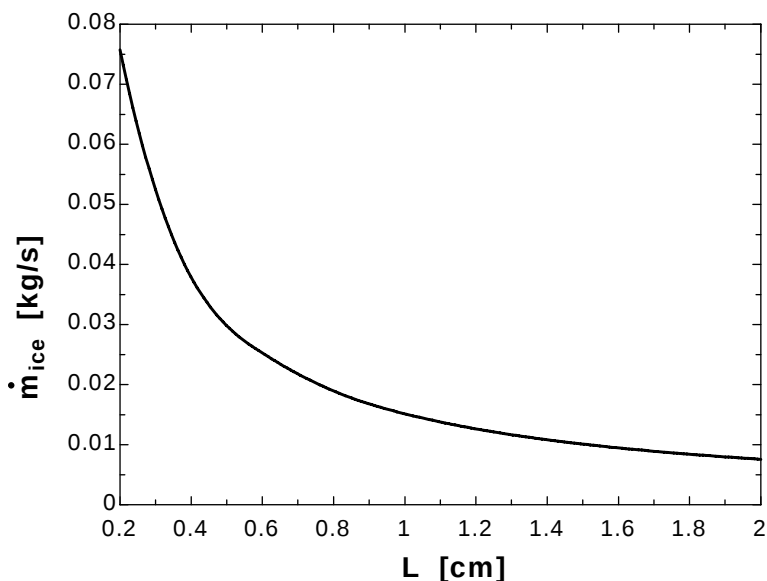
"ANALYSIS"

$$A=\pi D^2$$

$$Q_{\text{dot_cond}}=kA(T_2-T_1)/(L*\text{Convert(cm, m)})$$

$$\dot{m}_{\text{ice}}=(Q_{\text{dot_cond}}*\text{Convert(W, kW)})/h_{if}$$

L [cm]	$\dot{m}_{\text{ice}}$ [kg/s]
0.2	0.07574
0.4	0.03787
0.6	0.02525
0.8	0.01894
1	0.01515
1.2	0.01262
1.4	0.01082
1.6	0.009468
1.8	0.008416
2	0.007574



9-39E The inner and outer glasses of a double pane window with a 0.5-in air space are at specified temperatures. The rate of heat transfer through the window is to be determined

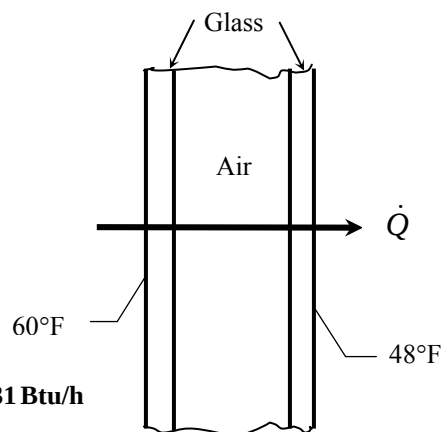
Assumptions 1 Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. 2 Heat transfer through the window is one-dimensional. 3 Thermal properties of the air are constant.

Properties The thermal conductivity of air at the average temperature of $(60+48)/2 = 54^\circ\text{F}$ is $k = 0.01419 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ (Table A-22E).

Analysis The area of the window and the rate of heat loss through it are

$$A = (4 \text{ ft}) \times (4 \text{ ft}) = 16 \text{ m}^2$$

$$\dot{Q} = kA \frac{T_1 - T_2}{L} = (0.01419 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(16 \text{ ft}^2) \frac{(60 - 48)^\circ\text{F}}{0.25/12 \text{ ft}} = \mathbf{131 \text{ Btu/h}}$$

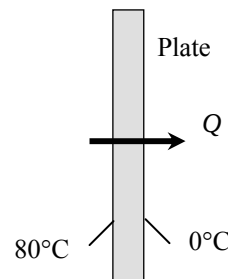


9-40 Two surfaces of a flat plate are maintained at specified temperatures, and the rate of heat transfer through the plate is measured. The thermal conductivity of the plate material is to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the plate remain constant at the specified values. 2 Heat transfer through the plate is one-dimensional. 3 Thermal properties of the plate are constant.

Analysis The thermal conductivity is determined directly from the steady one-dimensional heat conduction relation to be

$$\dot{Q} = kA \frac{T_1 - T_2}{L} \rightarrow k = \frac{(\dot{Q}/A)L}{(T_1 - T_2)} = \frac{(500 \text{ W/m}^2)(0.02 \text{ m})}{(80 - 0)^\circ\text{C}} = \mathbf{0.125 \text{ W/m}\cdot^\circ\text{C}}$$



9-41 Four power transistors are mounted on a thin vertical aluminum plate that is cooled by a fan. The temperature of the aluminum plate is to be determined.

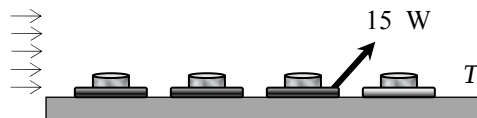
Assumptions 1 Steady operating conditions exist. 2 The entire plate is nearly isothermal. 3 Thermal properties of the wall are constant. 4 The exposed surface area of the transistor can be taken to be equal to its base area. 5 Heat transfer by radiation is disregarded. 6 The convection heat transfer coefficient is constant and uniform over the surface.

Analysis The total rate of heat dissipation from the aluminum plate and the total heat transfer area are

$$\dot{Q} = 4 \times 15 \text{ W} = 60 \text{ W}$$

$$A_s = (0.22 \text{ m})(0.22 \text{ m}) = 0.0484 \text{ m}^2$$

Disregarding any radiation effects, the temperature of the aluminum plate is determined to be



$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 25^\circ\text{C} + \frac{60 \text{ W}}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0484 \text{ m}^2)} = \mathbf{74.6^\circ\text{C}}$$

9-42 A styrofoam ice chest is initially filled with 40 kg of ice at 0°C. The time it takes for the ice in the chest to melt completely is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The inner and outer surface temperatures of the ice chest remain constant at 0°C and 8°C, respectively, at all times. 3 Thermal properties of the chest are constant. 4 Heat transfer from the base of the ice chest is negligible.

Properties The thermal conductivity of the styrofoam is given to be $k = 0.033 \text{ W/m} \cdot ^\circ\text{C}$. The heat of fusion of ice at 0°C is 333.7 kJ/kg.

Analysis Disregarding any heat loss through the bottom of the ice chest and using the average thicknesses, the total heat transfer area becomes

$$A = (40 - 3)(40 - 3) + 4 \times (40 - 3)(30 - 3) = 5365 \text{ cm}^2 = 0.5365 \text{ m}^2$$

The rate of heat transfer to the ice chest becomes

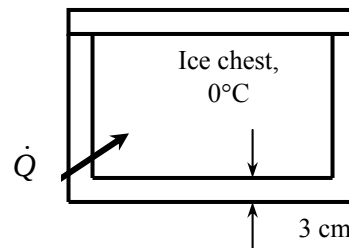
$$\dot{Q} = kA \frac{\Delta T}{L} = (0.033 \text{ W/m} \cdot ^\circ\text{C})(0.5365 \text{ m}^2) \frac{(8 - 0)^\circ\text{C}}{0.03 \text{ m}} = 4.72 \text{ W}$$

The total amount of heat needed to melt the ice completely is

$$Q = mh_{if} = (28 \text{ kg})(333.7 \text{ kJ/kg}) = 9344 \text{ kJ}$$

Then transferring this much heat to the cooler to melt the ice completely will take

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{9344,000 \text{ J}}{4.72 \text{ J/s}} = 1.98 \times 10^6 \text{ s} = \mathbf{22.9 \text{ days}}$$



9-43 A transistor mounted on a circuit board is cooled by air flowing over it. The transistor case temperature is not to exceed 70°C when the air temperature is 55°C. The amount of power this transistor can dissipate safely is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is disregarded. 3 The convection heat transfer coefficient is constant and uniform over the surface. 4 Heat transfer from the base of the transistor is negligible.

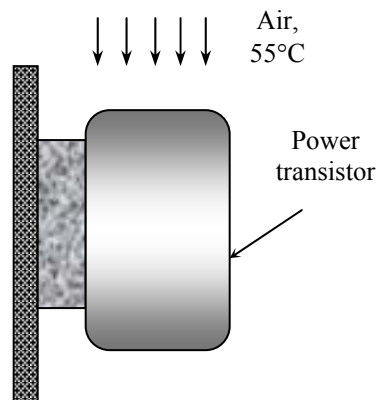
Analysis Disregarding the base area, the total heat transfer area of the transistor is

$$\begin{aligned} A_s &= \pi DL + \pi D^2 / 4 \\ &= \pi(0.6 \text{ cm})(0.4 \text{ cm}) + \pi(0.6 \text{ cm})^2 / 4 = 1.037 \text{ cm}^2 \\ &= 1.037 \times 10^{-4} \text{ m}^2 \end{aligned}$$

Then the rate of heat transfer from the power transistor at specified conditions is

$$\dot{Q} = hA_s(T_s - T_\infty) = (30 \text{ W/m}^2 \cdot ^\circ\text{C})(1.037 \times 10^{-4} \text{ m}^2)(70 - 55)^\circ\text{C} = \mathbf{0.047 \text{ W}}$$

Therefore, the amount of power this transistor can dissipate safely is 0.047 W.



9-44 EES Prob. 9-43 is reconsidered. The amount of power the transistor can dissipate safely as a function of the maximum case temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$L=0.004 \text{ [m]}$$

$$D=0.006 \text{ [m]}$$

$$h=30 \text{ [W/m}^2\text{-C]}$$

$$T_{\text{infinity}}=55 \text{ [C]}$$

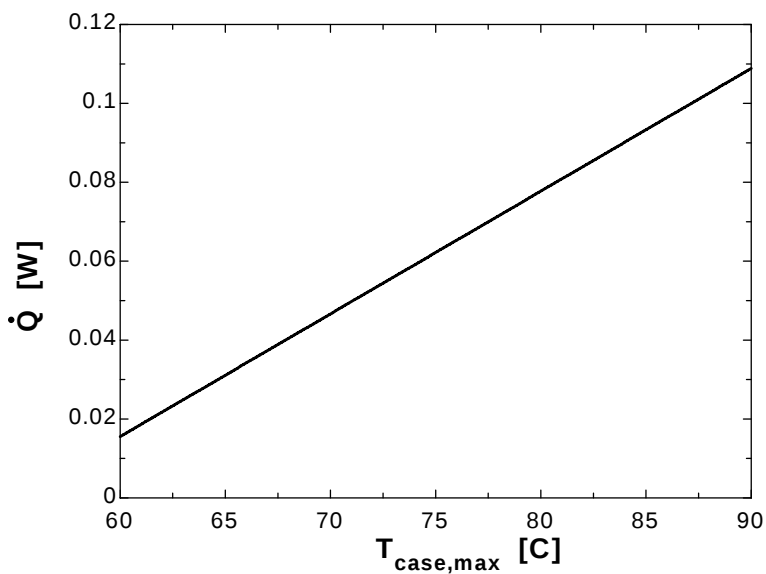
$$T_{\text{case_max}}=70 \text{ [C]}$$

"ANALYSIS"

$$A=\pi D L + \pi D^2/4$$

$$\dot{Q}_{\text{dot}}=h A (T_{\text{case_max}} - T_{\text{infinity}})$$

$T_{\text{case, max}} \text{ [C]}$	$\dot{Q} \text{ [W]}$
60	0.01555
62.5	0.02333
65	0.0311
67.5	0.03888
70	0.04665
72.5	0.05443
75	0.0622
77.5	0.06998
80	0.07775
82.5	0.08553
85	0.09331
87.5	0.1011
90	0.1089



9-45E A 200-ft long section of a steam pipe passes through an open space at a specified temperature. The rate of heat loss from the steam pipe and the annual cost of this energy lost are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is disregarded. 3 The convection heat transfer coefficient is constant and uniform over the surface.

Analysis (a) The rate of heat loss from the steam pipe is

$$A_s = \pi DL = \pi(4/12 \text{ ft})(200 \text{ ft}) = 209.4 \text{ ft}^2$$

$$\begin{aligned}\dot{Q}_{\text{pipe}} &= hA_s(T_s - T_{\text{air}}) = (6 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(209.4 \text{ ft}^2)(280 - 50)^\circ\text{F} \\ &= \mathbf{289,000 \text{ Btu/h}}\end{aligned}$$

(b) The amount of heat loss per year is

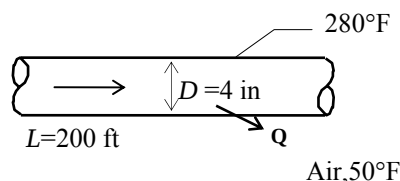
$$Q = \dot{Q}\Delta t = (289,000 \text{ Btu/h})(365 \times 24 \text{ h/yr}) = 2.531 \times 10^9 \text{ Btu/yr}$$

The amount of gas consumption per year in the furnace that has an efficiency of 86% is

$$\text{Annual Energy Loss} = \frac{2.531 \times 10^9 \text{ Btu/yr}}{0.86} \left(\frac{1 \text{ therm}}{100,000 \text{ Btu}} \right) = 29,435 \text{ therms/yr}$$

Then the annual cost of the energy lost becomes

$$\begin{aligned}\text{Energy cost} &= (\text{Annual energy loss})(\text{Unit cost of energy}) \\ &= (29,435 \text{ therms/yr})(\$1.10 / \text{therm}) = \mathbf{\$32,380/\text{yr}}\end{aligned}$$



9-46 A 4-m diameter spherical tank filled with liquid nitrogen at 1 atm and -196°C is exposed to convection with ambient air. The rate of evaporation of liquid nitrogen in the tank as a result of the heat transfer from the ambient air is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is disregarded. 3 The convection heat transfer coefficient is constant and uniform over the surface. 4 The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the nitrogen inside.

Properties The heat of vaporization and density of liquid nitrogen at 1 atm are given to be 198 kJ/kg and 810 kg/m^3 , respectively.

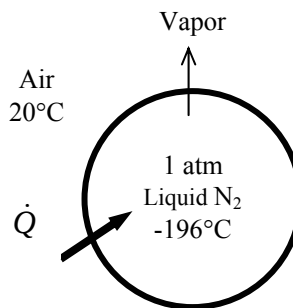
Analysis The rate of heat transfer to the nitrogen tank is

$$A_s = \pi D^2 = \pi(4 \text{ m})^2 = 50.27 \text{ m}^2$$

$$\begin{aligned}\dot{Q} &= hA_s(T_s - T_{\text{air}}) = (25 \text{ W/m}^2 \cdot ^\circ\text{C})(50.27 \text{ m}^2)[20 - (-196)]^\circ\text{C} \\ &= \mathbf{271,430 \text{ W}}\end{aligned}$$

Then the rate of evaporation of liquid nitrogen in the tank is determined to be

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{271.430 \text{ kJ/s}}{198 \text{ kJ/kg}} = \mathbf{1.37 \text{ kg/s}}$$



9-47 A 4-m diameter spherical tank filled with liquid oxygen at 1 atm and -183°C is exposed to convection with ambient air. The rate of evaporation of liquid oxygen in the tank as a result of the heat transfer from the ambient air is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by radiation is disregarded. 3 The convection heat transfer coefficient is constant and uniform over the surface. 4 The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the oxygen inside.

Properties The heat of vaporization and density of liquid oxygen at 1 atm are given to be 213 kJ/kg and 1140 kg/m^3 , respectively.

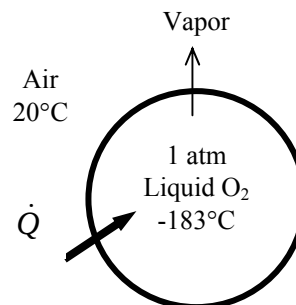
Analysis The rate of heat transfer to the oxygen tank is

$$A_s = \pi D^2 = \pi (4\text{ m})^2 = 50.27\text{ m}^2$$

$$\begin{aligned}\dot{Q} &= hA_s(T_s - T_{\text{air}}) = (25\text{ W/m}^2 \cdot ^{\circ}\text{C})(50.27\text{ m}^2)[20 - (-183)]^{\circ}\text{C} \\ &= \mathbf{255,120\text{ W}}\end{aligned}$$

Then the rate of evaporation of liquid oxygen in the tank is determined to be

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{255.120\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{1.20\text{ kg/s}}$$



9-48 EES Prob. 9-46 is reconsidered. The rate of evaporation of liquid nitrogen as a function of the ambient air temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$D=4$ [m]

$T_s=-196$ [C]

$T_{air}=20$ [C]

$h=25$ [W/m²-C]

"PROPERTIES"

$h_{fg}=198$ [kJ/kg]

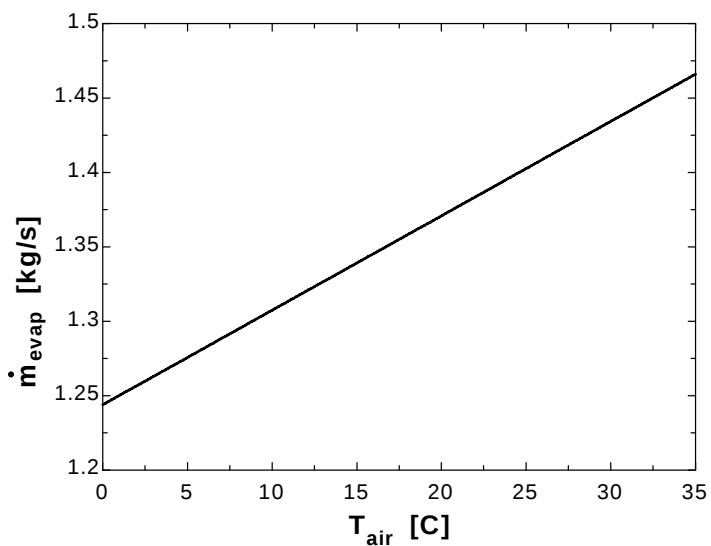
"ANALYSIS"

$A=\pi \cdot D^2$

$\dot{Q}=h \cdot A \cdot (T_{air}-T_s)$

$\dot{m}_{dot_evap}=(\dot{Q} \cdot \text{Convert(J/s, kJ/s)})/h_{fg}$

T_{air} [C]	$\dot{m}_{evap}$ [kg/s]
0	1.244
2.5	1.26
5	1.276
7.5	1.292
10	1.307
12.5	1.323
15	1.339
17.5	1.355
20	1.371
22.5	1.387
25	1.403
27.5	1.418
30	1.434
32.5	1.45
35	1.466



9-49 A person with a specified surface temperature is subjected to radiation heat transfer in a room at specified wall temperatures. The rate of radiation heat loss from the person is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by convection is disregarded. 3 The emissivity of the person is constant and uniform over the exposed surface.

Properties The average emissivity of the person is given to be 0.5.

Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are

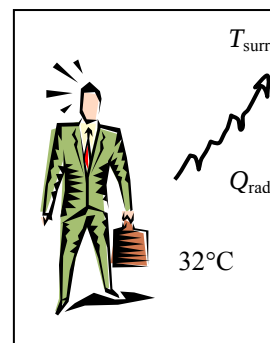
(a) $T_{\text{surr}} = 300 \text{ K}$

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.5)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1.7 \text{ m}^2)[(32 + 273)^4 - (300 \text{ K})^4] \text{ K}^4 \\ &= \mathbf{26.7 \text{ W}}\end{aligned}$$

(b) $T_{\text{surr}} = 280 \text{ K}$

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.5)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1.7 \text{ m}^2)[(32 + 273)^4 - (280 \text{ K})^4] \text{ K}^4 \\ &= \mathbf{121 \text{ W}}\end{aligned}$$

Discussion Note that the radiation heat transfer goes up by more than 4 times as the temperature of the surrounding surfaces drops from 300 K to 280 K.



9-50 A circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.06 W. All the heat generated in the chips is conducted across the circuit board. The temperature difference between the two sides of the circuit board is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Thermal properties of the board are constant. 3 All the heat generated in the chips is conducted across the circuit board.

Properties The effective thermal conductivity of the board is given to be $k = 16 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The total rate of heat dissipated by the chips is

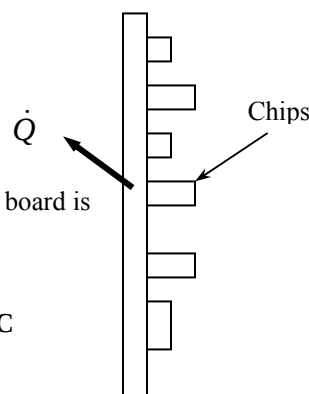
$$\dot{Q} = 80 \times (0.06 \text{ W}) = 4.8 \text{ W}$$

Then the temperature difference between the front and back surfaces of the board is

$$A = (0.12 \text{ m})(0.18 \text{ m}) = 0.0216 \text{ m}^2$$

$$\dot{Q} = kA \frac{\Delta T}{L} \longrightarrow \Delta T = \frac{\dot{Q}L}{kA} = \frac{(4.8 \text{ W})(0.003 \text{ m})}{(16 \text{ W/m} \cdot ^\circ\text{C})(0.0216 \text{ m}^2)} = \mathbf{0.042^\circ\text{C}}$$

Discussion Note that the circuit board is nearly isothermal.



9-51 A sealed electronic box dissipating a total of 100 W of power is placed in a vacuum chamber. If this box is to be cooled by radiation alone and the outer surface temperature of the box is not to exceed 55°C, the temperature the surrounding surfaces must be kept is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer by convection is disregarded. 3 The emissivity of the box is constant and uniform over the exposed surface. 4 Heat transfer from the bottom surface of the box to the stand is negligible.

Properties The emissivity of the outer surface of the box is given to be 0.95.

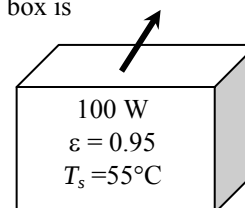
Analysis Disregarding the base area, the total heat transfer area of the electronic box is

$$A_s = (0.4 \text{ m})(0.4 \text{ m}) + 4 \times (0.2 \text{ m})(0.4 \text{ m}) = 0.48 \text{ m}^2$$

The radiation heat transfer from the box can be expressed as

$$\dot{Q}_{\text{rad}} = \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4)$$

$$100 \text{ W} = (0.95)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(0.48 \text{ m}^2) \left[(55 + 273 \text{ K})^4 - T_{\text{surr}}^4 \right]$$



which gives $T_{\text{surr}} = 296.3 \text{ K} = 23.3^\circ\text{C}$. Therefore, the temperature of the surrounding surfaces must be less than 23.3°C.

9-52E Using the conversion factors between W and Btu/h, m and ft, and K and R, the Stefan-Boltzmann constant $\sigma = 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4$ is to be expressed in the English unit, $\text{Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4$.

Analysis The conversion factors for W, m, and K are given in conversion tables to be

$$1 \text{ W} = 3.41214 \text{ Btu/h}$$

$$1 \text{ m} = 3.2808 \text{ ft}$$

$$1 \text{ K} = 1.8 \text{ R}$$

Substituting gives the Stefan-Boltzmann constant in the desired units,

$$\sigma = 5.67 \text{ W/m}^2 \cdot \text{K}^4 = 5.67 \times \frac{3.41214 \text{ Btu/h}}{(3.2808 \text{ ft})^2 (1.8 \text{ R})^4} = 0.171 \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4$$

9-53E Using the conversion factors between W and Btu/h, m and ft, and °C and °F, the convection coefficient in SI units is to be expressed in Btu/h·ft²·°F.

Analysis The conversion factors for W and m are straightforward, and are given in conversion tables to be

$$1 \text{ W} = 3.41214 \text{ Btu/h}$$

$$1 \text{ m} = 3.2808 \text{ ft}$$

The proper conversion factor between °C into °F in this case is

$$1^\circ\text{C} = 1.8^\circ\text{F}$$

since the °C in the unit W/m²·°C represents *per °C change in temperature*, and 1°C change in temperature corresponds to a change of 1.8°F. Substituting, we get

$$1 \text{ W/m}^2 \cdot ^\circ\text{C} = \frac{3.41214 \text{ Btu/h}}{(3.2808 \text{ ft})^2 (1.8^\circ\text{F})} = 0.1761 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

which is the desired conversion factor. Therefore, the given convection heat transfer coefficient in English units is

$$h = 14 \text{ W/m}^2 \cdot ^\circ\text{C} = 14 \times 0.1761 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F} = \mathbf{2.47 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}}$$

9-54 A cylindrical sample of a material is used to determine its thermal conductivity. The temperatures measured along the sample are tabulated. The variation of temperature along the sample is to be plotted and the thermal conductivity of the sample material is to be calculated.

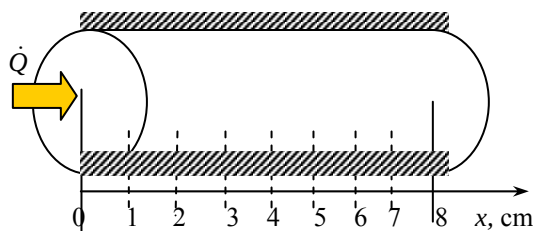
Assumptions 1 Steady operating conditions exist. 2 Heat transfer is one-dimensional (axial direction).

Analysis The following table gives the results of the calculations. The plot of temperatures is also given below. A sample calculation for the thermal conductivity is as follows:

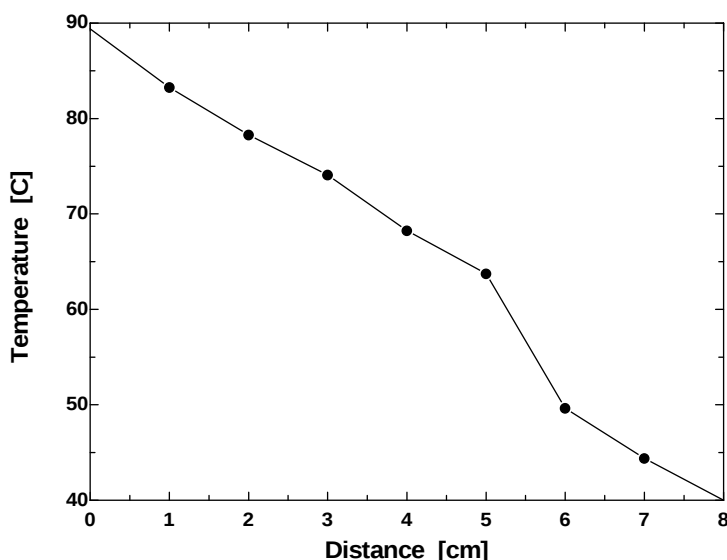
$$A = \frac{\pi D^2}{4} = \frac{\pi (0.025 \text{ m})^2}{4} = 0.00049 \text{ m}^2$$

$$k_{12} = \frac{\dot{Q}L}{A(T_1 - T_2)}$$

$$= \frac{(83.45 \text{ W})(0.010 \text{ m})}{(0.00049 \text{ m}^2)(6.13^\circ\text{C})} = 277.8 \text{ W/m}\cdot^\circ\text{C}$$



Distance from left face, cm	Temperature, °C	Temperature difference (°C)	Thermal conductivity (W/m·°C)
0	T ₁ = 89.38	T ₉ -T ₂ = 6.13	277.8
1	T ₂ = 83.25	T ₂ -T ₃ = 4.97	342.7
2	T ₃ = 78.28	T ₃ -T ₄ = 4.18	407.4
3	T ₄ = 74.10	T ₄ -T ₅ = 5.85	291.1
4	T ₅ = 68.25	T ₅ -T ₆ = 4.52	376.8
5	T ₆ = 63.73	T ₆ -T ₇ = 14.08	120.9
6	T ₇ = 49.65	T ₇ -T ₈ = 5.25	324.4
7	T ₈ = 44.40	T ₈ -T ₉ = 4.40	387.1
8	T ₉ = 40.00	T ₉ -T ₂ = 6.13	277.8



Discussion It is observed from the calculations in the table and the plot of temperatures that the temperature reading corresponding to the calculated thermal conductivity of 120.9 is probably not right, and it should be discarded.

9-55 An aircraft flying under icing conditions is considered. The temperature of the wings to prevent ice from forming on them is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer coefficient is constant.

Properties The heat of fusion and the density of ice are given to be 333.7 kJ/kg and 920 kg/m³, respectively.

Analysis The temperature of the wings to prevent ice from forming on them is determined to be

$$T_{\text{wing}} = T_{\text{ice}} + \frac{\rho V h_{if}}{h} = 0^{\circ}\text{C} + \frac{(920 \text{ kg/m}^3)(0.001/60 \text{ m/s})(333,700 \text{ J/kg})}{150 \text{ W/m}^2 \cdot ^{\circ}\text{C}} = \mathbf{34.1^{\circ}\text{C}}$$

Simultaneous Heat Transfer Mechanisms

9-56C All three modes of heat transfer can not occur simultaneously in a medium. A medium may involve two of them simultaneously.

9-57C (a) Conduction and convection: No. (b) Conduction and radiation: Yes. Example: A hot surface on the ceiling. (c) Convection and radiation: Yes. Example: Heat transfer from the human body.

9-58C The human body loses heat by convection, radiation, and evaporation in both summer and winter. In summer, we can keep cool by dressing lightly, staying in cooler environments, turning a fan on, avoiding humid places and direct exposure to the sun. In winter, we can keep warm by dressing heavily, staying in a warmer environment, and avoiding drafts.

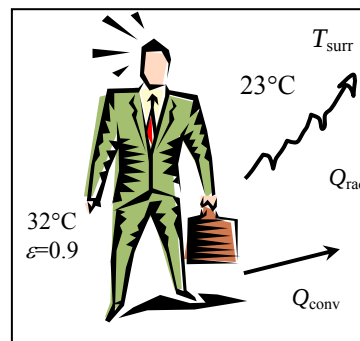
9-59C The fan increases the air motion around the body and thus the convection heat transfer coefficient, which increases the rate of heat transfer from the body by convection and evaporation. In rooms with high ceilings, ceiling fans are used in winter to force the warm air at the top downward to increase the air temperature at the body level. This is usually done by forcing the air up which hits the ceiling and moves downward in a gently manner to avoid drafts.

9-60 The total rate of heat transfer from a person by both convection and radiation to the surrounding air and surfaces at specified temperatures is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The person is completely surrounded by the interior surfaces of the room. 3 The surrounding surfaces are at the same temperature as the air in the room. 4 Heat conduction to the floor through the feet is negligible. 5 The convection coefficient is constant and uniform over the entire surface of the person.

Properties The emissivity of a person is given to be $\varepsilon = 0.9$.

Analysis The person is completely enclosed by the surrounding surfaces, and he or she will lose heat to the surrounding air by convection and to the surrounding surfaces by radiation. The total rate of heat loss from the person is determined from



$$\dot{Q}_{\text{rad}} = \varepsilon \alpha A_s (T_s^4 - T_{\text{surr}}^4) = (0.90)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1.7 \text{ m}^2)[(32 + 273)^4 - (23 + 273)^4] \text{ K}^4 = 84.8 \text{ W}$$

$$\dot{Q}_{\text{conv}} = h A_s \Delta T = (5 \text{ W/m}^2 \cdot \text{K})(1.7 \text{ m}^2)(32 - 23)^\circ\text{C} = 76.5 \text{ W}$$

$$\text{and } \dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 84.8 + 76.5 = \mathbf{161.3 \text{ W}}$$

Discussion Note that heat transfer from the person by evaporation, which is of comparable magnitude, is not considered in this problem.

9-61 Two large plates at specified temperatures are held parallel to each other. The rate of heat transfer between the plates is to be determined for the cases of still air, evacuation, regular insulation, and super insulation between the plates.

Assumptions 1 Steady operating conditions exist since the plate temperatures remain constant. 2 Heat transfer is one-dimensional since the plates are large. 3 The surfaces are black and thus $\varepsilon = 1$. 4 There are no convection currents in the air space between the plates.

Properties The thermal conductivities are $k = 0.00015 \text{ W/m} \cdot ^\circ\text{C}$ for super insulation, $k = 0.01979 \text{ W/m} \cdot ^\circ\text{C}$ at -50°C (Table A-22) for air, and $k = 0.036 \text{ W/m} \cdot ^\circ\text{C}$ for fiberglass insulation.

Analysis (a) Disregarding any natural convection currents, the rates of conduction and radiation heat transfer

$$\dot{Q}_{\text{cond}} = kA \frac{T_1 - T_2}{L} = (0.01979 \text{ W/m}^2 \cdot ^\circ\text{C})(1 \text{ m}^2) \frac{(290 - 150) \text{ K}}{0.02 \text{ m}} = 139 \text{ W}$$

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_1^4 - T_2^4) \\ &= 1(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1 \text{ m}^2) [(290 \text{ K})^4 - (150 \text{ K})^4] = 372 \text{ W} \end{aligned}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} + \dot{Q}_{\text{rad}} = 139 + 372 = \mathbf{511 \text{ W}}$$

(b) When the air space between the plates is evacuated, there will be radiation heat transfer only. Therefore,

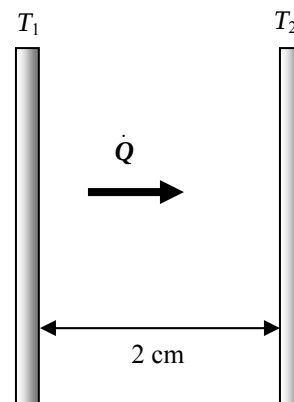
$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{rad}} = \mathbf{372 \text{ W}}$$

(c) In this case there will be conduction heat transfer through the fiberglass insulation only,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} = kA \frac{T_1 - T_2}{L} = (0.036 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m}^2) \frac{(290 - 150) \text{ K}}{0.02 \text{ m}} = \mathbf{252 \text{ W}}$$

(d) In the case of superinsulation, the rate of heat transfer will be

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} = kA \frac{T_1 - T_2}{L} = (0.00015 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m}^2) \frac{(290 - 150) \text{ K}}{0.02 \text{ m}} = \mathbf{1.05 \text{ W}}$$



Discussion Note that superinsulators are very effective in reducing heat transfer between to surfaces.

9-62 The outer surface of a wall is exposed to solar radiation. The effective thermal conductivity of the wall is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat transfer coefficient is constant and uniform over the surface.

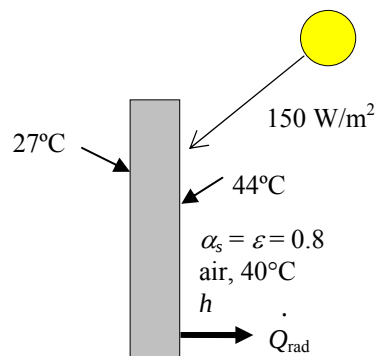
Properties Both the solar absorptivity and emissivity of the wall surface are given to be 0.8.

Analysis The heat transfer through the wall by conduction is equal to net heat transfer to the outer wall surface:

$$\begin{aligned}\dot{q}_{\text{cond}} &= \dot{q}_{\text{conv}} + \dot{q}_{\text{rad}} + \dot{q}_{\text{solar}} \\ k \frac{T_2 - T_1}{L} &= h(T_o - T_2) + \varepsilon\sigma(T_{\text{surr}}^4 - T_2^4) + \alpha_s q_{\text{solar}} \\ k \frac{(44 - 27)^\circ\text{C}}{0.25 \text{ m}} &= (8 \text{ W/m}^2 \cdot ^\circ\text{C})(40 - 44)^\circ\text{C} + (0.8)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(40 + 273 \text{ K})^4 - (44 + 273 \text{ K})^4] \\ &\quad + (0.8)(150 \text{ W/m}^2)\end{aligned}$$

Solving for k gives

$$k = \mathbf{0.961 \text{ W/m} \cdot ^\circ\text{C}}$$



9-63 The convection heat transfer coefficient for heat transfer from an electrically heated wire to air is to be determined by measuring temperatures when steady operating conditions are reached and the electric power consumed.

Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. 2 Radiation heat transfer is negligible.

Analysis In steady operation, the rate of heat loss from the wire equals the rate of heat generation in the wire as a result of resistance heating. That is,

$$\dot{Q} = \dot{E}_{\text{generated}} = \mathbf{VI = (110 \text{ V})(3 \text{ A}) = 330 \text{ W}}$$

The surface area of the wire is

$$A_s = \pi DL = \pi(0.002 \text{ m})(1.4 \text{ m}) = 0.00880 \text{ m}^2$$

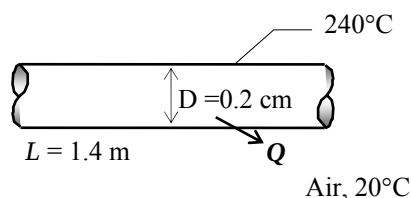
The Newton's law of cooling for convection heat transfer is expressed as

$$\dot{Q} = hA_s(T_s - T_\infty)$$

Disregarding any heat transfer by radiation, the convection heat transfer coefficient is determined to be

$$h = \frac{\dot{Q}}{A_s(T_s - T_\infty)} = \frac{330 \text{ W}}{(0.00880 \text{ m}^2)(240 - 20)^\circ\text{C}} = \mathbf{170.5 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

Discussion If the temperature of the surrounding surfaces is equal to the air temperature in the room, the value obtained above actually represents the combined convection and radiation heat transfer coefficient.



9-64 EES Prob. 9-63 is reconsidered. The convection heat transfer coefficient as a function of the wire surface temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$L=1.4$ [m]

$D=0.002$ [m]

$T_{\infty}=20$ [C]

$T_s=240$ [C]

$V=110$ [Volt]

$I=3$ [Ampere]

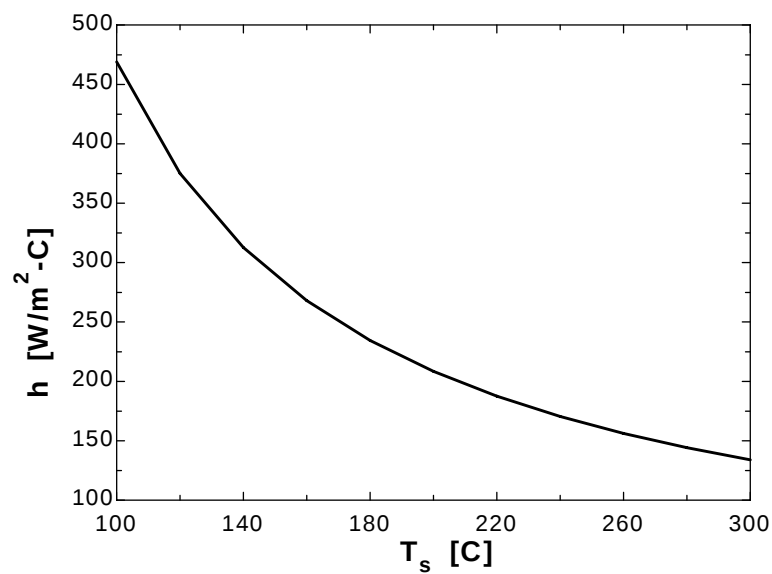
"ANALYSIS"

$\dot{Q}=V \cdot I$

$A=\pi \cdot D \cdot L$

$\dot{Q}=h \cdot A \cdot (T_s - T_{\infty})$

T_s [C]	h [W/m ² .C]
100	468.9
120	375.2
140	312.6
160	268
180	234.5
200	208.4
220	187.6
240	170.5
260	156.3
280	144.3
300	134



9-65E A spherical ball whose surface is maintained at a temperature of 170°F is suspended in the middle of a room at 70°F. The total rate of heat transfer from the ball is to be determined.

Assumptions 1 Steady operating conditions exist since the ball surface and the surrounding air and surfaces remain at constant temperatures. 2 The thermal properties of the ball and the convection heat transfer coefficient are constant and uniform.

Properties The emissivity of the ball surface is given to be $\varepsilon = 0.8$.

Analysis The heat transfer surface area is

$$A_s = \pi D^2 = \pi (2/12 \text{ ft})^2 = 0.08727 \text{ ft}^2$$

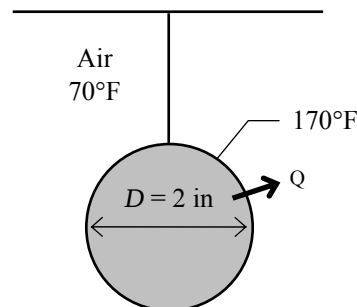
Under steady conditions, the rates of convection and radiation heat transfer are

$$\dot{Q}_{\text{conv}} = hA_s \Delta T = (15 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(0.08727 \text{ ft}^2)(170 - 70)^\circ\text{F} = 130.9 \text{ Btu/h}$$

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_o^4) \\ &= 0.8(0.08727 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)[(170 + 460 \text{ R})^4 - (70 + 460 \text{ R})^4] \\ &= 9.4 \text{ Btu/h} \end{aligned}$$

Therefore, $\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 130.9 + 9.4 = \mathbf{140.3 \text{ Btu/h}}$

Discussion Note that heat loss by convection is several times that of heat loss by radiation. The radiation heat loss can further be reduced by coating the ball with a low-emissivity material.



9-66 CD EES A 1000-W iron is left on the iron board with its base exposed to the air at 20°C. The temperature of the base of the iron is to be determined in steady operation.

Assumptions 1 Steady operating conditions exist. 2 The thermal properties of the iron base and the convection heat transfer coefficient are constant and uniform. 3 The temperature of the surrounding surfaces is the same as the temperature of the surrounding air.

Properties The emissivity of the base surface is given to be $\varepsilon = 0.6$.

Analysis At steady conditions, the 1000 W energy supplied to the iron will be dissipated to the surroundings by convection and radiation heat transfer. Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 1000 \text{ W}$$

where $\dot{Q}_{\text{conv}} = hA_s \Delta T = (35 \text{ W/m}^2 \cdot \text{K})(0.02 \text{ m}^2)(T_s - 293 \text{ K}) = 0.7(T_s - 293 \text{ K})$

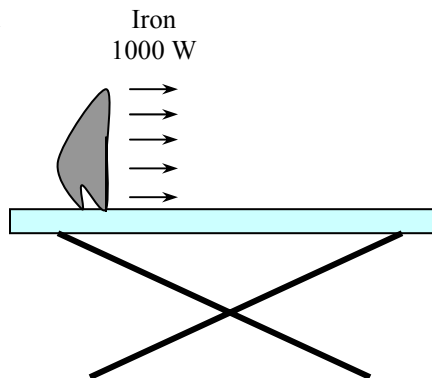
and $\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_o^4) = 0.6(0.02 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_s^4 - (293 \text{ K})^4] \\ &= 0.06804 \times 10^{-8} [T_s^4 - (293 \text{ K})^4] \end{aligned}$

Substituting, $1000 \text{ W} = 0.7(T_s - 293 \text{ K}) + 0.06804 \times 10^{-8} [T_s^4 - (293 \text{ K})^4]$

Solving by trial and error gives

$$T_s = \mathbf{947 \text{ K} = 674^\circ\text{C}}$$

Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K.



9-67 A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. The surface temperature of the spacecraft is to be determined when steady conditions are reached.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. 2 Thermal properties of the wall are constant.

Properties The outer surface of a spacecraft has an emissivity of 0.8 and an absorptivity of 0.3.

Analysis When the heat loss from the outer surface of the spacecraft by radiation equals the solar radiation absorbed, the surface temperature can be determined from

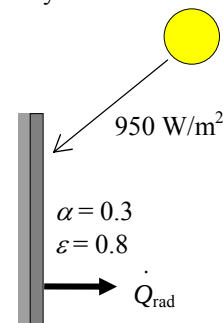
$$\dot{Q}_{\text{solar absorbed}} = \dot{Q}_{\text{rad}}$$

$$\alpha \dot{Q}_{\text{solar}} = \varepsilon \sigma A_s (T_s^4 - T_{\text{space}}^4)$$

$$0.3 \times A_s \times (950 \text{ W/m}^2) = 0.8 \times A_s \times (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [T_s^4 - (0 \text{ K})^4]$$

Canceling the surface area A and solving for T_s gives

$$T_s = 281.5 \text{ K}$$



9-68 A spherical tank located outdoors is used to store iced water at 0°C . The rate of heat transfer to the iced water in the tank and the amount of ice at 0°C that melts during a 24-h period are to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. 2 Thermal properties of the tank and the convection heat transfer coefficient is constant and uniform. 3 The average surrounding surface temperature for radiation exchange is 15°C . 4 The thermal resistance of the tank is negligible, and the entire steel tank is at 0°C .

Properties The heat of fusion of water at atmospheric pressure is $h_{if} = 333.7 \text{ kJ/kg}$. The emissivity of the outer surface of the tank is 0.75.

Analysis (a) The outer surface area of the spherical tank is

$$A_s = \pi D^2 = \pi (3.02 \text{ m})^2 = 28.65 \text{ m}^2$$

Then the rates of heat transfer to the tank by convection and radiation become

$$\dot{Q}_{\text{conv}} = h A_s (T_\infty - T_s) = (30 \text{ W/m}^2 \cdot ^\circ\text{C})(28.65 \text{ m}^2)(25 - 0)^\circ\text{C} = 21,488 \text{ W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) = (0.75)(28.65 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(288 \text{ K})^4 - (273 \text{ K})^4] = 1614 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 21,488 + 1614 = 23,102 \text{ W} = \mathbf{23.1 \text{ kW}}$$

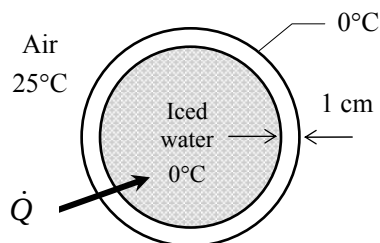
(b) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q} \Delta t = (23.102 \text{ kJ/s})(24 \times 3600 \text{ s}) = 1,996,000 \text{ kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = m h_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{1,996,000 \text{ kJ}}{333.7 \text{ kJ/kg}} = \mathbf{5980 \text{ kg}}$$

Discussion The amount of ice that melts can be reduced to a small fraction by insulating the tank.



9-69 CD EES The roof of a house with a gas furnace consists of a 15-cm thick concrete that is losing heat to the outdoors by radiation and convection. The rate of heat transfer through the roof and the money lost through the roof that night during a 14 hour period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The emissivity and thermal conductivity of the roof are constant.

Properties The thermal conductivity of the concrete is given to be $k = 2 \text{ W/m}\cdot^\circ\text{C}$. The emissivity of the outer surface of the roof is given to be 0.9.

Analysis In steady operation, heat transfer from the outer surface of the roof to the surroundings by convection and radiation must be equal to the heat transfer through the roof by conduction. That is,

$$\dot{Q} = \dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{roof to surroundings, conv+rad}}$$

The inner surface temperature of the roof is given to be $T_{s,\text{in}} = 15^\circ\text{C}$. Letting $T_{s,\text{out}}$ denote the outer surface temperatures of the roof, the energy balance above can be expressed as

$$\dot{Q} = kA \frac{T_{s,\text{in}} - T_{s,\text{out}}}{L} = h_o A (T_{s,\text{out}} - T_{\text{surr}}) + \varepsilon A \sigma (T_{s,\text{out}}^4 - T_{\text{surr}}^4)$$

$$\begin{aligned} \dot{Q} &= (2 \text{ W/m}\cdot^\circ\text{C})(300 \text{ m}^2) \frac{15^\circ\text{C} - T_{s,\text{out}}}{0.15 \text{ m}} \\ &= (15 \text{ W/m}^2\cdot^\circ\text{C})(300 \text{ m}^2)(T_{s,\text{out}} - 10)^\circ\text{C} \\ &\quad + (0.9)(300 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(T_{s,\text{out}} + 273 \text{ K})^4 - (255 \text{ K})^4] \end{aligned}$$

Solving the equations above using an equation solver (or by trial and error) gives

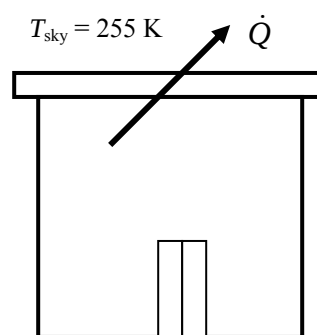
$$\dot{Q} = 25,450 \text{ W} \text{ and } T_{s,\text{out}} = 8.64^\circ\text{C}$$

Then the amount of natural gas consumption during a 9-hour period is

$$E_{\text{gas}} = \frac{Q_{\text{total}}}{0.85} = \frac{\dot{Q} \Delta t}{0.85} = \frac{(25.450 \text{ kJ/s})(14 \times 3600 \text{ s})}{0.85} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 14.3 \text{ therms}$$

Finally, the money lost through the roof during that period is

$$\text{Money lost} = (14.3 \text{ therms})(\$0.60 / \text{therm}) = \mathbf{\$8.58}$$



9-70E A flat plate solar collector is placed horizontally on the roof of a house. The rate of heat loss from the collector by convection and radiation during a calm day are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The emissivity and convection heat transfer coefficient are constant and uniform. 3 The exposed surface, ambient, and sky temperatures remain constant.

Properties The emissivity of the outer surface of the collector is given to be 0.9.

Analysis The exposed surface area of the collector is

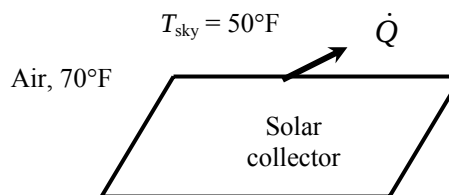
$$A_s = (5 \text{ ft})(15 \text{ ft}) = 75 \text{ ft}^2$$

Noting that the exposed surface temperature of the collector is 100°F , the total rate of heat loss from the collector to the environment by convection and radiation becomes

$$\dot{Q}_{\text{conv}} = hA_s (T_\infty - T_s) = (2.5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(75 \text{ ft}^2)(100 - 70)^\circ\text{F} = 5625 \text{ Btu/h}$$

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) = (0.9)(75 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}^4) [(100 + 460 \text{ R})^4 - (50 + 460 \text{ R})^4] \\ &= 3551 \text{ Btu/h} \end{aligned}$$

$$\text{and } \dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5625 + 3551 = \mathbf{9176 \text{ Btu/h}}$$



Review Problems

9-71 A standing man is subjected to high winds and thus high convection coefficients. The rate of heat loss from this man by convection in still air at 20°C, in windy air, and the wind-chill factor are to be determined.

Assumptions **1** A standing man can be modeled as a 30-cm diameter, 170-cm long vertical cylinder with both the top and bottom surfaces insulated. **2** The exposed surface temperature of the person and the convection heat transfer coefficient is constant and uniform. **3** Heat loss by radiation is negligible.

Analysis The heat transfer surface area of the person is

$$A_s = \pi DL = \pi(0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2$$

The rate of heat loss from this man by convection in still air is

$$Q_{\text{still air}} = hA_s \Delta T = (15 \text{ W/m}^2 \cdot ^\circ\text{C})(1.60 \text{ m}^2)(34 - 20)^\circ\text{C} = \mathbf{336 \text{ W}}$$

In windy air it would be

$$Q_{\text{windy air}} = hA_s \Delta T = (50 \text{ W/m}^2 \cdot ^\circ\text{C})(1.60 \text{ m}^2)(34 - 20)^\circ\text{C} = \mathbf{1120 \text{ W}}$$

To lose heat at this rate in still air, the air temperature must be

$$1120 \text{ W} = (hA_s \Delta T)_{\text{still air}} = (15 \text{ W/m}^2 \cdot ^\circ\text{C})(1.60 \text{ m}^2)(34 - T_{\text{effective}})^\circ\text{C}$$

which gives

$$T_{\text{effective}} = -12.7^\circ\text{C}$$

That is, the windy air at 20°C feels as cold as still air at -12.7°C as a result of the wind-chill effect. Therefore, the wind-chill factor in this case is

$$F_{\text{wind-chill}} = 20 - (-12.7) = \mathbf{32.7^\circ\text{C}}$$



Windy weather

9-72 The backside of the thin metal plate is insulated and the front side is exposed to solar radiation. The surface temperature of the plate is to be determined when it stabilizes.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the insulated side of the plate is negligible. **3** The heat transfer coefficient is constant and uniform over the plate. **4** Radiation heat transfer is negligible.

Properties The solar absorptivity of the plate is given to be $\alpha = 0.7$.

Analysis When the heat loss from the plate by convection equals the solar radiation absorbed, the surface temperature of the plate can be determined from

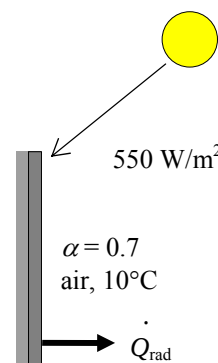
$$\dot{Q}_{\text{solar absorbed}} = \dot{Q}_{\text{conv}}$$

$$\alpha \dot{Q}_{\text{solar}} = hA_s (T_s - T_o)$$

$$0.7 \times A \times 550 \text{ W/m}^2 = (25 \text{ W/m}^2 \cdot ^\circ\text{C})A_s (T_s - 10)$$

Canceling the surface area A_s and solving for T_s gives

$$T_s = \mathbf{25.4^\circ\text{C}}$$



9-73 A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it is to meet the heating requirements of this room for a 24-h period.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

Properties The specific heat of water at room temperature is $c = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-15).

Analysis Heat loss from the room during a 24-h period is

$$Q_{\text{loss}} = (10,000 \text{ kJ/h})(24 \text{ h}) = 240,000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}} \rightarrow -Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}} \approx 0$$

or

$$-Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

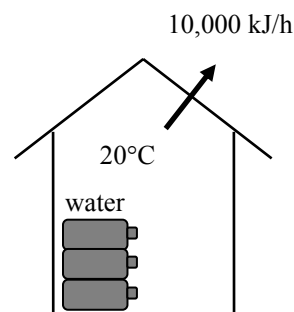
Substituting,

$$-240,000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(20 - T_1)$$

It gives

$$T_1 = 77.4^\circ\text{C}$$

where T_1 is the temperature of the water when it is first brought into the room.



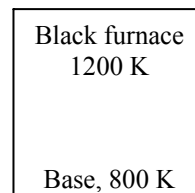
9-74 The base surface of a cubical furnace is surrounded by black surfaces at a specified temperature. The net rate of radiation heat transfer to the base surface from the top and side surfaces is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The top and side surfaces of the furnace closely approximate black surfaces. **3** The properties of the surfaces are constant.

Properties The emissivity of the base surface is $\varepsilon = 0.7$.

Analysis The base surface is completely surrounded by the top and side surfaces. Then using the radiation relation for a surface completely surrounded by another large (or black) surface, the net rate of radiation heat transfer from the top and side surfaces to the base is determined to be

$$\begin{aligned} \dot{Q}_{\text{rad,base}} &= \varepsilon A \sigma (T_{\text{base}}^4 - T_{\text{surr}}^4) \\ &= (0.7)(3 \times 3 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(1200 \text{ K})^4 - (800 \text{ K})^4] \\ &= 594,400 \text{ W} = 594 \text{ kW} \end{aligned}$$



9-75 A refrigerator consumes 600 W of power when operating, and its motor remains on for 5 min and then off for 15 min periodically. The average thermal conductivity of the refrigerator walls and the annual cost of operating this refrigerator are to be determined.

Assumptions 1 Quasi-steady operating conditions exist. 2 The inner and outer surface temperatures of the refrigerator remain constant.

Analysis The total surface area of the refrigerator where heat transfer takes place is

$$A_{\text{total}} = 2[(1.8 \times 1.2) + (1.8 \times 0.8) + (1.2 \times 0.8)] = 9.12 \text{ m}^2$$

Since the refrigerator has a COP of 1.5, the rate of heat removal from the refrigerated space, which is equal to the rate of heat gain in steady operation, is

$$\dot{Q} = \dot{W}_e \times \text{COP} = (600 \text{ W}) \times 1.5 = 900 \text{ W}$$

But the refrigerator operates a quarter of the time (5 min on, 15 min off). Therefore, the average rate of heat gain is

$$\dot{Q}_{\text{ave}} = \dot{Q} / 4 = (900 \text{ W}) / 4 = 225 \text{ W}$$

Then the thermal conductivity of refrigerator walls is determined to be

$$\dot{Q}_{\text{ave}} = kA \frac{\Delta T_{\text{ave}}}{L} \longrightarrow k = \frac{\dot{Q}_{\text{ave}} L}{A \Delta T_{\text{ave}}} = \frac{(225 \text{ W})(0.03 \text{ m})}{(9.12 \text{ m}^2)(17 - 6)^\circ\text{C}} = \mathbf{0.0673 \text{ W/m} \cdot ^\circ\text{C}}$$

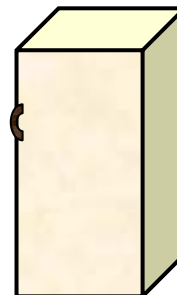
The total number of hours this refrigerator remains on per year is

$$\Delta t = 365 \times 24 / 4 = 2190 \text{ h}$$

Then the total amount of electricity consumed during a one-year period and the annular cost of operating this refrigerator are

$$\text{Annual Electricity Usage} = \dot{W}_e \Delta t = (0.6 \text{ kW})(2190 \text{ h/yr}) = 1314 \text{ kWh/yr}$$

$$\text{Annual cost} = (1314 \text{ kWh/yr})(\$0.08 / \text{kWh}) = \mathbf{\$105.1/\text{yr}}$$



9-76 Engine valves are to be heated in a heat treatment section. The amount of heat transfer, the average rate of heat transfer, the average heat flux, and the number of valves that can be heat treated daily are to be determined.

Assumptions Constant properties given in the problem can be used.

Properties The average specific heat and density of valves are given to be $c_p = 440 \text{ J/kg} \cdot ^\circ\text{C}$ and $\rho = 7840 \text{ kg/m}^3$.

Analysis (a) The amount of heat transferred to the valve is simply the change in its internal energy, and is determined from

$$Q = \Delta U = mc_p(T_2 - T_1) \\ = (0.0788 \text{ kg})(0.440 \text{ kJ/kg} \cdot ^\circ\text{C})(800 - 40)^\circ\text{C} = \mathbf{26.35 \text{ kJ}}$$

(b) The average rate of heat transfer can be determined from

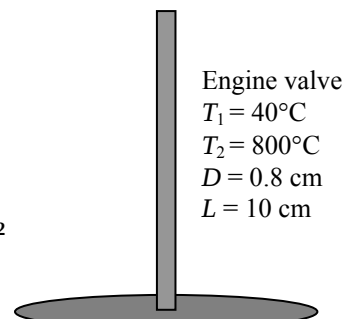
$$\dot{Q}_{\text{avg}} = \frac{Q}{\Delta t} = \frac{26.35 \text{ kJ}}{5 \times 60 \text{ s}} = 0.0878 \text{ kW} = \mathbf{87.8 \text{ W}}$$

(c) The average heat flux is determined from

$$\dot{q}_{\text{ave}} = \frac{\dot{Q}_{\text{avg}}}{A_s} = \frac{\dot{Q}_{\text{avg}}}{2\pi DL} = \frac{87.8 \text{ W}}{2\pi(0.008 \text{ m})(0.1 \text{ m})} = \mathbf{1.75 \times 10^4 \text{ W/m}^2}$$

(d) The number of valves that can be heat treated daily is

$$\text{Number of valves} = \frac{(10 \times 60 \text{ min})(25 \text{ valves})}{5 \text{ min}} = \mathbf{3000 \text{ valves}}$$



9-77 The glass cover of a flat plate solar collector with specified inner and outer surface temperatures is considered. The fraction of heat lost from the glass cover by radiation is to be determined.

Assumptions 1 Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. 2 Thermal properties of the glass are constant.

Properties The thermal conductivity of the glass is given to be $k = 0.7 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is

$$\dot{Q}_{\text{cond}} = kA \frac{\Delta T}{L} = (0.7 \text{ W/m} \cdot ^\circ\text{C})(2.5 \text{ m}^2) \frac{(28 - 25)^\circ\text{C}}{0.006 \text{ m}} = 875 \text{ W}$$

The rate of heat transfer from the glass by convection is

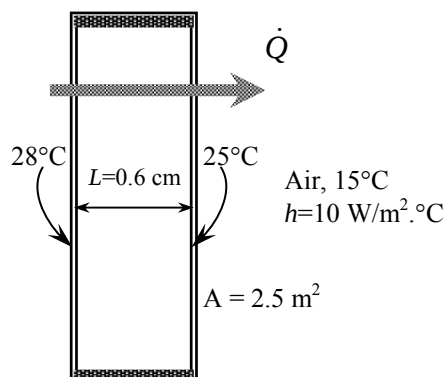
$$\dot{Q}_{\text{conv}} = hA\Delta T = (10 \text{ W/m}^2 \cdot ^\circ\text{C})(2.5 \text{ m}^2)(25 - 15)^\circ\text{C} = 250 \text{ W}$$

Under steady conditions, the heat transferred through the cover by conduction should be transferred from the outer surface by convection and radiation. That is,

$$\dot{Q}_{\text{rad}} = \dot{Q}_{\text{cond}} - \dot{Q}_{\text{conv}} = 875 - 250 = 625 \text{ W}$$

Then the fraction of heat transferred by radiation becomes

$$f = \frac{\dot{Q}_{\text{rad}}}{\dot{Q}_{\text{cond}}} = \frac{625}{875} = \mathbf{0.714} \quad (\text{or } 71.4\%)$$



9-78 The range of U-factors for windows are given. The range for the rate of heat loss through the window of a house is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses associated with the infiltration of air through the cracks/openings are not considered.

Analysis The rate of heat transfer through the window can be determined from

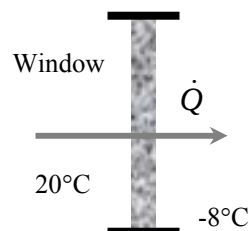
$$\dot{Q}_{\text{window}} = U_{\text{overall}} A_{\text{window}} (T_i - T_o)$$

where T_i and T_o are the indoor and outdoor air temperatures, respectively, U_{overall} is the U-factor (the overall heat transfer coefficient) of the window, and A_{window} is the window area. Substituting,

Maximum heat loss: $\dot{Q}_{\text{window, max}} = (6.25 \text{ W/m}^2 \cdot ^\circ\text{C})(1.2 \times 1.8 \text{ m}^2)[20 - (-8)]^\circ\text{C} = \mathbf{378 \text{ W}}$

Minimum heat loss: $\dot{Q}_{\text{window, min}} = (1.25 \text{ W/m}^2 \cdot ^\circ\text{C})(1.2 \times 1.8 \text{ m}^2)[20 - (-8)]^\circ\text{C} = \mathbf{76 \text{ W}}$

Discussion Note that the rate of heat loss through windows of identical size may differ by a factor of 5, depending on how the windows are constructed.



9-79 EES Prob. 9-78 is reconsidered. The rate of heat loss through the window as a function of the U -factor is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$A = 1.2 \times 1.8 \text{ [m}^2\text{]}$$

$$T_1 = 20 \text{ [C]}$$

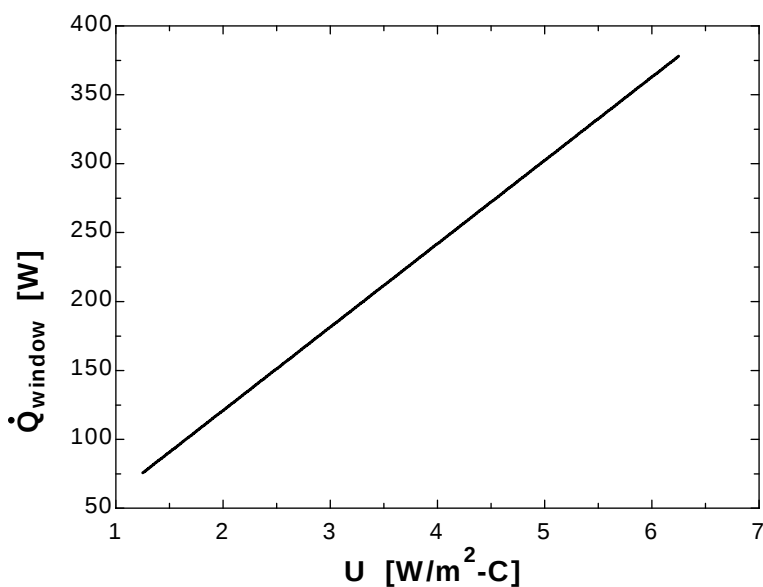
$$T_2 = -8 \text{ [C]}$$

$$U = 1.25 \text{ [W/m}^2\text{-C]}$$

"ANALYSIS"

$$\dot{Q}_{\text{dot_window}} = U \cdot A \cdot (T_1 - T_2)$$

U [W/m ² .C]	Q _{window} [W]
1.25	75.6
1.75	105.8
2.25	136.1
2.75	166.3
3.25	196.6
3.75	226.8
4.25	257
4.75	287.3
5.25	317.5
5.75	347.8
6.25	378



9-80 The windows of a house in Atlanta are of double door type with wood frames and metal spacers. The average rate of heat loss through the windows in winter is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses associated with the infiltration of air through the cracks/openings are not considered.

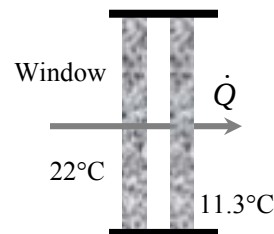
Analysis The rate of heat transfer through the window can be determined from

$$\dot{Q}_{\text{window, avg}} = U_{\text{overall}} A_{\text{window}} (T_i - T_o)$$

where T_i and T_o are the indoor and outdoor air temperatures, respectively, U_{overall} is the U -factor (the overall heat transfer coefficient) of the window, and A_{window} is the window area. Substituting,

$$\dot{Q}_{\text{window, avg}} = (2.50 \text{ W/m}^2 \cdot ^\circ\text{C})(20 \text{ m}^2)(22 - 11.3)^\circ\text{C} = \mathbf{535 \text{ W}}$$

Discussion This is the “average” rate of heat transfer through the window in winter in the absence of any infiltration.



9-81 Boiling experiments are conducted by heating water at 1 atm pressure with an electric resistance wire, and measuring the power consumed by the wire as well as temperatures. The boiling heat transfer coefficient is to be determined.

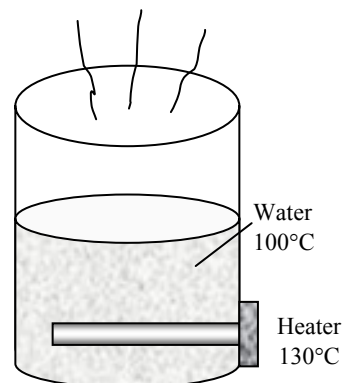
Assumptions 1 Steady operating conditions exist. 2 Heat losses from the water container are negligible.

Analysis The heat transfer area of the heater wire is

$$A = \pi DL = \pi(0.002 \text{ m})(0.50 \text{ m}) = 0.003142 \text{ m}^2$$

Noting that 4100 W of electric power is consumed when the heater surface temperature is 130°C , the boiling heat transfer coefficient is determined from Newton's law of cooling $\dot{Q} = hA(T_s - T_{\text{sat}})$ to be

$$h = \frac{\dot{Q}}{A(T_s - T_{\text{sat}})} = \frac{4100 \text{ W}}{(0.003142 \text{ m}^2)(130 - 100)^\circ\text{C}} = \mathbf{43,500 \text{ W/m}^2 \cdot ^\circ\text{C}}$$



9-82 An electric heater placed in a room consumes 500 W power when its surfaces are at 120°C. The surface temperature when the heater consumes 700 W is to be determined without and with the consideration of radiation.

Assumptions 1 Steady operating conditions exist. 2 The temperature is uniform over the surface.

Analysis (a) Neglecting radiation, the convection heat transfer coefficient is determined from

$$h = \frac{\dot{Q}}{A(T_s - T_\infty)} = \frac{500 \text{ W}}{(0.25 \text{ m}^2)(120 - 20)^\circ\text{C}} = 20 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The surface temperature when the heater consumes 700 W is

$$T_s = T_\infty + \frac{\dot{Q}}{hA} = 20^\circ\text{C} + \frac{700 \text{ W}}{(20 \text{ W/m}^2 \cdot ^\circ\text{C})(0.25 \text{ m}^2)} = \mathbf{160^\circ\text{C}}$$

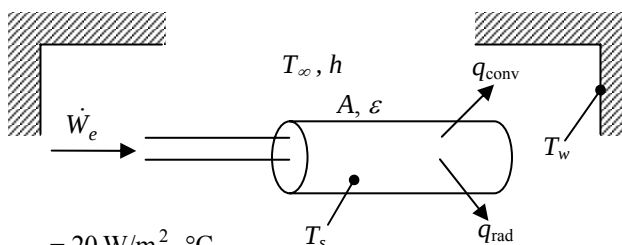
(b) Considering radiation, the convection heat transfer coefficient is determined from

$$\begin{aligned} h &= \frac{\dot{Q} - \varepsilon A \sigma (T_s^4 - T_{\text{surr}}^4)}{A(T_s - T_\infty)} \\ &= \frac{500 \text{ W} - (0.75)(0.25 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(393 \text{ K})^4 - (283 \text{ K})^4]}{(0.25 \text{ m}^2)(120 - 20)^\circ\text{C}} = 12.58 \text{ W/m}^2 \cdot ^\circ\text{C} \end{aligned}$$

Then the surface temperature becomes

$$\begin{aligned} \dot{Q} &= hA(T_s - T_\infty) + \varepsilon A \sigma (T_s^4 - T_{\text{surr}}^4) \\ 700 &= (12.58)(0.25)(T_s - 293) + (0.75)(0.25)(5.67 \times 10^{-8})[T_s^4 - (283 \text{ K})^4] \\ T_s &= 425.9 \text{ K} = \mathbf{152.9^\circ\text{C}} \end{aligned}$$

Discussion Neglecting radiation changed T_s by more than 7°C, so assumption is not correct in this case.



9-83 . . . 9-85 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 10
STEADY HEAT CONDUCTION

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Steady Heat Conduction in Plane Walls

10-1C (a) If the lateral surfaces of the rod are insulated, the heat transfer surface area of the cylindrical rod is the bottom or the top surface area of the rod, $A_s = \pi D^2 / 4$. (b) If the top and the bottom surfaces of the rod are insulated, the heat transfer area of the rod is the lateral surface area of the rod, $A = \pi DL$.

10-2C In steady heat conduction, the rate of heat transfer into the wall is equal to the rate of heat transfer out of it. Also, the temperature at any point in the wall remains constant. Therefore, the energy content of the wall does not change during steady heat conduction. However, the temperature along the wall and thus the energy content of the wall will change during transient conduction.

10-3C The temperature distribution in a plane wall will be a straight line during steady and one dimensional heat transfer with constant wall thermal conductivity.

10-4C The thermal resistance of a medium represents the resistance of that medium against heat transfer.

10-5C The combined heat transfer coefficient represents the combined effects of radiation and convection heat transfers on a surface, and is defined as $h_{\text{combined}} = h_{\text{convection}} + h_{\text{radiation}}$. It offers the convenience of incorporating the effects of radiation in the convection heat transfer coefficient, and to ignore radiation in heat transfer calculations.

10-6C Yes. The convection resistance can be defined as the inverse of the convection heat transfer coefficient per unit surface area since it is defined as $R_{\text{conv}} = 1/(hA)$.

10-7C The convection and the radiation resistances at a surface are parallel since both the convection and radiation heat transfers occur simultaneously.

10-8C For a surface of A at which the convection and radiation heat transfer coefficients are h_{conv} and h_{rad} , the single equivalent heat transfer coefficient is $h_{\text{eqv}} = h_{\text{conv}} + h_{\text{rad}}$ when the medium and the surrounding surfaces are at the same temperature. Then the equivalent thermal resistance will be $R_{\text{eqv}} = 1/(h_{\text{eqv}}A)$.

10-9C The thermal resistance network associated with a five-layer composite wall involves five single-layer resistances connected in series.

10-10C Once the rate of heat transfer $\dot{Q}$ is known, the temperature drop across any layer can be determined by multiplying heat transfer rate by the thermal resistance across that layer, $\Delta T_{\text{layer}} = \dot{Q}R_{\text{layer}}$

10-11C The temperature of each surface in this case can be determined from

$$\dot{Q} = (T_{\infty 1} - T_{s1}) / R_{\infty 1-s1} \longrightarrow T_{s1} = T_{\infty 1} - (\dot{Q}R_{\infty 1-s1})$$

$$\dot{Q} = (T_{s2} - T_{\infty 2}) / R_{s2-\infty 2} \longrightarrow T_{s2} = T_{\infty 2} + (\dot{Q}R_{s2-\infty 2})$$

where $R_{\infty-i}$ is the thermal resistance between the environment ∞ and surface i .

10-12C Yes, it is.

10-13C The window glass which consists of two 4 mm thick glass sheets pressed tightly against each other will probably have thermal contact resistance which serves as an additional thermal resistance to heat transfer through window, and thus the heat transfer rate will be smaller relative to the one which consists of a single 8 mm thick glass sheet.

10-14C Convection heat transfer through the wall is expressed as $\dot{Q} = hA_s(T_s - T_\infty)$. In steady heat transfer, heat transfer rate to the wall and from the wall are equal. Therefore at the outer surface which has convection heat transfer coefficient three times that of the inner surface will experience three times smaller temperature drop compared to the inner surface. Therefore, at the outer surface, the temperature will be closer to the surrounding air temperature.

10-15C The new design introduces the thermal resistance of the copper layer in addition to the thermal resistance of the aluminum which has the same value for both designs. Therefore, the new design will be a poorer conductor of heat.

10-16C The blanket will introduce additional resistance to heat transfer and slow down the heat gain of the drink wrapped in a blanket. Therefore, the drink left on a table will warm up faster.

10-17 The two surfaces of a wall are maintained at specified temperatures. The rate of heat loss through the wall is to be determined.

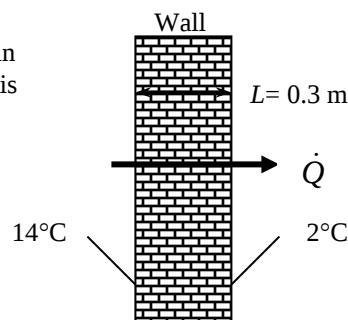
Assumptions 1 Heat transfer through the wall is steady since the surface temperatures remain constant at the specified values. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. 3 Thermal conductivity is constant.

Properties The thermal conductivity is given to be $k = 0.8 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The surface area of the wall and the rate of heat loss through the wall are

$$A = (3 \text{ m}) \times (6 \text{ m}) = 18 \text{ m}^2$$

$$\dot{Q} = kA \frac{T_1 - T_2}{L} = (0.8 \text{ W/m}\cdot^\circ\text{C})(18 \text{ m}^2) \frac{(14 - 2)^\circ\text{C}}{0.3 \text{ m}} = \mathbf{576 \text{ W}}$$



10-18 A double-pane window is considered. The rate of heat loss through the window and the temperature difference across the largest thermal resistance are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer coefficients are constant.

Properties The thermal conductivities of glass and air are given to be $0.78 \text{ W/m}\cdot\text{K}$ and $0.025 \text{ W/m}\cdot\text{K}$, respectively.

Analysis (a) The rate of heat transfer through the window is determined to be

$$\begin{aligned}\dot{Q} &= \frac{A\Delta T}{\frac{1}{h_i} + \frac{L_g}{k_g} + \frac{L_a}{k_a} + \frac{L_g}{k_g} + \frac{1}{h_o}} \\ &= \frac{(1 \times 1.5 \text{ m}^2)[20 - (-20)]^\circ\text{C}}{\frac{1}{40 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{0.004 \text{ m}}{0.78 \text{ W/m} \cdot ^\circ\text{C}} + \frac{0.005 \text{ m}}{0.025 \text{ W/m} \cdot ^\circ\text{C}} + \frac{0.004 \text{ m}}{0.78 \text{ W/m} \cdot ^\circ\text{C}} + \frac{1}{20 \text{ W/m}^2 \cdot ^\circ\text{C}}} \\ &= \frac{(1 \times 1.5 \text{ m}^2)[20 - (-20)]^\circ\text{C}}{0.025 + 0.000513 + 0.2 + 0.000513 + 0.05} = \mathbf{210 \text{ W}}\end{aligned}$$

(b) Noting that the largest resistance is through the air gap, the temperature difference across the air gap is determined from

$$\Delta T_a = \dot{Q}R_a = \dot{Q} \frac{L_a}{k_a A} = (210 \text{ W}) \frac{0.005 \text{ m}}{(0.025 \text{ W/m} \cdot ^\circ\text{C})(1 \times 1.5 \text{ m}^2)} = \mathbf{28^\circ\text{C}}$$

10-19 The two surfaces of a window are maintained at specified temperatures. The rate of heat loss through the window and the inner surface temperature are to be determined.

Assumptions 1 Heat transfer through the window is steady since the surface temperatures remain constant at the specified values. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. 3 Thermal conductivity is constant. 4 Heat transfer by radiation is negligible.

Properties The thermal conductivity of the glass is given to be $k = 0.78 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The area of the window and the individual resistances are

$$A = (1.2 \text{ m}) \times (2 \text{ m}) = 2.4 \text{ m}^2$$

$$R_i = R_{\text{conv},1} = \frac{1}{h_1 A} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(2.4 \text{ m}^2)} = 0.04167^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L}{k_1 A} = \frac{0.006 \text{ m}}{(0.78 \text{ W/m}\cdot^\circ\text{C})(2.4 \text{ m}^2)} = 0.00321^\circ\text{C/W}$$

$$R_o = R_{\text{conv},2} = \frac{1}{h_2 A} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(2.4 \text{ m}^2)} = 0.01667^\circ\text{C/W}$$

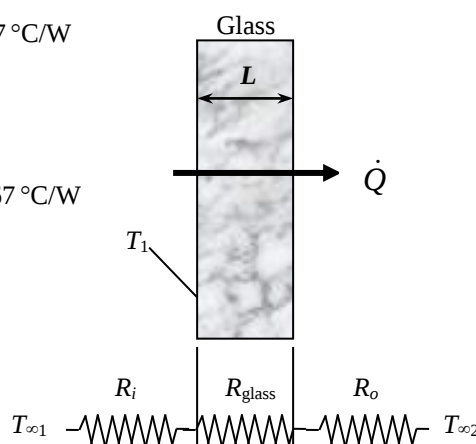
$$\begin{aligned}R_{\text{total}} &= R_{\text{conv},1} + R_{\text{glass}} + R_{\text{conv},2} \\ &= 0.04167 + 0.00321 + 0.01667 = 0.06155^\circ\text{C/W}\end{aligned}$$

The steady rate of heat transfer through window glass is then

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[24 - (-5)]^\circ\text{C}}{0.06155^\circ\text{C/W}} = \mathbf{471 \text{ W}}$$

The inner surface temperature of the window glass can be determined from

$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{\text{conv},1}} \longrightarrow T_1 = T_{\infty 1} - \dot{Q}R_{\text{conv},1} = 24^\circ\text{C} - (471 \text{ W})(0.04167^\circ\text{C/W}) = \mathbf{4.4^\circ\text{C}}$$



10-20 A double-pane window consists of two layers of glass separated by a stagnant air space. For specified indoors and outdoors temperatures, the rate of heat loss through the window and the inner surface temperature of the window are to be determined.

Assumptions 1 Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. 3 Thermal conductivities of the glass and air are constant. 4 Heat transfer by radiation is negligible.

Properties The thermal conductivity of the glass and air are given to be $k_{\text{glass}} = 0.78 \text{ W/m}\cdot^\circ\text{C}$ and $k_{\text{air}} = 0.026 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The area of the window and the individual resistances are

$$A = (1.2 \text{ m}) \times (2 \text{ m}) = 2.4 \text{ m}^2$$

$$R_i = R_{\text{conv},1} = \frac{1}{h_1 A} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(2.4 \text{ m}^2)} = 0.0417 ^\circ\text{C/W}$$

$$R_1 = R_3 = R_{\text{glass}} = \frac{L_1}{k_1 A} = \frac{0.003 \text{ m}}{(0.78 \text{ W/m}\cdot^\circ\text{C})(2.4 \text{ m}^2)} = 0.0016 ^\circ\text{C/W}$$

$$R_2 = R_{\text{air}} = \frac{L_2}{k_2 A} = \frac{0.012 \text{ m}}{(0.026 \text{ W/m}\cdot^\circ\text{C})(2.4 \text{ m}^2)} = 0.1923 ^\circ\text{C/W}$$

$$R_o = R_{\text{conv},2} = \frac{1}{h_2 A} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(2.4 \text{ m}^2)} = 0.0167 ^\circ\text{C/W}$$

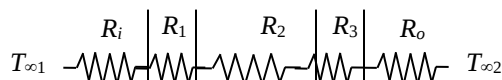
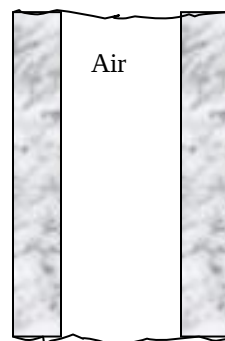
$$R_{\text{total}} = R_{\text{conv},1} + 2R_1 + R_2 + R_{\text{conv},2} = 0.0417 + 2(0.0016) + 0.1923 + 0.0167 \\ = 0.2539 ^\circ\text{C/W}$$

The steady rate of heat transfer through window glass then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[24 - (-5)]^\circ\text{C}}{0.2539 ^\circ\text{C/W}} = \mathbf{114 \text{ W}}$$

The inner surface temperature of the window glass can be determined from

$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{\text{conv},1}} \longrightarrow T_1 = T_{\infty 1} - \dot{Q}R_{\text{conv},1} = 24^\circ\text{C} - (114 \text{ W})(0.0417 ^\circ\text{C/W}) = \mathbf{19.2^\circ\text{C}}$$



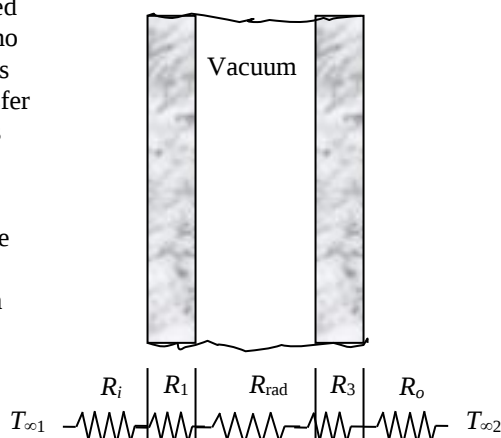
10-21 A double-pane window consists of two layers of glass separated by an evacuated space. For specified indoors and outdoors temperatures, the rate of heat loss through the window and the inner surface temperature of the window are to be determined.

Assumptions 1 Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. 3 Thermal conductivity of the glass is constant. 4 Heat transfer by radiation is negligible.

Properties The thermal conductivity of the glass is given to be $k_{\text{glass}} = 0.78 \text{ W/m}\cdot^\circ\text{C}$.

Analysis Heat cannot be conducted through an evacuated space since the thermal conductivity of vacuum is zero (no medium to conduct heat) and thus its thermal resistance is zero. Therefore, if radiation is disregarded, the heat transfer through the window will be zero. Then the answer of this problem is **zero** since the problem states to disregard radiation.

Discussion In reality, heat will be transferred between the glasses by radiation. We do not know the inner surface temperatures of windows. In order to determine radiation heat resistance we assume them to be 5°C and 15°C , respectively, and take the emissivity to be 1. Then individual resistances are



$$A = (1.2 \text{ m}) \times (2 \text{ m}) = 2.4 \text{ m}^2$$

$$R_i = R_{\text{conv},1} = \frac{1}{h_1 A} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(2.4 \text{ m}^2)} = 0.0417^\circ\text{C/W}$$

$$R_1 = R_3 = R_{\text{glass}} = \frac{L_1}{k_1 A} = \frac{0.003 \text{ m}}{(0.78 \text{ W/m}\cdot^\circ\text{C})(2.4 \text{ m}^2)} = 0.0016^\circ\text{C/W}$$

$$\begin{aligned} R_{\text{rad}} &= \frac{1}{\varepsilon \sigma A (T_s^2 + T_{\text{surr}}^2)(T_s + T_{\text{surr}})} \\ &= \frac{1}{1(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(2.4 \text{ m}^2)[288^2 + 278^2][288 + 278] \text{ K}^3} \\ &= 0.0810^\circ\text{C/W} \end{aligned}$$

$$R_o = R_{\text{conv},2} = \frac{1}{h_2 A} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(2.4 \text{ m}^2)} = 0.0167^\circ\text{C/W}$$

$$\begin{aligned} R_{\text{total}} &= R_{\text{conv},1} + 2R_1 + R_{\text{rad}} + R_{\text{conv},2} = 0.0417 + 2(0.0016) + 0.0810 + 0.0167 \\ &= 0.1426^\circ\text{C/W} \end{aligned}$$

The steady rate of heat transfer through window glass then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[24 - (-5)]^\circ\text{C}}{0.1426^\circ\text{C/W}} = \mathbf{203 \text{ W}}$$

The inner surface temperature of the window glass can be determined from

$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{\text{conv},1}} \longrightarrow T_1 = T_{\infty 1} - \dot{Q} R_{\text{conv},1} = 24^\circ\text{C} - (203 \text{ W})(0.0417^\circ\text{C/W}) = \mathbf{15.5^\circ\text{C}}$$

Similarly, the inner surface temperatures of the glasses are calculated to be 15.2°C and -1.2°C (we had assumed them to be 15°C and 5°C when determining the radiation resistance). We can improve the result obtained by reevaluating the radiation resistance and repeating the calculations.

10-22 EES Prob. 10-20 is reconsidered. The rate of heat transfer through the window as a function of the width of air space is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

A=1.2*2 [m^2]
 L_glass=3 [mm]
 k_glass=0.78 [W/m-C]
 L_air=12 [mm]
 T_infinity_1=24 [C]
 T_infinity_2=-5 [C]
 h_1=10 [W/m^2-C]
 h_2=25 [W/m^2-C]

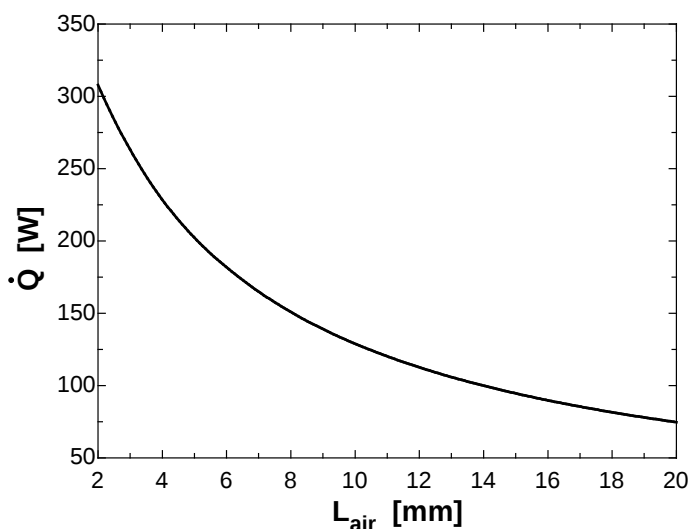
"PROPERTIES"

k_air=conductivity(Air,T=25)

"ANALYSIS"

R_conv_1=1/(h_1*A)
 R_glass=(L_glass*Convert(mm, m))/(k_glass*A)
 R_air=(L_air*Convert(mm, m))/(k_air*A)
 R_conv_2=1/(h_2*A)
 R_total=R_conv_1+2*R_glass+R_air+R_conv_2
 Q_dot=(T_infinity_1-T_infinity_2)/R_total

L _{air} [mm]	Q [W]
2	307.8
4	228.6
6	181.8
8	150.9
10	129
12	112.6
14	99.93
16	89.82
18	81.57
20	74.7



10-23E The inner and outer surfaces of the walls of an electrically heated house remain at specified temperatures during a winter day. The amount of heat lost from the house that day and its cost are to be determined.

Assumptions 1 Heat transfer through the walls is steady since the surface temperatures of the walls remain constant at the specified values during the time period considered. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. 3 Thermal conductivity of the walls is constant.

Properties The thermal conductivity of the brick wall is given to be $k = 0.40 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F}$.

Analysis We consider heat loss through the walls only. The total heat transfer area is

$$A = 2(50 \times 9 + 35 \times 9) = 1530 \text{ ft}^2$$

The rate of heat loss during the daytime is

$$\dot{Q}_{\text{day}} = kA \frac{T_1 - T_2}{L} = (0.40 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F})(1530 \text{ ft}^2) \frac{(55 - 45)^\circ\text{F}}{1 \text{ ft}} = 6120 \text{ Btu/h}$$

The rate of heat loss during nighttime is

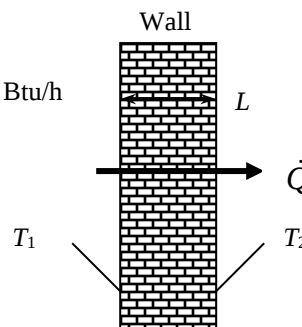
$$\begin{aligned} \dot{Q}_{\text{night}} &= kA \frac{T_1 - T_2}{L} \\ &= (0.40 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F})(1530 \text{ ft}^2) \frac{(55 - 35)^\circ\text{C}}{1 \text{ ft}} = 12,240 \text{ Btu/h} \end{aligned}$$

The amount of heat loss from the house that night will be

$$\begin{aligned} \dot{Q} = \frac{Q}{\Delta t} \longrightarrow Q &= \dot{Q} \Delta t = 10 \dot{Q}_{\text{day}} + 14 \dot{Q}_{\text{night}} = (10 \text{ h})(6120 \text{ Btu/h}) + (14 \text{ h})(12,240 \text{ Btu/h}) \\ &= \mathbf{232,560 \text{ Btu}} \end{aligned}$$

Then the cost of this heat loss for that day becomes

$$\text{Cost} = (232,560 / 3412 \text{ kWh})(\$0.09 / \text{kWh}) = \mathbf{\$6.13}$$



10-24 A cylindrical resistor on a circuit board dissipates 0.15 W of power steadily in a specified environment. The amount of heat dissipated in 24 h, the surface heat flux, and the surface temperature of the resistor are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat is transferred uniformly from all surfaces of the resistor.

Analysis (a) The amount of heat this resistor dissipates during a 24-hour period is

$$Q = \dot{Q} \Delta t = (0.15 \text{ W})(24 \text{ h}) = \mathbf{3.6 \text{ Wh}}$$

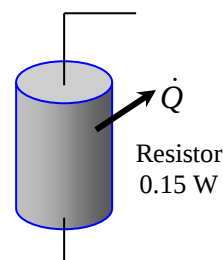
(b) The heat flux on the surface of the resistor is

$$A_s = 2 \frac{\pi D^2}{4} + \pi DL = 2 \frac{\pi (0.003 \text{ m})^2}{4} + \pi (0.003 \text{ m})(0.012 \text{ m}) = 0.000127 \text{ m}^2$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{0.15 \text{ W}}{0.000127 \text{ m}^2} = \mathbf{1179 \text{ W/m}^2}$$

(c) The surface temperature of the resistor can be determined from

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 40^\circ\text{C} + \frac{0.15 \text{ W}}{(9 \text{ W/m}^2 \cdot ^\circ\text{C})(0.000127 \text{ m}^2)} = \mathbf{171^\circ\text{C}}$$



10-25 A power transistor dissipates 0.2 W of power steadily in a specified environment. The amount of heat dissipated in 24 h, the surface heat flux, and the surface temperature of the resistor are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat is transferred uniformly from all surfaces of the transistor.

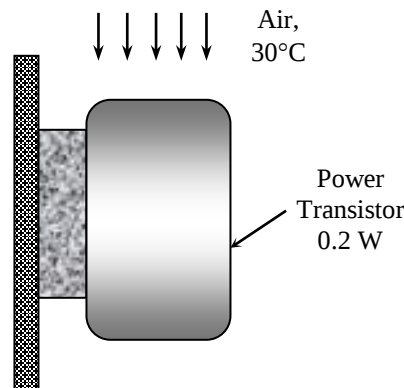
Analysis (a) The amount of heat this transistor dissipates during a 24-hour period is

$$Q = \dot{Q}\Delta t = (0.2 \text{ W})(24 \text{ h}) = 4.8 \text{ Wh} = \mathbf{0.0048 \text{ kWh}}$$

(b) The heat flux on the surface of the transistor is

$$\begin{aligned} A_s &= 2 \frac{\pi D^2}{4} + \pi DL \\ &= 2 \frac{\pi (0.005 \text{ m})^2}{4} + \pi (0.005 \text{ m})(0.004 \text{ m}) = 0.0001021 \text{ m}^2 \end{aligned}$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{0.2 \text{ W}}{0.0001021 \text{ m}^2} = \mathbf{1959 \text{ W/m}^2}$$



(c) The surface temperature of the transistor can be determined from

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 30^\circ\text{C} + \frac{0.2 \text{ W}}{(18 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0001021 \text{ m}^2)} = \mathbf{139^\circ\text{C}}$$

10-26 A circuit board houses 100 chips, each dissipating 0.06 W. The surface heat flux, the surface temperature of the chips, and the thermal resistance between the surface of the board and the cooling medium are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer from the back surface of the board is negligible. 2 Heat is transferred uniformly from the entire front surface.

Analysis (a) The heat flux on the surface of the circuit board is

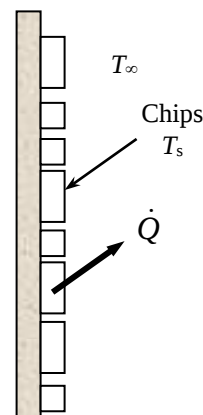
$$\begin{aligned} A_s &= (0.12 \text{ m})(0.18 \text{ m}) = 0.0216 \text{ m}^2 \\ \dot{q} &= \frac{\dot{Q}}{A_s} = \frac{(100 \times 0.06) \text{ W}}{0.0216 \text{ m}^2} = \mathbf{278 \text{ W/m}^2} \end{aligned}$$

(b) The surface temperature of the chips is

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) \\ T_s &= T_\infty + \frac{\dot{Q}}{hA_s} = 40^\circ\text{C} + \frac{(100 \times 0.06) \text{ W}}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0216 \text{ m}^2)} = \mathbf{67.8^\circ\text{C}} \end{aligned}$$

(c) The thermal resistance is

$$R_{\text{conv}} = \frac{1}{hA_s} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0216 \text{ m}^2)} = \mathbf{4.63^\circ\text{C/W}}$$



10-27 A person is dissipating heat at a rate of 150 W by natural convection and radiation to the surrounding air and surfaces. For a given deep body temperature, the outer skin temperature is to be determined.

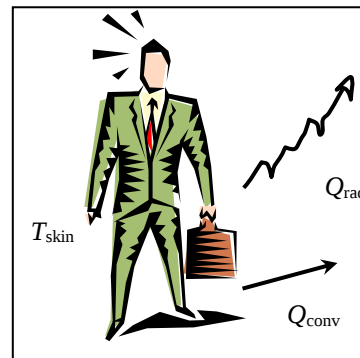
Assumptions 1 Steady operating conditions exist. 2 The heat transfer coefficient is constant and uniform over the entire exposed surface of the person. 3 The surrounding surfaces are at the same temperature as the indoor air temperature. 4 Heat generation within the 0.5-cm thick outer layer of the tissue is negligible.

Properties The thermal conductivity of the tissue near the skin is given to be $k = 0.3 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The skin temperature can be determined directly from

$$\dot{Q} = kA \frac{T_1 - T_{skin}}{L}$$

$$T_{skin} = T_1 - \frac{\dot{Q}L}{kA} = 37^\circ\text{C} - \frac{(150 \text{ W})(0.005 \text{ m})}{(0.3 \text{ W/m} \cdot ^\circ\text{C})(1.7 \text{ m}^2)} = \mathbf{35.5^\circ\text{C}}$$



10-28 Heat is transferred steadily to the boiling water in an aluminum pan. The inner surface temperature of the bottom of the pan is given. The boiling heat transfer coefficient and the outer surface temperature of the bottom of the pan are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is one-dimensional since the thickness of the bottom of the pan is small relative to its diameter. 3 The thermal conductivity of the pan is constant.

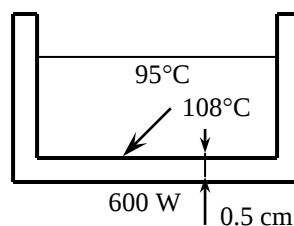
Properties The thermal conductivity of the aluminum pan is given to be $k = 237 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis (a) The boiling heat transfer coefficient is

$$A_s = \frac{\pi D^2}{4} = \frac{\pi (0.25 \text{ m})^2}{4} = 0.0491 \text{ m}^2$$

$$\dot{Q} = hA_s (T_s - T_\infty)$$

$$h = \frac{\dot{Q}}{A_s (T_s - T_\infty)} = \frac{800 \text{ W}}{(0.0491 \text{ m}^2)(108 - 95)^\circ\text{C}} = \mathbf{1254 \text{ W/m}^2 \cdot ^\circ\text{C}}$$



(b) The outer surface temperature of the bottom of the pan is

$$\dot{Q} = kA \frac{T_{s,outer} - T_{s,inner}}{L}$$

$$T_{s,outer} = T_{s,inner} + \frac{\dot{Q}L}{kA} = 108^\circ\text{C} + \frac{(800 \text{ W})(0.005 \text{ m})}{(237 \text{ W/m} \cdot ^\circ\text{C})(0.0491 \text{ m}^2)} = \mathbf{108.3^\circ\text{C}}$$

10-29E A wall is constructed of two layers of sheetrock with fiberglass insulation in between. The thermal resistance of the wall and its R-value of insulation are to be determined.

Assumptions 1 Heat transfer through the wall is one-dimensional. 2 Thermal conductivities are constant.

Properties The thermal conductivities are given to be

$$k_{\text{sheetrock}} = 0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F} \text{ and } k_{\text{insulation}} = 0.020 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}.$$

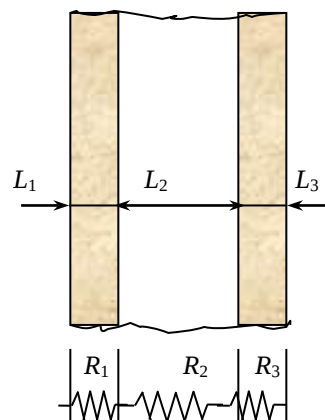
Analysis (a) The surface area of the wall is not given and thus we consider a unit surface area ($A = 1 \text{ ft}^2$). Then the R-value of insulation of the wall becomes equivalent to its thermal resistance, which is determined from.

$$R_{\text{sheetrock}} = R_1 = R_3 = \frac{L_1}{k_1} = \frac{0.7/12 \text{ ft}}{(0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 0.583 \text{ ft}^2\cdot^\circ\text{F}\cdot\text{h/Btu}$$

$$R_{\text{fiberglass}} = R_2 = \frac{L_2}{k_2} = \frac{7/12 \text{ ft}}{(0.020 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 29.17 \text{ ft}^2\cdot^\circ\text{F}\cdot\text{h/Btu}$$

$$R_{\text{total}} = 2R_1 + R_2 = 2 \times 0.583 + 29.17 = \mathbf{30.34 \text{ ft}^2\cdot^\circ\text{F}\cdot\text{h/Btu}}$$

(b) Therefore, this is approximately a **R-30** wall in English units.



10-30 The roof of a house with a gas furnace consists of a concrete that is losing heat to the outdoors by radiation and convection. The rate of heat transfer through the roof and the money lost through the roof that night during a 14 hour period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The emissivity and thermal conductivity of the roof are constant.

Properties The thermal conductivity of the concrete is given to be $k = 2 \text{ W/m} \cdot ^\circ\text{C}$. The emissivity of both surfaces of the roof is given to be 0.9.

Analysis When the surrounding surface temperature is different than the ambient temperature, the thermal resistances network approach becomes cumbersome in problems that involve radiation. Therefore, we will use a different but intuitive approach.

In steady operation, heat transfer from the room to the roof (by convection and radiation) must be equal to the heat transfer from the roof to the surroundings (by convection and radiation), that must be equal to the heat transfer through the roof by conduction. That is,

$$\dot{Q} = \dot{Q}_{\text{room to roof, conv+rad}} = \dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{roof to surroundings, conv+rad}}$$

Taking the inner and outer surface temperatures of the roof to be $T_{s,in}$ and $T_{s,out}$, respectively, the quantities above can be expressed as

$$\begin{aligned} \dot{Q}_{\text{room to roof, conv+rad}} &= h_i A (T_{\text{room}} - T_{s,in}) + \varepsilon A \sigma (T_{\text{room}}^4 - T_{s,in}^4) = (5 \text{ W/m}^2 \cdot ^\circ\text{C})(300 \text{ m}^2)(20 - T_{s,in})^\circ\text{C} \\ &\quad + (0.9)(300 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) \left[(20 + 273 \text{ K})^4 - (T_{s,in} + 273 \text{ K})^4 \right] \end{aligned}$$

$$\dot{Q}_{\text{roof, cond}} = kA \frac{T_{s,in} - T_{s,out}}{L} = (2 \text{ W/m} \cdot ^\circ\text{C})(300 \text{ m}^2) \frac{T_{s,in} - T_{s,out}}{0.15 \text{ m}}$$

$$\begin{aligned} \dot{Q}_{\text{roof to surr, conv+rad}} &= h_o A (T_{s,out} - T_{\text{surr}}) + \varepsilon A \sigma (T_{s,out}^4 - T_{\text{surr}}^4) = (12 \text{ W/m}^2 \cdot ^\circ\text{C})(300 \text{ m}^2)(T_{s,out} - 10)^\circ\text{C} \\ &\quad + (0.9)(300 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) \left[(T_{s,out} + 273 \text{ K})^4 - (100 \text{ K})^4 \right] \end{aligned}$$

Solving the equations above simultaneously gives

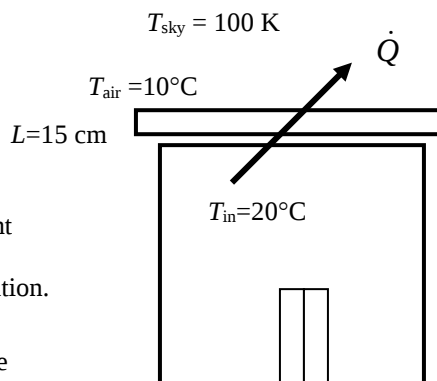
$$\dot{Q} = 37,440 \text{ W}, T_{s,in} = 7.3^\circ\text{C}, \text{ and } T_{s,out} = -2.1^\circ\text{C}$$

The total amount of natural gas consumption during a 14-hour period is

$$Q_{\text{gas}} = \frac{Q_{\text{total}}}{0.80} = \frac{\dot{Q} \Delta t}{0.80} = \frac{(37.440 \text{ kJ/s})(14 \times 3600 \text{ s})}{0.80} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 22.36 \text{ therms}$$

Finally, the money lost through the roof during that period is

$$\text{Money lost} = (22.36 \text{ therms})(\$1.20 / \text{therm}) = \mathbf{\$26.8}$$



10-31 An exposed hot surface of an industrial natural gas furnace is to be insulated to reduce the heat loss through that section of the wall by 90 percent. The thickness of the insulation that needs to be used is to be determined. Also, the length of time it will take for the insulation to pay for itself from the energy it saves will be determined.

Assumptions 1 Heat transfer through the wall is steady and one-dimensional. 2 Thermal conductivities are constant. 3 The furnace operates continuously. 4 The given heat transfer coefficient accounts for the radiation effects.

Properties The thermal conductivity of the glass wool insulation is given to be $k = 0.038 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The rate of heat transfer without insulation is

$$A = (2 \text{ m})(1.5 \text{ m}) = 3 \text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) = (10 \text{ W/m}^2 \cdot ^\circ\text{C})(3 \text{ m}^2)(80 - 30)^\circ\text{C} = 1500 \text{ W}$$

In order to reduce heat loss by 90%, the new heat transfer rate and thermal resistance must be

$$\dot{Q} = 0.10 \times 1500 \text{ W} = 150 \text{ W}$$

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} \rightarrow R_{\text{total}} = \frac{\Delta T}{\dot{Q}} = \frac{(80 - 30)^\circ\text{C}}{150 \text{ W}} = 0.333^\circ\text{C/W}$$

and in order to have this thermal resistance, the thickness of insulation must be

$$\begin{aligned} R_{\text{total}} &= R_{\text{conv}} + R_{\text{insulation}} = \frac{1}{hA} + \frac{L}{kA} \\ &= \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(3 \text{ m}^2)} + \frac{L}{(0.038 \text{ W/m} \cdot ^\circ\text{C})(3 \text{ m}^2)} = 0.333^\circ\text{C/W} \\ L &= 0.034 \text{ m} = \mathbf{3.4 \text{ cm}} \end{aligned}$$

Noting that heat is saved at a rate of $0.9 \times 1500 = 1350 \text{ W}$ and the furnace operates continuously and thus $365 \times 24 = 8760 \text{ h}$ per year, and that the furnace efficiency is 78%, the amount of natural gas saved per year is

$$\text{Energy Saved} = \frac{\dot{Q}_{\text{saved}} \Delta t}{\text{Efficiency}} = \frac{(1.350 \text{ kJ/s})(8760 \text{ h})}{0.78} \left(\frac{3600 \text{ s}}{1 \text{ h}} \right) \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 517.4 \text{ therms}$$

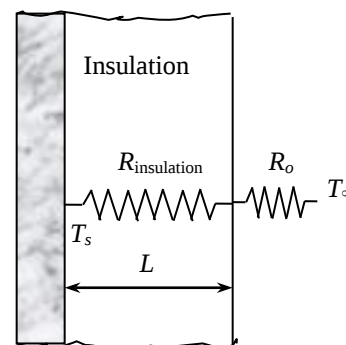
The money saved is

$$\text{Money saved} = (\text{Energy Saved})(\text{Cost of energy}) = (517.4 \text{ therms})(\$1.10/\text{therm}) = \$569.1 (\text{per year})$$

The insulation will pay for its cost of \$250 in

$$\text{Payback period} = \frac{\text{Money spent}}{\text{Money saved}} = \frac{\$250}{\$569.1/\text{yr}} = \mathbf{0.44 \text{ yr}}$$

which is equal to 5.3 months.



10-32 An exposed hot surface of an industrial natural gas furnace is to be insulated to reduce the heat loss through that section of the wall by 90 percent. The thickness of the insulation that needs to be used is to be determined. Also, the length of time it will take for the insulation to pay for itself from the energy it saves will be determined.

Assumptions 1 Heat transfer through the wall is steady and one-dimensional. 2 Thermal conductivities are constant. 3 The furnace operates continuously. 4 The given heat transfer coefficients accounts for the radiation effects.

Properties The thermal conductivity of the expanded perlite insulation is given to be $k = 0.052 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The rate of heat transfer without insulation is

$$A = (2 \text{ m})(1.5 \text{ m}) = 3 \text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) = (10 \text{ W/m}^2 \cdot ^\circ\text{C})(3 \text{ m}^2)(80 - 30)^\circ\text{C} = 1500 \text{ W}$$

In order to reduce heat loss by 90%, the new heat transfer rate and thermal resistance must be

$$\dot{Q} = 0.10 \times 1500 \text{ W} = 150 \text{ W}$$

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} \rightarrow R_{\text{total}} = \frac{\Delta T}{\dot{Q}} = \frac{(80 - 30)^\circ\text{C}}{150 \text{ W}} = 0.333^\circ\text{C/W}$$

and in order to have this thermal resistance, the thickness of insulation must be

$$\begin{aligned} R_{\text{total}} &= R_{\text{conv}} + R_{\text{insulation}} = \frac{1}{hA} + \frac{L}{kA} \\ &= \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(3 \text{ m}^2)} + \frac{L}{(0.052 \text{ W/m}\cdot^\circ\text{C})(3 \text{ m}^2)} = 0.333^\circ\text{C/W} \\ L &= 0.047 \text{ m} = \mathbf{4.7 \text{ cm}} \end{aligned}$$

Noting that heat is saved at a rate of $0.9 \times 1500 = 1350 \text{ W}$ and the furnace operates continuously and thus $365 \times 24 = 8760 \text{ h}$ per year, and that the furnace efficiency is 78%, the amount of natural gas saved per year is

$$\text{Energy Saved} = \frac{\dot{Q}_{\text{saved}} \Delta t}{\text{Efficiency}} = \frac{(1.350 \text{ kJ/s})(8760 \text{ h})}{0.78} \left(\frac{3600 \text{ s}}{1 \text{ h}} \right) \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 517.4 \text{ therms}$$

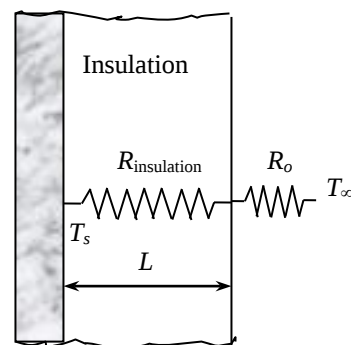
The money saved is

$$\text{Money saved} = (\text{Energy Saved})(\text{Cost of energy}) = (517.4 \text{ therms})(\$1.10/\text{therm}) = \$569.1 \text{ (per year)}$$

The insulation will pay for its cost of \$250 in

$$\text{Payback period} = \frac{\text{Money spent}}{\text{Money saved}} = \frac{\$250}{\$569.1/\text{yr}} = \mathbf{0.44 \text{ yr}}$$

which is equal to 5.3 months.



10-33 EES Prob. 10-31 is reconsidered. The effect of thermal conductivity on the required insulation thickness is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$A = 2 \times 1.5 \text{ [m}^2\text{]}$$

$$T_s = 80 \text{ [C]}$$

$$T_{\infty} = 30 \text{ [C]}$$

$$h = 10 \text{ [W/m}^2\text{-C]}$$

$$k_{\text{ins}} = 0.038 \text{ [W/m-C]}$$

$$f_{\text{reduce}} = 0.90$$

"ANALYSIS"

$$Q_{\text{dot_old}} = h \cdot A \cdot (T_s - T_{\infty})$$

$$Q_{\text{dot_new}} = (1 - f_{\text{reduce}}) \cdot Q_{\text{dot_old}}$$

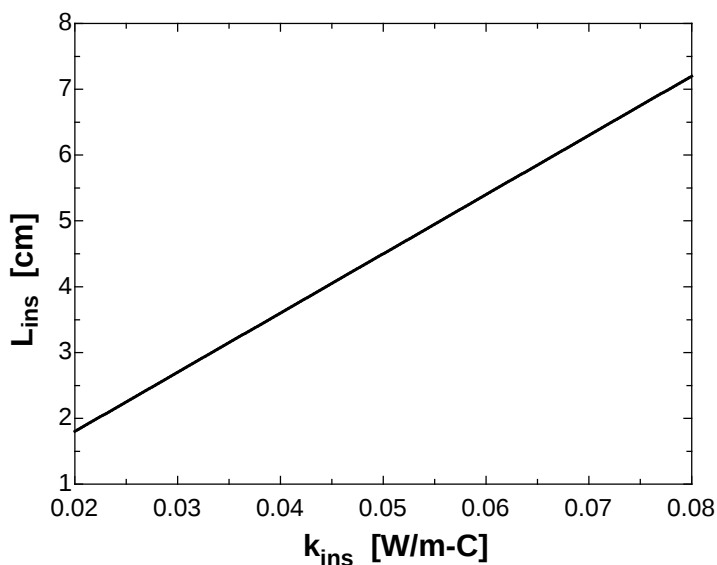
$$Q_{\text{dot_new}} = (T_s - T_{\infty}) / R_{\text{total}}$$

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{ins}}$$

$$R_{\text{conv}} = 1 / (h \cdot A)$$

$$R_{\text{ins}} = (L_{\text{ins}} \cdot \text{Convert}(\text{cm}, \text{m})) / (k_{\text{ins}} \cdot A) \text{ "L}_{\text{ins}} \text{ is in cm"}$$

k_{ins} [W/m.C]	L_{ins} [cm]
0.02	1.8
0.025	2.25
0.03	2.7
0.035	3.15
0.04	3.6
0.045	4.05
0.05	4.5
0.055	4.95
0.06	5.4
0.065	5.85
0.07	6.3
0.075	6.75
0.08	7.2



10-34E Two of the walls of a house have no windows while the other two walls have 4 windows each. The ratio of heat transfer through the walls with and without windows is to be determined.

Assumptions 1 Heat transfer through the walls and the windows is steady and one-dimensional. 2 Thermal conductivities are constant. 3 Any direct radiation gain or loss through the windows is negligible. 4 Heat transfer coefficients are constant and uniform over the entire surface.

Properties The thermal conductivity of the glass is given to be $k_{\text{glass}} = 0.45 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F}$. The R-value of the wall is given to be $19 \text{ h} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}$.

Analysis The thermal resistances through the wall without windows are

$$A = (12 \text{ ft})(40 \text{ ft}) = 480 \text{ ft}^2$$

$$R_i = \frac{1}{h_i A} = \frac{1}{(2 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(480 \text{ ft}^2)} = 0.0010417 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{\text{wall}} = \frac{L}{kA} = \frac{19 \text{ h} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}}{480 \text{ ft}^2} = 0.03958 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(4 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(480 \text{ ft}^2)} = 0.00052 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{\text{total},1} = R_i + R_{\text{wall}} + R_o = 0.0010417 + 0.03958 + 0.00052 = 0.0411417 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

The thermal resistances through the wall with windows are

$$A_{\text{windows}} = 4(3 \times 5) = 60 \text{ ft}^2$$

$$A_{\text{wall}} = A_{\text{total}} - A_{\text{windows}} = 480 - 60 = 420 \text{ ft}^2$$

$$R_2 = R_{\text{glass}} = \frac{L}{kA} = \frac{0.25 / 12 \text{ ft}}{(0.45 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F})(60 \text{ ft}^2)} = 0.0007716 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

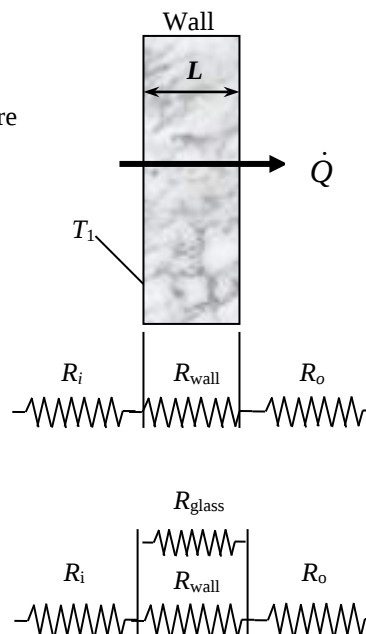
$$R_4 = R_{\text{wall}} = \frac{L}{kA} = \frac{19 \text{ h} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}}{(420 \text{ ft}^2)} = 0.04524 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_{\text{glass}}} + \frac{1}{R_{\text{wall}}} = \frac{1}{0.0007716} + \frac{1}{0.04524} \longrightarrow R_{\text{eqv}} = 0.00076 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{\text{total},2} = R_i + R_{\text{eqv}} + R_o = 0.001047 + 0.00076 + 0.00052 = 0.002327 \text{ h} \cdot ^\circ\text{F}/\text{Btu}$$

Then the ratio of the heat transfer through the walls with and without windows becomes

$$\frac{\dot{Q}_{\text{total},2}}{\dot{Q}_{\text{total},1}} = \frac{\Delta T / R_{\text{total},2}}{\Delta T / R_{\text{total},1}} = \frac{R_{\text{total},1}}{R_{\text{total},2}} = \frac{0.0411417}{0.002327} = \mathbf{17.7}$$



10-35 Two of the walls of a house have no windows while the other two walls have single- or double-pane windows. The average rate of heat transfer through each wall, and the amount of money this household will save per heating season by converting the single pane windows to double pane windows are to be determined.

Assumptions 1 Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. 3 Thermal conductivities of the glass and air are constant. 4 Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be $k = 0.026 \text{ W/m} \cdot ^\circ\text{C}$ for air, and $0.78 \text{ W/m} \cdot ^\circ\text{C}$ for glass.

Analysis The rate of heat transfer through each wall can be determined by applying thermal resistance network. The convection resistances at the inner and outer surfaces are common in all cases.

Walls without windows:

$$R_i = \frac{1}{h_i A} = \frac{1}{(7 \text{ W/m}^2 \cdot ^\circ\text{C})(10 \times 4 \text{ m}^2)} = 0.003571 ^\circ\text{C/W}$$

$$R_{\text{wall}} = \frac{L_{\text{wall}}}{kA} = \frac{R\text{-value}}{A} = \frac{2.31 \text{ m}^2 \cdot ^\circ\text{C/W}}{(10 \times 4 \text{ m}^2)} = 0.05775 ^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(18 \text{ W/m}^2 \cdot ^\circ\text{C})(10 \times 4 \text{ m}^2)} = 0.001389 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{wall}} + R_o = 0.003571 + 0.05775 + 0.001389 = 0.06271 ^\circ\text{C/W}$$

$$\text{Then } \dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(24 - 8) ^\circ\text{C}}{0.06271 ^\circ\text{C/W}} = \mathbf{255.1 \text{ W}}$$

Wall with single pane windows:

$$R_i = \frac{1}{h_i A} = \frac{1}{(7 \text{ W/m}^2 \cdot ^\circ\text{C})(20 \times 4 \text{ m}^2)} = 0.001786 ^\circ\text{C/W}$$

$$R_{\text{wall}} = \frac{L_{\text{wall}}}{kA} = \frac{R\text{-value}}{A} = \frac{2.31 \text{ m}^2 \cdot ^\circ\text{C/W}}{(20 \times 4) - 5(1.2 \times 1.8) \text{ m}^2} = 0.033382 ^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{kA} = \frac{0.005 \text{ m}}{(0.78 \text{ W/m}^2 \cdot ^\circ\text{C})(1.2 \times 1.8) \text{ m}^2} = 0.002968 ^\circ\text{C/W}$$

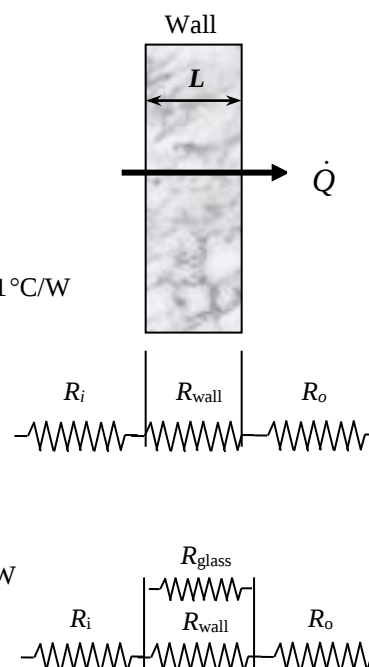
$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_{\text{wall}}} + 5 \frac{1}{R_{\text{glass}}} = \frac{1}{0.033382} + 5 \frac{1}{0.002968} \rightarrow R_{\text{eqv}} = 0.000583 ^\circ\text{C/W}$$

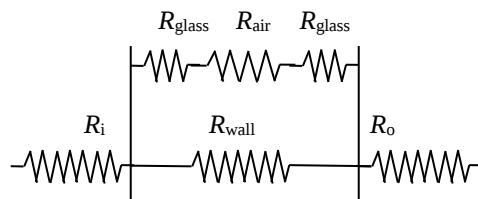
$$R_o = \frac{1}{h_o A} = \frac{1}{(18 \text{ W/m}^2 \cdot ^\circ\text{C})(20 \times 4 \text{ m}^2)} = 0.000694 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{eqv}} + R_o = 0.001786 + 0.000583 + 0.000694 = 0.003063 ^\circ\text{C/W}$$

Then

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(24 - 8) ^\circ\text{C}}{0.003063 ^\circ\text{C/W}} = \mathbf{5224 \text{ W}}$$



4th wall with double pane windows:

$$R_{\text{wall}} = \frac{L_{\text{wall}}}{kA} = \frac{R\text{-value}}{A} = \frac{2.31 \text{ m}^2 \cdot ^\circ\text{C}/\text{W}}{(20 \times 4) - 5(1.2 \times 1.8) \text{ m}^2} = 0.033382 ^\circ\text{C}/\text{W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{kA} = \frac{0.005 \text{ m}}{(0.78 \text{ W}/\text{m}^2 \cdot ^\circ\text{C})(1.2 \times 1.8) \text{ m}^2} = 0.002968 ^\circ\text{C}/\text{W}$$

$$R_{\text{air}} = \frac{L_{\text{air}}}{kA} = \frac{0.015 \text{ m}}{(0.026 \text{ W}/\text{m}^2 \cdot ^\circ\text{C})(1.2 \times 1.8) \text{ m}^2} = 0.267094 ^\circ\text{C}/\text{W}$$

$$R_{\text{window}} = 2R_{\text{glass}} + R_{\text{air}} = 2 \times 0.002968 + 0.267094 = 0.27303 ^\circ\text{C}/\text{W}$$

$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_{\text{wall}}} + 5 \frac{1}{R_{\text{window}}} = \frac{1}{0.033382} + 5 \frac{1}{0.27303} \longrightarrow R_{\text{eqv}} = 0.020717 ^\circ\text{C}/\text{W}$$

$$R_{\text{total}} = R_i + R_{\text{eqv}} + R_o = 0.001786 + 0.020717 + 0.000694 = 0.023197 ^\circ\text{C}/\text{W}$$

Then $\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(24 - 8) ^\circ\text{C}}{0.023197 ^\circ\text{C}/\text{W}} = \mathbf{690 \text{ W}}$

The rate of heat transfer which will be saved if the single pane windows are converted to double pane windows is

$$\dot{Q}_{\text{save}} = \dot{Q}_{\text{single pane}} - \dot{Q}_{\text{double pane}} = 5224 - 690 = 4534 \text{ W}$$

The amount of energy and money saved during a 7-month long heating season by switching from single pane to double pane windows become

$$Q_{\text{save}} = \dot{Q}_{\text{save}} \Delta t = (4.534 \text{ kW})(7 \times 30 \times 24 \text{ h}) = 22,851 \text{ kWh}$$

$$\text{Money savings} = (\text{Energy saved})(\text{Unit cost of energy}) = (22,851 \text{ kWh})(\$0.08/\text{kWh}) = \mathbf{\$1828}$$

10-36 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between two layers of sheet metal. The minimum thickness of insulation that needs to be used in the wall in order to avoid condensation on the outer surfaces is to be determined.

Assumptions 1 Heat transfer through the refrigerator walls is steady since the temperatures of the food compartment and the kitchen air remain constant at the specified values. 2 Heat transfer is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer coefficients account for the radiation effects.

Properties The thermal conductivities are given to be $k = 15.1 \text{ W/m}\cdot^\circ\text{C}$ for sheet metal and $0.035 \text{ W/m}\cdot^\circ\text{C}$ for fiberglass insulation.

Analysis The minimum thickness of insulation can be determined by assuming the outer surface temperature of the refrigerator to be 20°C . In steady operation, the rate of heat transfer through the refrigerator wall is constant, and thus heat transfer between the room and the refrigerated space is equal to the heat transfer between the room and the outer surface of the refrigerator. Considering a unit surface area,

$$\begin{aligned}\dot{Q} &= h_o A(T_{\text{room}} - T_{s,\text{out}}) \\ &= (9 \text{ W/m}^2 \cdot ^\circ\text{C})(1 \text{ m}^2)(25 - 20)^\circ\text{C} = 45 \text{ W}\end{aligned}$$

Using the thermal resistance network, heat transfer between the room and the refrigerated space can be expressed as

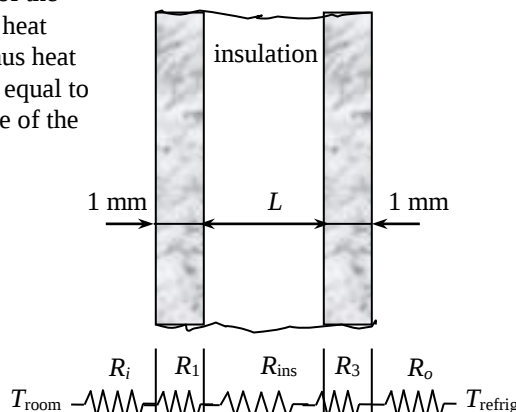
$$\begin{aligned}\dot{Q} &= \frac{T_{\text{room}} - T_{\text{refrig}}}{R_{\text{total}}} \\ \dot{Q} / A &= \frac{T_{\text{room}} - T_{\text{refrig}}}{\frac{1}{h_o} + 2\left(\frac{L}{k}\right)_{\text{metal}} + \left(\frac{L}{k}\right)_{\text{insulation}} + \frac{1}{h_i}}\end{aligned}$$

Substituting,

$$45 \text{ W/m}^2 = \frac{(25 - 3)^\circ\text{C}}{\frac{1}{9 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{2 \times 0.001 \text{ m}}{15.1 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{L}{0.035 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{1}{4 \text{ W/m}^2 \cdot ^\circ\text{C}}}$$

Solving for L , the minimum thickness of insulation is determined to be

$$L = 0.0045 \text{ m} = \mathbf{0.45 \text{ cm}}$$



10-37 EES Prob. 10-36 is reconsidered. The effects of the thermal conductivities of the insulation material and the sheet metal on the thickness of the insulation is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

k_ins=0.035 [W/m-C]
 L_metal=0.001 [m]
 k_metal=15.1 [W/m-C]
 T_refrig=3 [C]
 T_kitchen=25 [C]
 h_i=4 [W/m^2-C]
 h_o=9 [W/m^2-C]
 T_s_out=20 [C]

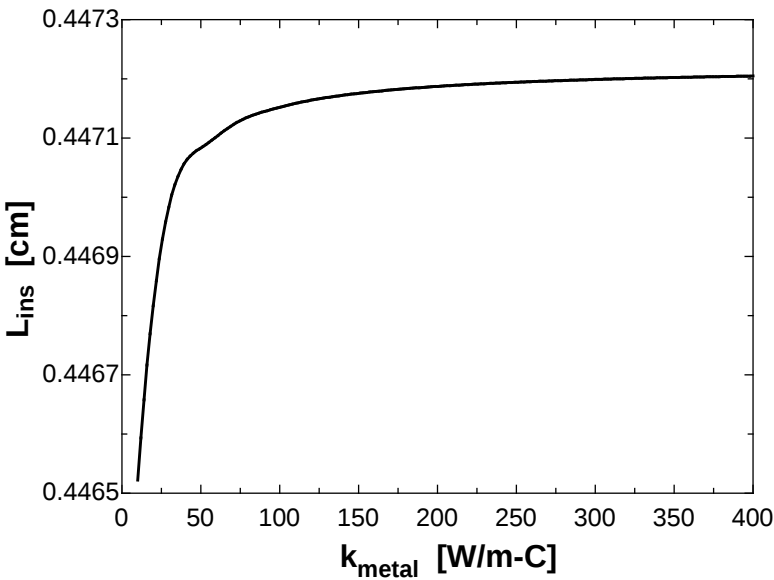
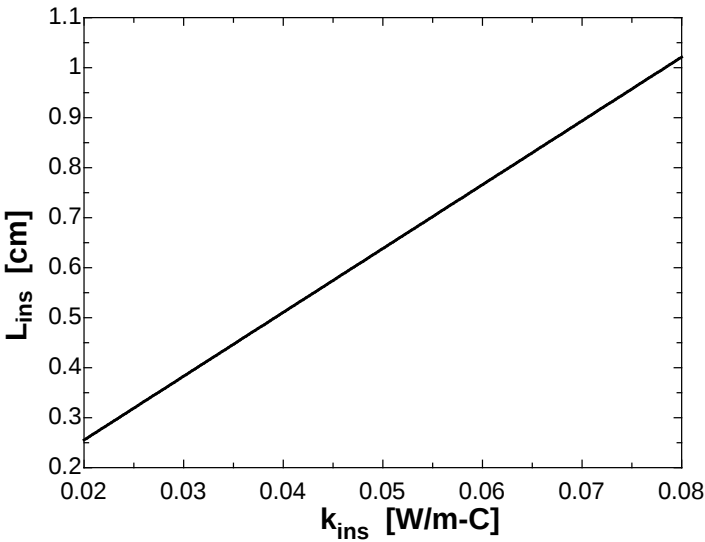
"ANALYSIS"

A=1 [m^2] "a unit surface area is considered"
 $Q_{\text{dot}} = h_o A (T_{\text{kitchen}} - T_{\text{s_out}})$
 $Q_{\text{dot}} = (T_{\text{kitchen}} - T_{\text{refrig}}) / R_{\text{total}}$
 $R_{\text{total}} = R_{\text{conv_i}} + 2R_{\text{metal}} + R_{\text{ins}} + R_{\text{conv_o}}$
 $R_{\text{conv_i}} = 1 / (h_i A)$
 $R_{\text{metal}} = L_{\text{metal}} / (k_{\text{metal}} A)$
 $R_{\text{ins}} = (L_{\text{ins}} \text{Convert}(\text{cm}, \text{m})) / (k_{\text{ins}} A)$ "L_ins is in cm"
 $R_{\text{conv_o}} = 1 / (h_o A)$

k_{ins} [W/m.C]	L_{ins} [cm]
0.02	0.2553
0.025	0.3191
0.03	0.3829
0.035	0.4468
0.04	0.5106
0.045	0.5744
0.05	0.6382
0.055	0.702
0.06	0.7659
0.065	0.8297
0.07	0.8935
0.075	0.9573
0.08	1.021

k_{metal} [W/m.C]	L_{ins} [cm]
10	0.4465
30.53	0.447
51.05	0.4471
71.58	0.4471
92.11	0.4471
112.6	0.4472
133.2	0.4472
153.7	0.4472
174.2	0.4472
194.7	0.4472
215.3	0.4472
235.8	0.4472

256.3	0.4472
276.8	0.4472
297.4	0.4472
317.9	0.4472
338.4	0.4472
358.9	0.4472
379.5	0.4472
400	0.4472



10-38 Heat is to be conducted along a circuit board with a copper layer on one side. The percentages of heat conduction along the copper and epoxy layers as well as the effective thermal conductivity of the board are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is one-dimensional since heat transfer from the side surfaces is disregarded. 3 Thermal conductivities are constant.

Properties The thermal conductivities are given to be $k = 386 \text{ W/m} \cdot ^\circ\text{C}$ for copper and $0.26 \text{ W/m} \cdot ^\circ\text{C}$ for epoxy layers.

Analysis We take the length in the direction of heat transfer to be L and the width of the board to be w . Then heat conduction along this two-layer board can be expressed as

$$\begin{aligned}\dot{Q} &= \dot{Q}_{\text{copper}} + \dot{Q}_{\text{epoxy}} = \left(kA \frac{\Delta T}{L} \right)_{\text{copper}} + \left(kA \frac{\Delta T}{L} \right)_{\text{epoxy}} \\ &= [(kt)_{\text{copper}} + (kt)_{\text{epoxy}}] w \frac{\Delta T}{L}\end{aligned}$$

Heat conduction along an “equivalent” board of thickness $t = t_{\text{copper}} + t_{\text{epoxy}}$ and thermal conductivity k_{eff} can be expressed as

$$\dot{Q} = \left(kA \frac{\Delta T}{L} \right)_{\text{board}} = k_{\text{eff}} (t_{\text{copper}} + t_{\text{epoxy}}) w \frac{\Delta T}{L}$$

Setting the two relations above equal to each other and solving for the effective conductivity gives

$$k_{\text{eff}} (t_{\text{copper}} + t_{\text{epoxy}}) = (kt)_{\text{copper}} + (kt)_{\text{epoxy}} \longrightarrow k_{\text{eff}} = \frac{(kt)_{\text{copper}} + (kt)_{\text{epoxy}}}{t_{\text{copper}} + t_{\text{epoxy}}}$$

Note that heat conduction is proportional to kt . Substituting, the fractions of heat conducted along the copper and epoxy layers as well as the effective thermal conductivity of the board are determined to be

$$(kt)_{\text{copper}} = (386 \text{ W/m} \cdot ^\circ\text{C})(0.0001 \text{ m}) = 0.0386 \text{ W/}^\circ\text{C}$$

$$(kt)_{\text{epoxy}} = (0.26 \text{ W/m} \cdot ^\circ\text{C})(0.0012 \text{ m}) = 0.000312 \text{ W/}^\circ\text{C}$$

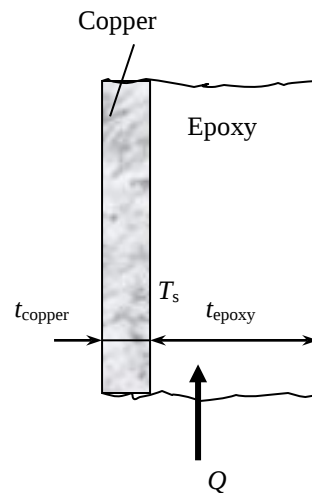
$$(kt)_{\text{total}} = (kt)_{\text{copper}} + (kt)_{\text{epoxy}} = 0.0386 + 0.000312 = 0.038912 \text{ W/}^\circ\text{C}$$

$$f_{\text{epoxy}} = \frac{(kt)_{\text{epoxy}}}{(kt)_{\text{total}}} = \frac{0.000312}{0.038912} = 0.008 = \mathbf{0.8\%}$$

$$f_{\text{copper}} = \frac{(kt)_{\text{copper}}}{(kt)_{\text{total}}} = \frac{0.0386}{0.038912} = 0.992 = \mathbf{99.2\%}$$

and

$$k_{\text{eff}} = \frac{(386 \times 0.0001 + 0.26 \times 0.0012) \text{ W/}^\circ\text{C}}{(0.0001 + 0.0012) \text{ m}} = \mathbf{29.9 \text{ W/m} \cdot ^\circ\text{C}}$$



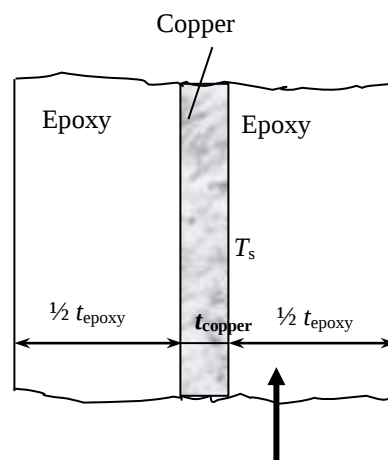
10-39E A thin copper plate is sandwiched between two layers of epoxy boards. The effective thermal conductivity of the board along its 9 in long side and the fraction of the heat conducted through copper along that side are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is one-dimensional since heat transfer from the side surfaces are disregarded 3 Thermal conductivities are constant.

Properties The thermal conductivities are given to be $k = 223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for copper and $0.15 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for epoxy layers.

Analysis We take the length in the direction of heat transfer to be L and the width of the board to be w . Then heat conduction along this two-layer plate can be expressed as (we treat the two layers of epoxy as a single layer that is twice as thick)

$$\begin{aligned}\dot{Q} &= \dot{Q}_{\text{copper}} + \dot{Q}_{\text{epoxy}} \\ &= \left(kA \frac{\Delta T}{L} \right)_{\text{copper}} + \left(kA \frac{\Delta T}{L} \right)_{\text{epoxy}} = \left[(kt)_{\text{copper}} + (kt)_{\text{epoxy}} \right] w \frac{\Delta T}{L}\end{aligned}$$



Heat conduction along an “equivalent” plate of thick ness $t = t_{\text{copper}} + t_{\text{epoxy}}$ and thermal conductivity k_{eff} can be expressed as

$$\dot{Q} = \left(kA \frac{\Delta T}{L} \right)_{\text{board}} = k_{\text{eff}} (t_{\text{copper}} + t_{\text{epoxy}}) w \frac{\Delta T}{L}$$

Setting the two relations above equal to each other and solving for the effective conductivity gives

$$k_{\text{eff}} (t_{\text{copper}} + t_{\text{epoxy}}) = (kt)_{\text{copper}} + (kt)_{\text{epoxy}} \longrightarrow k_{\text{eff}} = \frac{(kt)_{\text{copper}} + (kt)_{\text{epoxy}}}{t_{\text{copper}} + t_{\text{epoxy}}}$$

Note that heat conduction is proportional to kt . Substituting, the fraction of heat conducted along the copper layer and the effective thermal conductivity of the plate are determined to be

$$(kt)_{\text{copper}} = (223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.03/12 \text{ ft}) = 0.5575 \text{ Btu/h}\cdot^\circ\text{F}$$

$$(kt)_{\text{epoxy}} = 2(0.15 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.15/12 \text{ ft}) = 0.00375 \text{ Btu/h}\cdot^\circ\text{F}$$

$$(kt)_{\text{total}} = (kt)_{\text{copper}} + (kt)_{\text{epoxy}} = (0.5575 + 0.00375) = 0.56125 \text{ Btu/h}\cdot^\circ\text{F}$$

and

$$\begin{aligned}k_{\text{eff}} &= \frac{(kt)_{\text{copper}} + (kt)_{\text{epoxy}}}{t_{\text{copper}} + t_{\text{epoxy}}} \\ &= \frac{0.56125 \text{ Btu/h}\cdot^\circ\text{F}}{[(0.03/12) + 2(0.15/12)] \text{ ft}} = \mathbf{20.4 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}\end{aligned}$$

$$f_{\text{copper}} = \frac{(kt)_{\text{copper}}}{(kt)_{\text{total}}} = \frac{0.5575}{0.56125} = 0.993 = \mathbf{99.3\%}$$

Thermal Contact Resistance

10-40C The resistance that an interface offers to heat transfer per unit interface area is called thermal contact resistance, R_c . The inverse of thermal contact resistance is called the thermal contact conductance.

10-41C The thermal contact resistance will be greater for rough surfaces because an interface with rough surfaces will contain more air gaps whose thermal conductivity is low.

10-42C An interface acts like a very thin layer of insulation, and thus the thermal contact resistance has significance only for highly conducting materials like metals. Therefore, the thermal contact resistance can be ignored for two layers of insulation pressed against each other.

10-43C An interface acts like a very thin layer of insulation, and thus the thermal contact resistance is significant for highly conducting materials like metals. Therefore, the thermal contact resistance must be considered for two layers of metals pressed against each other.

10-44C Heat transfer through the voids at an interface is by conduction and radiation. Evacuating the interface eliminates heat transfer by conduction, and thus increases the thermal contact resistance.

10-45C Thermal contact resistance can be minimized by (1) applying a thermally conducting liquid on the surfaces before they are pressed against each other, (2) by replacing the air at the interface by a better conducting gas such as helium or hydrogen, (3) by increasing the interface pressure, and (4) by inserting a soft metallic foil such as tin, silver, copper, nickel, or aluminum between the two surfaces.

10-46 The thickness of copper plate whose thermal resistance is equal to the thermal contact resistance is to be determined.

Properties The thermal conductivity of copper is $k = 386 \text{ W/m}\cdot^\circ\text{C}$.

Analysis Noting that thermal contact resistance is the inverse of thermal contact conductance, the thermal contact resistance is determined to be

$$R_c = \frac{1}{h_c} = \frac{1}{18,000 \text{ W/m}^2\cdot^\circ\text{C}} = 5.556 \times 10^{-5} \text{ m}^2\cdot^\circ\text{C/W}$$

For a unit surface area, the thermal resistance of a flat plate is defined as $R = \frac{L}{k}$ where L is the thickness of the plate and k is the thermal conductivity. Setting $R = R_c$, the equivalent thickness is determined from the relation above to be

$$L = kR = kR_c = (386 \text{ W/m}\cdot^\circ\text{C})(5.556 \times 10^{-5} \text{ m}^2\cdot^\circ\text{C/W}) = 0.0214 \text{ m} = \mathbf{2.14 \text{ cm}}$$

Therefore, the interface between the two plates offers as much resistance to heat transfer as a 2.14 cm thick copper. Note that the thermal contact resistance in this case is greater than the sum of the thermal resistances of both plates.

10-47 Six identical power transistors are attached on a copper plate. For a maximum case temperature of 75°C, the maximum power dissipation and the temperature jump at the interface are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer can be approximated as being one-dimensional, although it is recognized that heat conduction in some parts of the plate will be two-dimensional since the plate area is much larger than the base area of the transistor. But the large thermal conductivity of copper will minimize this effect. 3 All the heat generated at the junction is dissipated through the back surface of the plate since the transistors are covered by a thick plexiglass layer. 4 Thermal conductivities are constant.

Properties The thermal conductivity of copper is given to be $k = 386 \text{ W/m}\cdot^\circ\text{C}$. The contact conductance at the interface of copper-aluminum plates for the case of 1.10-1.4 μm roughness and 10 MPa pressure is $h_c = 49,000 \text{ W/m}^2\cdot^\circ\text{C}$ (Table 10-2).

Analysis The contact area between the case and the plate is given to be 9 cm^2 , and the plate area for each transistor is 100 cm^2 . The thermal resistance network of this problem consists of three resistances in series (contact, plate, and convection) which are determined to be

$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(49,000 \text{ W/m}^2\cdot^\circ\text{C})(9 \times 10^{-4} \text{ m}^2)} = 0.0227^\circ\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.012 \text{ m}}{(386 \text{ W/m}\cdot^\circ\text{C})(0.01 \text{ m}^2)} = 0.0031^\circ\text{C/W}$$

$$R_{\text{convection}} = \frac{1}{h_o A} = \frac{1}{(30 \text{ W/m}^2\cdot^\circ\text{C})(0.01 \text{ m}^2)} = 3.333^\circ\text{C/W}$$

The total thermal resistance is then

$$\begin{aligned} R_{\text{total}} &= R_{\text{contact}} + R_{\text{plate}} + R_{\text{convection}} \\ &= 0.0227 + 0.0031 + 3.333 = 3.359^\circ\text{C/W} \end{aligned}$$

Note that the thermal resistance of copper plate is very small and can be ignored all together. Then the rate of heat transfer is determined to be

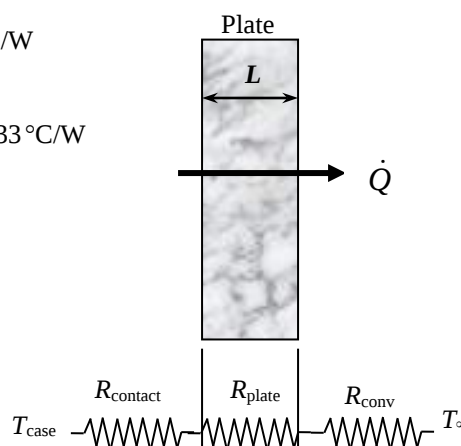
$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{(75 - 23)^\circ\text{C}}{3.359^\circ\text{C/W}} = 15.5 \text{ W}$$

Therefore, the power transistor should not be operated at power levels greater than 15.5 W if the case temperature is not to exceed 75°C.

The temperature jump at the interface is determined from

$$\Delta T_{\text{interface}} = \dot{Q} R_{\text{contact}} = (15.5 \text{ W})(0.0227^\circ\text{C/W}) = 0.35^\circ\text{C}$$

which is not very large. Therefore, even if we eliminate the thermal contact resistance at the interface completely, we will lower the operating temperature of the transistor in this case by less than 1°C.

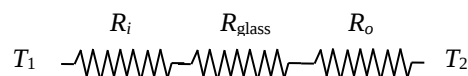
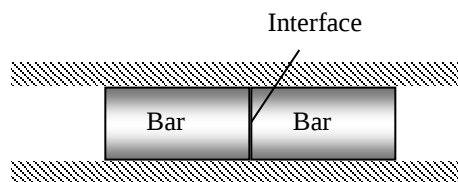


10-48 Two cylindrical aluminum bars with ground surfaces are pressed against each other in an insulation sleeve. For specified top and bottom surface temperatures, the rate of heat transfer along the cylinders and the temperature drop at the interface are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is one-dimensional in the axial direction since the lateral surfaces of both cylinders are well-insulated. 3 Thermal conductivities are constant.

Properties The thermal conductivity of aluminum bars is given to be $k = 176 \text{ W/m} \cdot ^\circ\text{C}$. The contact conductance at the interface of aluminum-aluminum plates for the case of ground surfaces and of 20 atm $\approx$ 2 MPa pressure is $h_c = 11,400 \text{ W/m}^2 \cdot ^\circ\text{C}$ (Table 10-2).

Analysis (a) The thermal resistance network in this case consists of two conduction resistance and the contact resistance, and they are determined to be



$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(11,400 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.05 \text{ m})^2/4]} = 0.0447 \text{ } ^\circ\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.15 \text{ m}}{(176 \text{ W/m} \cdot ^\circ\text{C})[\pi(0.05 \text{ m})^2/4]} = 0.4341 \text{ } ^\circ\text{C/W}$$

Then the rate of heat transfer is determined to be

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{\Delta T}{R_{\text{contact}} + 2R_{\text{bar}}} = \frac{(150 - 20)^\circ\text{C}}{(0.0447 + 2 \times 0.4341) ^\circ\text{C/W}} = \mathbf{142.4 \text{ W}}$$

Therefore, the rate of heat transfer through the bars is 142.4 W.

(b) The temperature drop at the interface is determined to be

$$\Delta T_{\text{interface}} = \dot{Q} R_{\text{contact}} = (142.4 \text{ W})(0.0447 ^\circ\text{C/W}) = \mathbf{6.4^\circ\text{C}}$$

10-49 A thin copper plate is sandwiched between two epoxy boards. The error involved in the total thermal resistance of the plate if the thermal contact conductances are ignored is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is one-dimensional since the plate is large. 3 Thermal conductivities are constant.

Properties The thermal conductivities are given to be $k = 386 \text{ W/m}\cdot^\circ\text{C}$ for copper plates and $k = 0.26 \text{ W/m}\cdot^\circ\text{C}$ for epoxy boards. The contact conductance at the interface of copper-epoxy layers is given to be $h_c = 6000 \text{ W/m}^2\cdot^\circ\text{C}$.

Analysis The thermal resistances of different layers for unit surface area of 1 m^2 are

$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(6000 \text{ W/m}^2\cdot^\circ\text{C})(1 \text{ m}^2)} = 0.00017^\circ\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.001 \text{ m}}{(386 \text{ W/m}\cdot^\circ\text{C})(1 \text{ m}^2)} = 2.6 \times 10^{-6}^\circ\text{C/W}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.005 \text{ m}}{(0.26 \text{ W/m}\cdot^\circ\text{C})(1 \text{ m}^2)} = 0.01923^\circ\text{C/W}$$

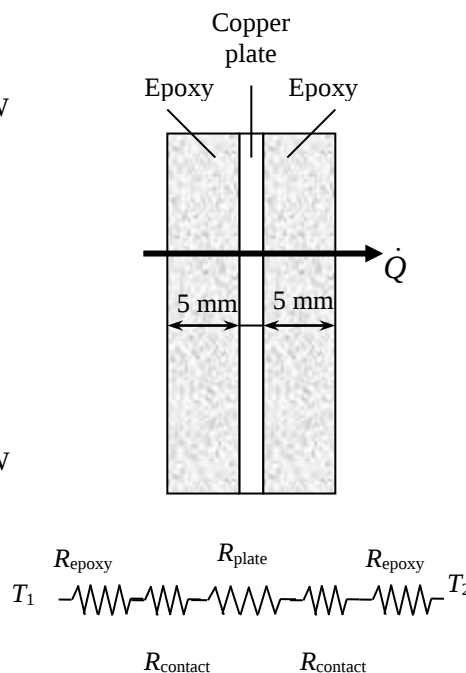
The total thermal resistance is

$$\begin{aligned} R_{\text{total}} &= 2R_{\text{contact}} + R_{\text{plate}} + 2R_{\text{epoxy}} \\ &= 2 \times 0.00017 + 2.6 \times 10^{-6} + 2 \times 0.01923 = 0.03880^\circ\text{C/W} \end{aligned}$$

Then the percent error involved in the total thermal resistance of the plate if the thermal contact resistances are ignored is determined to be

$$\% \text{Error} = \frac{2R_{\text{contact}}}{R_{\text{total}}} \times 100 = \frac{2 \times 0.00017}{0.03880} \times 100 = \mathbf{0.88\%}$$

which is negligible.



Generalized Thermal Resistance Networks

10-50C Parallel resistances indicate simultaneous heat transfer (such as convection and radiation on a surface). Series resistances indicate sequential heat transfer (such as two homogeneous layers of a wall).

10-51C The thermal resistance network approach will give adequate results for multi-dimensional heat transfer problems if heat transfer occurs predominantly in one direction.

10-52C Two approaches used in development of the thermal resistance network in the x-direction for multi-dimensional problems are (1) to assume any plane wall normal to the x-axis to be isothermal and (2) to assume any plane parallel to the x-axis to be adiabatic.

10-53 A typical section of a building wall is considered. The average heat flux through the wall is to be determined.

Assumptions 1 Steady operating conditions exist.

Properties The thermal conductivities are given to be $k_{23b} = 50$ W/m·K, $k_{23a} = 0.03$ W/m·K, $k_{12} = 0.5$ W/m·K, $k_{34} = 1.0$ W/m·K.

Analysis We consider 1 m² of wall area. The thermal resistances are

$$R_{12} = \frac{t_{12}}{k_{12}} = \frac{0.01 \text{ m}}{(0.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.02 \text{ m}^2 \cdot ^\circ\text{C/W}$$

$$R_{23a} = t_{23} \frac{L_a}{k_{23a}(L_a + L_b)} = (0.08 \text{ m}) \frac{0.6 \text{ m}}{(0.03 \text{ W/m} \cdot ^\circ\text{C})(0.6 + 0.005)} = 2.645 \text{ m}^2 \cdot ^\circ\text{C/W}$$

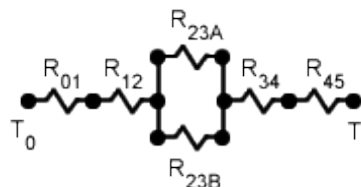
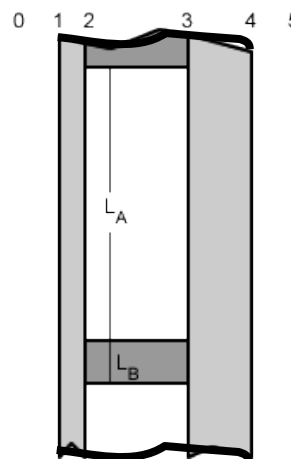
$$R_{23b} = t_{23} \frac{L_b}{k_{23b}(L_a + L_b)} = (0.08 \text{ m}) \frac{0.005 \text{ m}}{(50 \text{ W/m} \cdot ^\circ\text{C})(0.6 + 0.005)} = 1.32 \times 10^{-5} \text{ m}^2 \cdot ^\circ\text{C/W}$$

$$R_{34} = \frac{t_{34}}{k_{34}} = \frac{0.1 \text{ m}}{(1.0 \text{ W/m} \cdot ^\circ\text{C})} = 0.1 \text{ m}^2 \cdot ^\circ\text{C/W}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_{12} + \left(\frac{R_{23a} R_{23b}}{R_{23a} + R_{23b}} \right) + R_{34} = 0.02 + 2.645 \left(\frac{1.32 \times 10^{-5}}{2.645 + 1.32 \times 10^{-5}} \right) + 0.1 = 0.120 \text{ m}^2 \cdot ^\circ\text{C/W}$$

$$\dot{q} = \frac{T_4 - T_1}{R_{\text{total}}} = \frac{(35 - 20)^\circ\text{C}}{0.120 \text{ m}^2 \cdot ^\circ\text{C/W}} = \mathbf{125 \text{ W/m}^2}$$

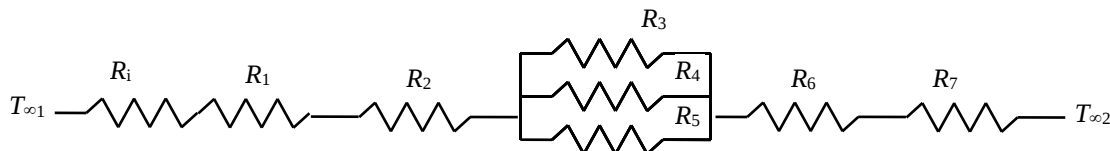


10-54 A wall consists of horizontal bricks separated by plaster layers. There are also plaster layers on each side of the wall, and a rigid foam on the inner side of the wall. The rate of heat transfer through the wall is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the wall is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be $k = 0.72 \text{ W/m}\cdot^\circ\text{C}$ for bricks, $k = 0.22 \text{ W/m}\cdot^\circ\text{C}$ for plaster layers, and $k = 0.026 \text{ W/m}\cdot^\circ\text{C}$ for the rigid foam.

Analysis We consider 1 m deep and 0.33 m high portion of wall which is representative of the entire wall. The thermal resistance network and individual resistances are



$$R_i = R_{conv,1} = \frac{1}{h_1 A} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.33 \times 1 \text{ m}^2)} = 0.303 ^\circ\text{C/W}$$

$$R_1 = R_{foam} = \frac{L}{kA} = \frac{0.02 \text{ m}}{(0.026 \text{ W/m}\cdot^\circ\text{C})(0.33 \times 1 \text{ m}^2)} = 2.33 ^\circ\text{C/W}$$

$$R_2 = R_6 = R_{plaster\ side} = \frac{L}{kA} = \frac{0.02 \text{ m}}{(0.22 \text{ W/m}\cdot^\circ\text{C})(0.33 \times 1 \text{ m}^2)} = 0.275 ^\circ\text{C/W}$$

$$R_3 = R_5 = R_{plaster\ center} = \frac{L}{h_o A} = \frac{0.18 \text{ m}}{(0.22 \text{ W/m}\cdot^\circ\text{C})(0.015 \times 1 \text{ m}^2)} = 54.55 ^\circ\text{C/W}$$

$$R_4 = R_{brick} = \frac{L}{kA} = \frac{0.18 \text{ m}}{(0.72 \text{ W/m}\cdot^\circ\text{C})(0.30 \times 1 \text{ m}^2)} = 0.833 ^\circ\text{C/W}$$

$$R_o = R_{conv,2} = \frac{1}{h_2 A} = \frac{1}{(20 \text{ W/m}\cdot^\circ\text{C})(0.33 \times 1 \text{ m}^2)} = 0.152 ^\circ\text{C/W}$$

$$\frac{1}{R_{mid}} = \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5} = \frac{1}{54.55} + \frac{1}{0.833} + \frac{1}{54.55} \longrightarrow R_{mid} = 0.81 ^\circ\text{C/W}$$

$$R_{total} = R_i + R_1 + 2R_2 + R_{mid} + R_o = 0.303 + 2.33 + 2(0.275) + 0.81 + 0.152 = 4.145 ^\circ\text{C/W}$$

The steady rate of heat transfer through the wall per 0.33 m^2 is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[(22 - (-4))]^\circ\text{C}}{4.145 ^\circ\text{C/W}} = 6.27 \text{ W}$$

Then steady rate of heat transfer through the entire wall becomes

$$\dot{Q}_{total} = (6.27 \text{ W}) \frac{(4 \times 6) \text{ m}^2}{0.33 \text{ m}^2} = 456 \text{ W}$$

10-55 EES Prob. 10-54 is reconsidered. The rate of heat transfer through the wall as a function of the thickness of the rigid foam is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

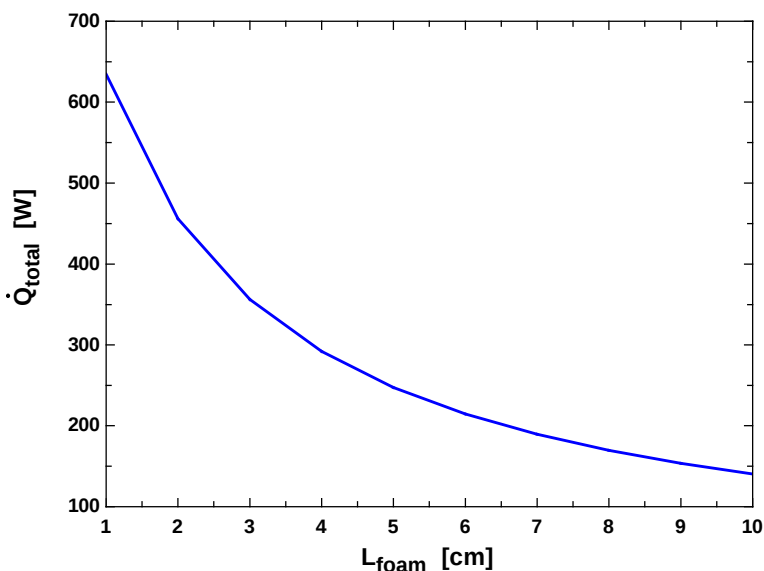
"GIVEN"

$A = 4 \times 6 \text{ [m}^2\text{]}$
 $L_{\text{brick}} = 0.18 \text{ [m]}$
 $L_{\text{plaster_center}} = 0.18 \text{ [m]}$
 $L_{\text{plaster_side}} = 0.02 \text{ [m]}$
 $L_{\text{foam}} = 2 \text{ [cm]}$
 $k_{\text{brick}} = 0.72 \text{ [W/m}\cdot\text{C]}$
 $k_{\text{plaster}} = 0.22 \text{ [W/m}\cdot\text{C]}$
 $k_{\text{foam}} = 0.026 \text{ [W/m}\cdot\text{C]}$
 $T_{\text{infinity_1}} = 22 \text{ [C]}$
 $T_{\text{infinity_2}} = -4 \text{ [C]}$
 $h_1 = 10 \text{ [W/m}^2\cdot\text{C]}$
 $h_2 = 20 \text{ [W/m}^2\cdot\text{C]}$

"ANALYSIS"

$R_{\text{conv_1}} = 1/(h_1 \cdot A_1)$
 $A_1 = 0.33 \times 1 \text{ [m}^2\text{]}$
 $R_{\text{foam}} = (L_{\text{foam}} \cdot \text{Convert}(\text{cm}, \text{m})) / (k_{\text{foam}} \cdot A_1)$ "L_foam is in cm"
 $R_{\text{plaster_side}} = L_{\text{plaster_side}} / (k_{\text{plaster}} \cdot A_1)$
 $A_2 = 0.30 \times 1 \text{ [m}^2\text{]}$
 $R_{\text{plaster_center}} = L_{\text{plaster_center}} / (k_{\text{plaster}} \cdot A_3)$
 $A_3 = 0.015 \times 1 \text{ [m}^2\text{]}$
 $R_{\text{brick}} = L_{\text{brick}} / (k_{\text{brick}} \cdot A_2)$
 $R_{\text{conv_2}} = 1/(h_2 \cdot A_1)$
 $1/R_{\text{mid}} = 2 \cdot 1/R_{\text{plaster_center}} + 1/R_{\text{brick}}$
 $R_{\text{total}} = R_{\text{conv_1}} + R_{\text{foam}} + 2 \cdot R_{\text{plaster_side}} + R_{\text{mid}} + R_{\text{conv_2}}$
 $\dot{Q}_{\text{dot}} = (T_{\text{infinity_1}} - T_{\text{infinity_2}}) / R_{\text{total}}$
 $\dot{Q}_{\text{dot_total}} = \dot{Q}_{\text{dot}} \cdot A / A_1$

$L_{\text{foam}} \text{ [cm]}$	$\dot{Q}_{\text{total}} \text{ [W]}$
1	634.6
2	456.2
3	356.1
4	292
5	247.4
6	214.7
7	189.6
8	169.8
9	153.7
10	140.4



10-56 A wall is to be constructed of 10-cm thick wood studs or with pairs of 5-cm thick wood studs nailed to each other. The rate of heat transfer through the solid stud and through a stud pair nailed to each other, as well as the effective conductivity of the nailed stud pair are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer can be approximated as being one-dimensional since it is predominantly in the x direction. 3 Thermal conductivities are constant. 4 The thermal contact resistance between the two layers is negligible. 4 Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be $k = 0.11 \text{ W/m}\cdot^\circ\text{C}$ for wood studs and $k = 50 \text{ W/m}\cdot^\circ\text{C}$ for manganese steel nails.

Analysis (a) The heat transfer area of the stud is $A = (0.1 \text{ m})(2.5 \text{ m}) = 0.25 \text{ m}^2$. The thermal resistance and heat transfer rate through the solid stud are

$$R_{stud} = \frac{L}{kA} = \frac{0.1 \text{ m}}{(0.11 \text{ W/m}\cdot^\circ\text{C})(0.25 \text{ m}^2)} = 3.636^\circ\text{C/W}$$

$$\dot{Q} = \frac{\Delta T}{R_{stud}} = \frac{8^\circ\text{C}}{3.636^\circ\text{C/W}} = 2.2 \text{ W}$$

(b) The thermal resistances of stud pair and nails are in parallel

$$A_{nails} = 50 \frac{\pi D^2}{4} = 50 \left[\frac{\pi (0.004 \text{ m})^2}{4} \right] = 0.000628 \text{ m}^2$$

$$R_{nails} = \frac{L}{kA} = \frac{0.1 \text{ m}}{(50 \text{ W/m}\cdot^\circ\text{C})(0.000628 \text{ m}^2)} = 3.18^\circ\text{C/W}$$

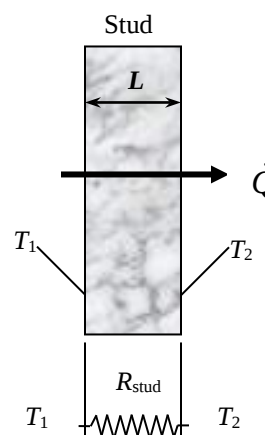
$$R_{stud} = \frac{L}{kA} = \frac{0.1 \text{ m}}{(0.11 \text{ W/m}\cdot^\circ\text{C})(0.25 - 0.000628 \text{ m}^2)} = 3.65^\circ\text{C/W}$$

$$\frac{1}{R_{total}} = \frac{1}{R_{stud}} + \frac{1}{R_{nails}} = \frac{1}{3.65} + \frac{1}{3.18} \longrightarrow R_{total} = 1.70^\circ\text{C/W}$$

$$\dot{Q} = \frac{\Delta T}{R_{stud}} = \frac{8^\circ\text{C}}{1.70^\circ\text{C/W}} = 4.7 \text{ W}$$

(c) The effective conductivity of the nailed stud pair can be determined from

$$\dot{Q} = k_{eff} A \frac{\Delta T}{L} \longrightarrow k_{eff} = \frac{\dot{Q}L}{\Delta TA} = \frac{(4.7 \text{ W})(0.1 \text{ m})}{(8^\circ\text{C})(0.25 \text{ m}^2)} = 0.235 \text{ W/m}\cdot^\circ\text{C}$$

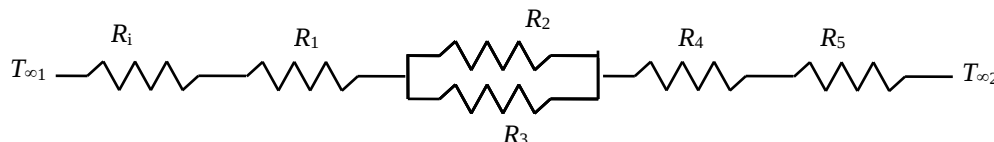


10-57 A wall is constructed of two layers of sheetrock spaced by 5 cm × 12 cm wood studs. The space between the studs is filled with fiberglass insulation. The thermal resistance of the wall and the rate of heat transfer through the wall are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the wall is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be $k = 0.17 \text{ W/m} \cdot ^\circ\text{C}$ for sheetrock, $k = 0.11 \text{ W/m} \cdot ^\circ\text{C}$ for wood studs, and $k = 0.034 \text{ W/m} \cdot ^\circ\text{C}$ for fiberglass insulation.

Analysis (a) The representative surface area is $A = 1 \times 0.65 = 0.65 \text{ m}^2$. The thermal resistance network and the individual thermal resistances are



$$R_i = \frac{1}{h_i A} = \frac{1}{(8.3 \text{ W/m}^2 \cdot ^\circ\text{C})(0.65 \text{ m}^2)} = 0.185 ^\circ\text{C/W}$$

$$R_1 = R_4 = R_{\text{sheetrock}} = \frac{L}{kA} = \frac{0.01 \text{ m}}{(0.17 \text{ W/m} \cdot ^\circ\text{C})(0.65 \text{ m}^2)} = 0.090 ^\circ\text{C/W}$$

$$R_2 = R_{\text{stud}} = \frac{L}{kA} = \frac{0.16 \text{ m}}{(0.11 \text{ W/m} \cdot ^\circ\text{C})(0.05 \text{ m}^2)} = 29.091 ^\circ\text{C/W}$$

$$R_3 = R_{\text{fiberglass}} = \frac{L}{kA} = \frac{0.16 \text{ m}}{(0.034 \text{ W/m} \cdot ^\circ\text{C})(0.60 \text{ m}^2)} = 7.843 ^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(34 \text{ W/m}^2 \cdot ^\circ\text{C})(0.65 \text{ m}^2)} = 0.045 ^\circ\text{C/W}$$

$$\frac{1}{R_{\text{mid}}} = \frac{1}{R_2} + \frac{1}{R_3} = \frac{1}{29.091} + \frac{1}{7.843} \longrightarrow R_{\text{mid}} = 6.178 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_1 + R_{\text{mid}} + R_4 + R_o = 0.185 + 0.090 + 6.178 + 0.090 + 0.045 = \mathbf{6.588 ^\circ\text{C/W}} \text{ (for a } 1 \text{ m} \times 0.65 \text{ m section)}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[20 - (-9)] ^\circ\text{C}}{6.588 ^\circ\text{C/W}} = 4.40 \text{ W}$$

(b) Then steady rate of heat transfer through entire wall becomes

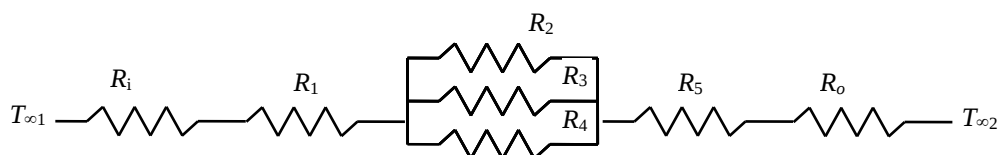
$$\dot{Q}_{\text{total}} = (4.40 \text{ W}) \frac{(12 \text{ m})(5 \text{ m})}{0.65 \text{ m}^2} = \mathbf{406 \text{ W}}$$

10-58E A wall is to be constructed using solid bricks or identical size bricks with 9 square air holes. There is a 0.5 in thick sheetrock layer between two adjacent bricks on all four sides, and on both sides of the wall. The rates of heat transfer through the wall constructed of solid bricks and of bricks with air holes are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the wall is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be $k = 0.40 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for bricks, $k = 0.015 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for air, and $k = 0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for sheetrock.

Analysis (a) The representative surface area is $A = (7.5/12)(7.5/12) = 0.3906 \text{ ft}^2$. The thermal resistance network and the individual thermal resistances if the wall is constructed of solid bricks are



$$R_i = \frac{1}{h_i A} = \frac{1}{(1.5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.3906 \text{ ft}^2)} = 1.7068 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_1 = R_5 = R_{\text{plaster}} = \frac{L}{kA} = \frac{0.5/12 \text{ ft}}{(0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.3906 \text{ ft}^2)} = 1.0667 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_2 = R_{\text{plaster}} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})[(7.5/12) \times (0.5/12)] \text{ ft}^2} = 288 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_3 = R_{\text{plaster}} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})[(7/12) \times (0.5/12)] \text{ ft}^2} = 308.57 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_4 = R_{\text{brick}} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.40 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})[(7/12) \times (7/12)] \text{ ft}^2} = 5.51 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(4 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.3906 \text{ ft}^2)} = 0.64 \text{ h}\cdot^\circ\text{F/Btu}$$

$$\frac{1}{R_{\text{mid}}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{288} + \frac{1}{308.57} + \frac{1}{5.51} \longrightarrow R_{\text{mid}} = 5.3135 \text{ h}\cdot^\circ\text{F/Btu}$$

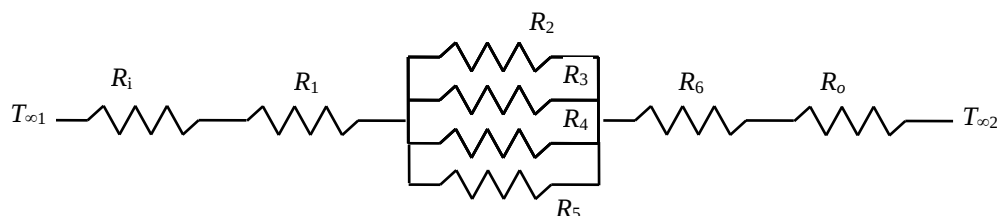
$$R_{\text{total}} = R_i + R_1 + R_{\text{mid}} + R_5 + R_o = 1.7068 + 1.0667 + 5.3135 + 1.0667 + 0.64 = 9.7937 \text{ h}\cdot^\circ\text{F/Btu}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(80 - 30)^\circ\text{F}}{9.7937 \text{ h}\cdot^\circ\text{F/Btu}} = 5.1053 \text{ Btu/h}$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{\text{total}} = (5.1053 \text{ Btu/h}) \frac{(30 \text{ ft})(10 \text{ ft})}{0.3906 \text{ m}^2} = \mathbf{3921 \text{ Btu/h}}$$

(b) The thermal resistance network and the individual thermal resistances if the wall is constructed of bricks with air holes are



$$A_{\text{airholes}} = 9(1.5/12) \times (1.5/12) = 0.1406 \text{ ft}^2$$

$$A_{\text{bricks}} = (7/12 \text{ ft})^2 - 0.1406 = 0.1997 \text{ ft}^2$$

$$R_4 = R_{\text{airholes}} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.015 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F})(0.1406 \text{ ft}^2)} = 355.62 \text{ h} \cdot ^\circ\text{F/Btu}$$

$$R_5 = R_{\text{brick}} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.40 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F})(0.1997 \text{ ft}^2)} = 9.389 \text{ h} \cdot ^\circ\text{F/Btu}$$

$$\frac{1}{R_{\text{mid}}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5} = \frac{1}{288} + \frac{1}{308.57} + \frac{1}{355.62} + \frac{1}{9.389} \longrightarrow R_{\text{mid}} = 8.618 \text{ h} \cdot ^\circ\text{F/Btu}$$

$$R_{\text{total}} = R_i + R_1 + R_{\text{mid}} + R_6 + R_o = 1.7068 + 1.0667 + 8.618 + 1.0667 + 0.64 = 13.0992 \text{ h} \cdot ^\circ\text{F/Btu}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(80 - 30)^\circ\text{F}}{13.0992 \text{ h} \cdot ^\circ\text{F/Btu}} = 3.817 \text{ Btu/h}$$

Then steady rate of heat transfer through entire wall becomes

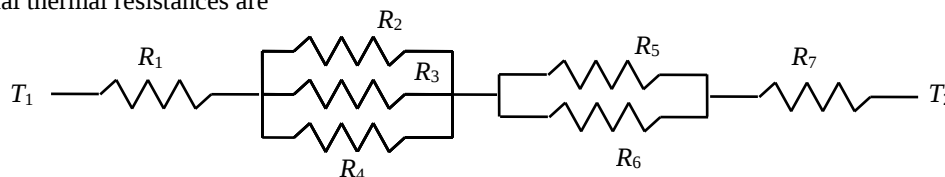
$$\dot{Q}_{\text{total}} = (3.817 \text{ Btu/h}) \frac{(30 \text{ ft})(10 \text{ ft})}{0.3906 \text{ ft}^2} = \mathbf{2932 \text{ Btu/h}}$$

10-59 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the wall is one-dimensional. 3 Thermal conductivities are constant. 4 Thermal contact resistances at the interfaces are disregarded.

Properties The thermal conductivities are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, $k_E = 35$ W/m·°C.

Analysis (a) The representative surface area is $A = 0.12 \times 1 = 0.12$ m². The thermal resistance network and the individual thermal resistances are



$$R_1 = R_A = \left(\frac{L}{kA} \right)_A = \frac{0.01 \text{ m}}{(2 \text{ W/m} \cdot ^\circ\text{C})(0.12 \text{ m}^2)} = 0.04 ^\circ\text{C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA} \right)_C = \frac{0.05 \text{ m}}{(20 \text{ W/m} \cdot ^\circ\text{C})(0.04 \text{ m}^2)} = 0.06 ^\circ\text{C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA} \right)_B = \frac{0.05 \text{ m}}{(8 \text{ W/m} \cdot ^\circ\text{C})(0.04 \text{ m}^2)} = 0.16 ^\circ\text{C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA} \right)_D = \frac{0.1 \text{ m}}{(15 \text{ W/m} \cdot ^\circ\text{C})(0.06 \text{ m}^2)} = 0.11 ^\circ\text{C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA} \right)_E = \frac{0.1 \text{ m}}{(35 \text{ W/m} \cdot ^\circ\text{C})(0.06 \text{ m}^2)} = 0.05 ^\circ\text{C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA} \right)_F = \frac{0.06 \text{ m}}{(2 \text{ W/m} \cdot ^\circ\text{C})(0.12 \text{ m}^2)} = 0.25 ^\circ\text{C/W}$$

$$\frac{1}{R_{\text{mid},1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{\text{mid},1} = 0.025 ^\circ\text{C/W}$$

$$\frac{1}{R_{\text{mid},2}} = \frac{1}{R_5} + \frac{1}{R_6} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{\text{mid},2} = 0.034 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_1 + R_{\text{mid},1} + R_{\text{mid},2} + R_7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349 ^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(300 - 100) ^\circ\text{C}}{0.349 ^\circ\text{C/W}} = 572 \text{ W (for a } 0.12 \text{ m} \times 1 \text{ m section)}$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{\text{total}} = (572 \text{ W}) \frac{(5 \text{ m})(8 \text{ m})}{0.12 \text{ m}^2} = \mathbf{1.91 \times 10^5 \text{ W}}$$

(b) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{\text{total}} = R_1 + R_{\text{mid},1} = 0.04 + 0.025 = 0.065 ^\circ\text{C/W}$$

Then the temperature at the point where the sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{\text{total}}} \longrightarrow T = T_1 - \dot{Q} R_{\text{total}} = 300 ^\circ\text{C} - (572 \text{ W})(0.065 ^\circ\text{C/W}) = \mathbf{263 ^\circ\text{C}}$$

(c) The temperature drop across the section F can be determined from

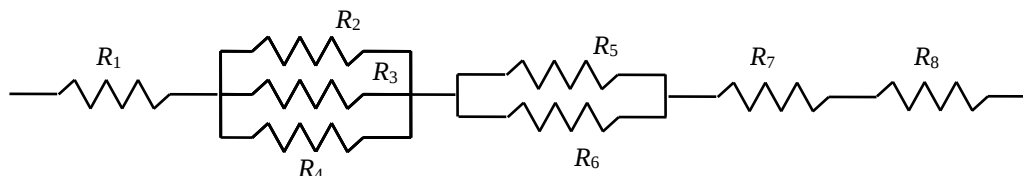
$$\dot{Q} = \frac{\Delta T}{R_F} \longrightarrow \Delta T = \dot{Q} R_F = (572 \text{ W})(0.25 ^\circ\text{C/W}) = \mathbf{143 ^\circ\text{C}}$$

10-60 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the wall is one-dimensional. 3 Thermal conductivities are constant. 4 Thermal contact resistances at the interfaces are to be considered.

Properties The thermal conductivities of various materials used are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, and $k_E = 35$ W/m·°C.

Analysis The representative surface area is $A = 0.12 \times 1 = 0.12$ m²



(a) The thermal resistance network and the individual thermal resistances are

$$R_1 = R_A = \left(\frac{L}{kA} \right)_A = \frac{0.01 \text{ m}}{(2 \text{ W/m} \cdot ^\circ\text{C})(0.12 \text{ m}^2)} = 0.04 ^\circ\text{C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA} \right)_C = \frac{0.05 \text{ m}}{(20 \text{ W/m} \cdot ^\circ\text{C})(0.04 \text{ m}^2)} = 0.06 ^\circ\text{C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA} \right)_B = \frac{0.05 \text{ m}}{(8 \text{ W/m} \cdot ^\circ\text{C})(0.04 \text{ m}^2)} = 0.16 ^\circ\text{C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA} \right)_D = \frac{0.1 \text{ m}}{(15 \text{ W/m} \cdot ^\circ\text{C})(0.06 \text{ m}^2)} = 0.11 ^\circ\text{C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA} \right)_E = \frac{0.1 \text{ m}}{(35 \text{ W/m} \cdot ^\circ\text{C})(0.06 \text{ m}^2)} = 0.05 ^\circ\text{C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA} \right)_F = \frac{0.06 \text{ m}}{(2 \text{ W/m} \cdot ^\circ\text{C})(0.12 \text{ m}^2)} = 0.25 ^\circ\text{C/W}$$

$$R_8 = \frac{0.00012 \text{ m}^2 \cdot ^\circ\text{C/W}}{0.12 \text{ m}^2} = 0.001 ^\circ\text{C/W}$$

$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{mid,1} = 0.025 ^\circ\text{C/W}$$

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_6} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{mid,2} = 0.034 ^\circ\text{C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 + R_8 = 0.04 + 0.025 + 0.034 + 0.25 + 0.001 = 0.350 ^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(300 - 100) ^\circ\text{C}}{0.350 ^\circ\text{C/W}} = 571 \text{ W (for a } 0.12 \text{ m} \times 1 \text{ m section)}$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (571 \text{ W}) \frac{(5 \text{ m})(8 \text{ m})}{0.12 \text{ m}^2} = 1.90 \times 10^5 \text{ W}$$

(b) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{total} = R_1 + R_{mid,1} = 0.04 + 0.025 = 0.065 \text{ } ^\circ\text{C/W}$$

Then the temperature at the point where The sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{total}} \longrightarrow T = T_1 - \dot{Q}R_{total} = 300^\circ\text{C} - (571 \text{ W})(0.065 \text{ } ^\circ\text{C/W}) = \mathbf{263^\circ\text{C}}$$

(c) The temperature drop across the section F can be determined from

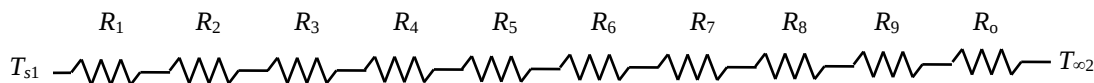
$$\dot{Q} = \frac{\Delta T}{R_F} \longrightarrow \Delta T = \dot{Q}R_F = (571 \text{ W})(0.25 \text{ } ^\circ\text{C/W}) = \mathbf{143^\circ\text{C}}$$

10-61 A coat is made of 5 layers of 0.1 mm thick synthetic fabric separated by 1.5 mm thick air space. The rate of heat loss through the jacket is to be determined, and the result is to be compared to the heat loss through a jacket without the air space. Also, the equivalent thickness of a wool coat is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the jacket is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be $k = 0.13 \text{ W/m}\cdot^\circ\text{C}$ for synthetic fabric, $k = 0.026 \text{ W/m}\cdot^\circ\text{C}$ for air, and $k = 0.035 \text{ W/m}\cdot^\circ\text{C}$ for wool fabric.

Analysis The thermal resistance network and the individual thermal resistances are



$$R_{\text{fabric}} = R_1 = R_3 = R_5 = R_7 = R_9 = \frac{L}{kA} = \frac{0.0001 \text{ m}}{(0.13 \text{ W/m}\cdot^\circ\text{C})(1.25 \text{ m}^2)} = 0.0006^\circ\text{C/W}$$

$$R_{\text{air}} = R_2 = R_4 = R_6 = R_8 = \frac{L}{kA} = \frac{0.0015 \text{ m}}{(0.026 \text{ W/m}\cdot^\circ\text{C})(1.25 \text{ m}^2)} = 0.0462^\circ\text{C/W}$$

$$R_o = \frac{1}{hA} = \frac{1}{(25 \text{ W/m}^2\cdot^\circ\text{C})(1.25 \text{ m}^2)} = 0.0320^\circ\text{C/W}$$

$$R_{\text{total}} = 5R_{\text{fabric}} + 4R_{\text{air}} + R_o = 5 \times 0.0006 + 4 \times 0.0462 + 0.0320 = 0.2198^\circ\text{C/W}$$

and

$$\dot{Q} = \frac{T_{s1} - T_{\infty2}}{R_{\text{total}}} = \frac{(28 - 0)^\circ\text{C}}{0.2198^\circ\text{C/W}} = \mathbf{127 \text{ W}}$$

If the jacket is made of a single layer of 0.5 mm thick synthetic fabric, the rate of heat transfer would be

$$\dot{Q} = \frac{T_{s1} - T_{\infty2}}{R_{\text{total}}} = \frac{T_{s1} - T_{\infty2}}{5 \times R_{\text{fabric}} + R_o} = \frac{(28 - 0)^\circ\text{C}}{(5 \times 0.0006 + 0.0320)^\circ\text{C/W}} = \mathbf{800 \text{ W}}$$

The thickness of a wool fabric that has the same thermal resistance is determined from

$$R_{\text{total}} = R_{\text{wool fabric}} + R_o = \frac{L}{kA} + \frac{1}{hA}$$

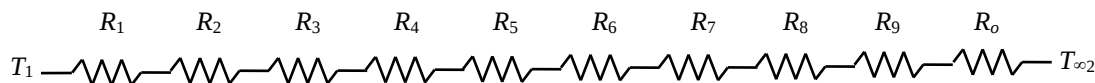
$$0.2198^\circ\text{C/W} = \frac{L}{(0.035 \text{ W/m}\cdot^\circ\text{C})(1.25 \text{ m}^2)} + 0.0320 \longrightarrow L = 0.0082 \text{ m} = \mathbf{8.2 \text{ mm}}$$

10-62 A coat is made of 5 layers of 0.1 mm thick cotton fabric separated by 1.5 mm thick air space. The rate of heat loss through the jacket is to be determined, and the result is to be compared to the heat loss through a jacket without the air space. Also, the equivalent thickness of a wool coat is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the jacket is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be $k = 0.06 \text{ W/m}\cdot^\circ\text{C}$ for cotton fabric, $k = 0.026 \text{ W/m}\cdot^\circ\text{C}$ for air, and $k = 0.035 \text{ W/m}\cdot^\circ\text{C}$ for wool fabric.

Analysis The thermal resistance network and the individual thermal resistances are



$$R_{\text{cotton}} = R_1 = R_3 = R_5 = R_7 = R_9 = \frac{L}{kA} = \frac{0.0001 \text{ m}}{(0.06 \text{ W/m}\cdot^\circ\text{C})(1.25 \text{ m}^2)} = 0.00133 \text{ }^\circ\text{C/W}$$

$$R_{\text{air}} = R_2 = R_4 = R_6 = R_8 = \frac{L}{kA} = \frac{0.0015 \text{ m}}{(0.026 \text{ W/m}\cdot^\circ\text{C})(1.25 \text{ m}^2)} = 0.0462 \text{ }^\circ\text{C/W}$$

$$R_o = \frac{1}{hA} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(1.25 \text{ m}^2)} = 0.0320 \text{ }^\circ\text{C/W}$$

$$R_{\text{total}} = 5R_{\text{fabric}} + 4R_{\text{air}} + R_o = 5 \times 0.00133 + 4 \times 0.0462 + 0.0320 = 0.2235 \text{ }^\circ\text{C/W}$$

and

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(28 - 0)^\circ\text{C}}{0.2235 \text{ }^\circ\text{C/W}} = \mathbf{125 \text{ W}}$$

If the jacket is made of a single layer of 0.5 mm thick cotton fabric, the rate of heat transfer will be

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{T_{s1} - T_{\infty 2}}{5 \times R_{\text{fabric}} + R_o} = \frac{(28 - 0)^\circ\text{C}}{(5 \times 0.00133 + 0.0320) \text{ }^\circ\text{C/W}} = \mathbf{724 \text{ W}}$$

The thickness of a wool fabric for that case can be determined from

$$R_{\text{total}} = R_{\text{wool fabric}} + R_o = \frac{L}{kA} + \frac{1}{hA}$$

$$0.2235 \text{ }^\circ\text{C/W} = \frac{L}{(0.035 \text{ W/m}\cdot^\circ\text{C})(1.25 \text{ m}^2)} + 0.0320 \longrightarrow L = 0.0084 \text{ m} = \mathbf{8.4 \text{ mm}}$$

10-63 A kiln is made of 20 cm thick concrete walls and ceiling. The two ends of the kiln are made of thin sheet metal covered with 2-cm thick styrofoam. For specified indoor and outdoor temperatures, the rate of heat transfer from the kiln is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the walls and ceiling is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer coefficients account for the radiation heat transfer. 5 Heat loss through the floor is negligible. 6 Thermal resistance of sheet metal is negligible.

Properties The thermal conductivities are given to be $k = 0.9 \text{ W/m}\cdot^\circ\text{C}$ for concrete and $k = 0.033 \text{ W/m}\cdot^\circ\text{C}$ for styrofoam insulation.

Analysis In this problem there is a question of which surface area to use. We will use the outer surface area for outer convection resistance, the inner surface area for inner convection resistance, and the average area for the conduction resistance. Or we could use the inner or the outer surface areas in the calculation of all thermal resistances with little loss in accuracy. For top and the two side surfaces:



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(3000 \text{ W/m}^2 \cdot ^\circ\text{C})[(40 \text{ m})(13 - 1.2) \text{ m}]} = 0.0071 \times 10^{-4} \text{ } ^\circ\text{C/W}$$

$$R_{\text{concrete}} = \frac{L}{k A_{\text{ave}}} = \frac{0.2 \text{ m}}{(0.9 \text{ W/m}\cdot^\circ\text{C})[(40 \text{ m})(13 - 0.6) \text{ m}]} = 4.480 \times 10^{-4} \text{ } ^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})[(40 \text{ m})(13 \text{ m})]} = 0.769 \times 10^{-4} \text{ } ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{concrete}} + R_o = (0.0071 + 4.480 + 0.769) \times 10^{-4} = 5.256 \times 10^{-4} \text{ } ^\circ\text{C/W}$$

$$\text{and } \dot{Q}_{\text{top+sides}} = \frac{T_{\text{in}} - T_{\text{out}}}{R_{\text{total}}} = \frac{[40 - (-4)]^\circ\text{C}}{5.256 \times 10^{-4} \text{ } ^\circ\text{C/W}} = 83,700 \text{ W}$$

Heat loss through the end surface of the kiln with styrofoam:



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(3000 \text{ W/m}^2 \cdot ^\circ\text{C})[(4 - 0.4)(5 - 0.4) \text{ m}^2]} = 0.201 \times 10^{-4} \text{ } ^\circ\text{C/W}$$

$$R_{\text{styrofoam}} = \frac{L}{k A_{\text{ave}}} = \frac{0.02 \text{ m}}{(0.033 \text{ W/m}\cdot^\circ\text{C})[(4 - 0.2)(5 - 0.2) \text{ m}^2]} = 0.0332 \text{ } ^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})[4 \times 5 \text{ m}^2]} = 0.0020 \text{ } ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{styrofoam}} + R_o = 0.201 \times 10^{-4} + 0.0332 + 0.0020 = 0.0352 \text{ } ^\circ\text{C/W}$$

$$\text{and } \dot{Q}_{\text{end surface}} = \frac{T_{\text{in}} - T_{\text{out}}}{R_{\text{total}}} = \frac{[40 - (-4)]^\circ\text{C}}{0.0352 \text{ } ^\circ\text{C/W}} = 1250 \text{ W}$$

Then the total rate of heat transfer from the kiln becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{top+sides}} + 2\dot{Q}_{\text{side}} = 83,700 + 2 \times 1250 = \mathbf{86,200 \text{ W}}$$

10-64 EES Prob. 10-63 is reconsidered. The effects of the thickness of the wall and the convection heat transfer coefficient on the outer surface of the rate of heat loss from the kiln are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

width=5 [m]
height=4 [m]
length=40 [m]
L_wall=0.2 [m]
k_concrete=0.9 [W/m-C]
T_in=40 [C]
T_out=-4 [C]
L_sheet=0.003 [m]
L_styrofoam=0.02 [m]
k_styrofoam=0.033 [W/m-C]
h_i=3000 [W/m^2-C]
h_o=25 [W/m^2-C]

"ANALYSIS"

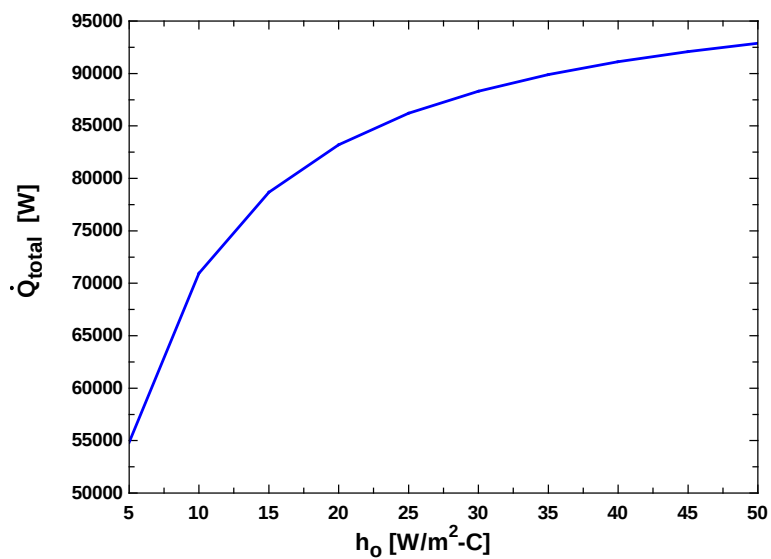
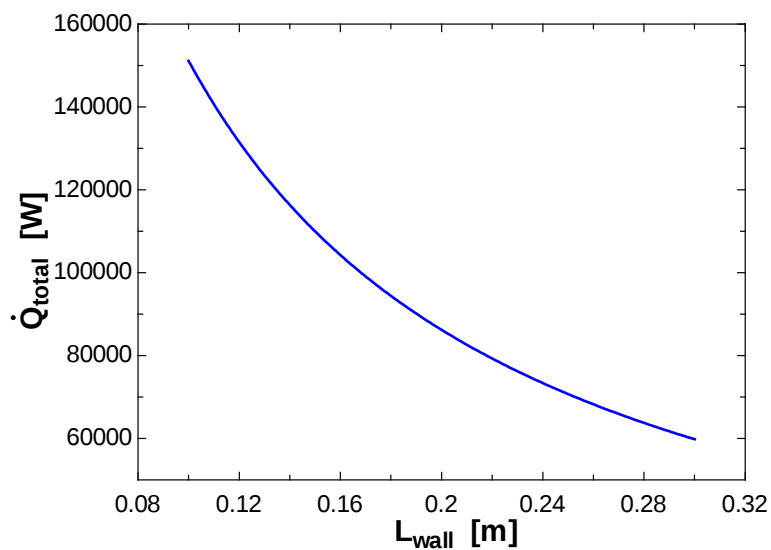
R_conv_i=1/(h_i*A_1)
A_1=(2*height+width-6*L_wall)*length
R_concrete=L_wall/(k_concrete*A_2)
A_2=(2*height+width-3*L_wall)*length
R_conv_o=1/(h_o*A_3)
A_3=(2*height+width)*length
R_total_top_sides=R_conv_i+R_concrete+R_conv_o
Q_dot_top_sides=(T_in-T_out)/R_total_top_sides "Heat loss from top and the two side surfaces"

R_conv_i_end=1/(h_i*A_4)
A_4=(height-2*L_wall)*(width-2*L_wall)
R_styrofoam=L_styrofoam/(k_styrofoam*A_5)
A_5=(height-L_wall)*(width-L_wall)
R_conv_o_end=1/(h_o*A_6)
A_6=height*width
R_total_end=R_conv_i_end+R_styrofoam+R_conv_o_end
Q_dot_end=(T_in-T_out)/R_total_end "Heat loss from one end surface"

Q_dot_total=Q_dot_top_sides+2*Q_dot_end

L _{wall} [m]	Q _{total} [W]
0.1	151098
0.12	131499
0.14	116335
0.16	104251
0.18	94395
0.2	86201
0.22	79281
0.24	73359
0.26	68233
0.28	63751
0.3	59800

h_o [W/m ² .C]	Q_{total} [W]
5	54834
10	70939
15	78670
20	83212
25	86201
30	88318
35	89895
40	91116
45	92089
50	92882



10-65E The thermal resistance of an epoxy glass laminate across its thickness is to be reduced by planting cylindrical copper fillings throughout. The thermal resistance of the epoxy board for heat conduction across its thickness as a result of this modification is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer through the plate is one-dimensional. 3 Thermal conductivities are constant.

Properties The thermal conductivities are given to be $k = 0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for epoxy glass laminate and $k = 223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for copper fillings.

Analysis The thermal resistances of copper fillings and the epoxy board are in parallel. The number of copper fillings in the board and the area they comprise are

$$A_{\text{total}} = (6/12 \text{ ft})(8/12 \text{ ft}) = 0.333 \text{ ft}^2$$

$$n_{\text{copper}} = \frac{0.33 \text{ ft}^2}{(0.06/12 \text{ ft})(0.06/12 \text{ ft})} = 13,333 \text{ (number of copper fillings)}$$

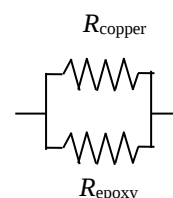
$$A_{\text{copper}} = n \frac{\pi D^2}{4} = 13,333 \frac{\pi (0.02/12 \text{ ft})^2}{4} = 0.0291 \text{ ft}^2$$

$$A_{\text{epoxy}} = A_{\text{total}} - A_{\text{copper}} = 0.3333 - 0.0291 = 0.3042 \text{ ft}^2$$

The thermal resistances are evaluated to be

$$R_{\text{copper}} = \frac{L}{kA} = \frac{0.05/12 \text{ ft}}{(223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.0291 \text{ ft}^2)} = 0.00064 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.05/12 \text{ ft}}{(0.10 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.3042 \text{ ft}^2)} = 0.137 \text{ h}\cdot^\circ\text{F/Btu}$$



Then the thermal resistance of the entire epoxy board becomes

$$\frac{1}{R_{\text{board}}} = \frac{1}{R_{\text{copper}}} + \frac{1}{R_{\text{epoxy}}} = \frac{1}{0.00064} + \frac{1}{0.137} \longrightarrow R_{\text{board}} = \mathbf{0.00064 \text{ h}\cdot^\circ\text{F/Btu}}$$

Heat Conduction in Cylinders and Spheres

10-66C When the diameter of cylinder is very small compared to its length, it can be treated as an infinitely long cylinder. Cylindrical rods can also be treated as being infinitely long when dealing with heat transfer at locations far from the top or bottom surfaces. However, it is not proper to use this model when finding temperatures near the bottom and the top of the cylinder.

10-67C Heat transfer in this short cylinder is one-dimensional since there will be no heat transfer in the axial and tangential directions.

10-68C No. In steady-operation the temperature of a solid cylinder or sphere does not change in radial direction (unless there is heat generation).

10-69 Chilled water is flowing inside a pipe. The thickness of the insulation needed to reduce the temperature rise of water to one-fourth of the original value is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal conductivities are constant. 4 The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity is given to be $k = 0.05 \text{ W/m} \cdot ^\circ\text{C}$ for insulation.

Analysis The rate of heat transfer without the insulation is

$$\dot{Q}_{\text{old}} = \dot{m} c_p \Delta T = (0.98 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C})(8 - 7)^\circ\text{C} = 4096 \text{ W}$$

The total resistance in this case is

$$\dot{Q}_{\text{old}} = \frac{T_\infty - T_w}{R_{\text{total}}}$$

$$4096 \text{ W} = \frac{(30 - 7.5)^\circ\text{C}}{R_{\text{total}}} \longrightarrow R_{\text{total}} = 0.005493^\circ\text{C/W}$$

The convection resistance on the outer surface is

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(9 \text{ W/m}^2 \cdot ^\circ\text{C})\pi(0.05 \text{ m})(150 \text{ m})} = 0.004716^\circ\text{C/W}$$

The rest of thermal resistances are due to convection resistance on the inner surface and the resistance of the pipe and it is determined from

$$R_1 = R_{\text{total}} - R_o = 0.005493 - 0.004716 = 0.0007769^\circ\text{C/W}$$

The rate of heat transfer with the insulation is

$$\dot{Q}_{\text{new}} = \dot{m} c_p \Delta T = (0.98 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C})(0.25^\circ\text{C}) = 1024 \text{ W}$$

The total thermal resistance with the insulation is

$$\dot{Q}_{\text{new}} = \frac{T_\infty - T_w}{R_{\text{total,new}}} \longrightarrow 1024 \text{ W} = \frac{[30 - (7 + 7.25)/2]^\circ\text{C}}{R_{\text{total,new}}} \longrightarrow R_{\text{total,new}} = 0.02234^\circ\text{C/W}$$

It is expressed by

$$R_{\text{total,new}} = R_1 + R_{o,\text{new}} + R_{\text{ins}} = R_1 + \frac{1}{h_o A_o} + \frac{\ln(D_2 / D_1)}{2\pi k_{\text{ins}} L}$$

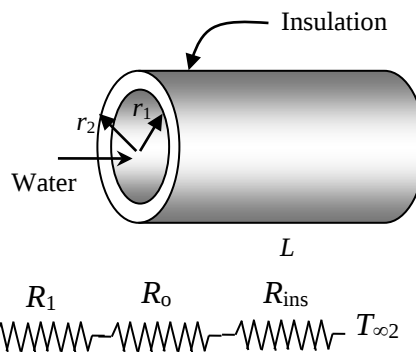
$$0.02234^\circ\text{C/W} = 0.0007769 + \frac{1}{(9 \text{ W/m}^2 \cdot ^\circ\text{C})\pi D_2 (150 \text{ m})} + \frac{\ln(D_2 / 0.05)}{2\pi(0.05 \text{ W/m} \cdot ^\circ\text{C})(150 \text{ m})}$$

Solving this equation by trial-error or by using an equation solver such as EES, we obtain

$$D_2 = 0.1265 \text{ m}$$

Then the required thickness of the insulation becomes

$$t_{\text{ins}} = (D_2 - D_1) / 2 = (0.05 - 0.1265) / 2 = 0.0382 \text{ m} = \mathbf{3.8 \text{ cm}}$$



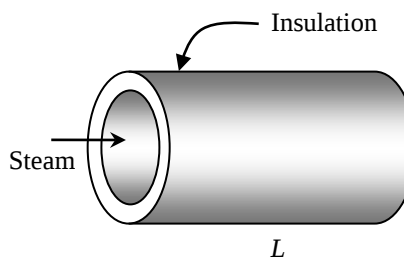
10-70 Steam flows in a steel pipe, which is insulated by gypsum plaster. The rate of heat transfer from the steam and the temperature on the outside surface of the insulation are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal conductivities are constant. 4 The thermal contact resistance at the interface is negligible.

Properties (a) The thermal conductivities of steel and gypsum plaster are given to be 50 and 0.5 W/m·°C, respectively.

Analysis The thermal resistances are

$$T_i \text{ --- } R_i \text{ --- } R_{\text{steel}} \text{ --- } R_{\text{ins}} \text{ --- } R_o \text{ --- } T_o$$



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(800 \text{ W/m}^2 \cdot ^\circ\text{C})\pi(0.06 \text{ m})(20 \text{ m})} = 0.0003316^\circ\text{C/W}$$

$$R_{\text{steel}} = \frac{\ln(D_2 / D_1)}{2\pi k_{\text{steel}} L} = \frac{\ln(8 / 6)}{2\pi(50 \text{ W/m} \cdot ^\circ\text{C})(20 \text{ m})} = 0.0000458^\circ\text{C/W}$$

$$R_{\text{ins}} = \frac{\ln(D_3 / D_2)}{2\pi k_{\text{ins}} L} = \frac{\ln(16 / 8)}{2\pi(0.5 \text{ W/m} \cdot ^\circ\text{C})(20 \text{ m})} = 0.011032^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(200 \text{ W/m}^2 \cdot ^\circ\text{C})\pi(0.16 \text{ m})(20 \text{ m})} = 0.0004974^\circ\text{C/W}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_i + R_{\text{steel}} + R_{\text{ins}} + R_o = 0.0003316 + 0.0000458 + 0.011032 + 0.0004974 = 0.011907^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_i - T_o}{R_{\text{total}}} = \frac{(200 - 10)^\circ\text{C}}{0.011907 \text{ m}^2 \cdot ^\circ\text{C/W}} = \mathbf{15,957 \text{ W}}$$

(b) The temperature at the outer surface of the insulation is determined from

$$\dot{Q} = \frac{T_s - T_o}{R_o} \longrightarrow 15,957 \text{ W} = \frac{(T_s - 10)^\circ\text{C}}{0.0004974 \text{ m}^2 \cdot ^\circ\text{C/W}} \longrightarrow T_s = \mathbf{17.9^\circ\text{C}}$$

10-71 A spherical container filled with iced water is subjected to convection and radiation heat transfer at its outer surface. The rate of heat transfer and the amount of ice that melts per day are to be determined.

Assumptions 1 Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. 3 Thermal conductivity is constant.

Properties The thermal conductivity of steel is given to be $k = 15 \text{ W/m}\cdot^\circ\text{C}$. The heat of fusion of water at 1 atm is $h_{if} = 333.7 \text{ kJ/kg}$. The outer surface of the tank is black and thus its emissivity is $\varepsilon = 1$.

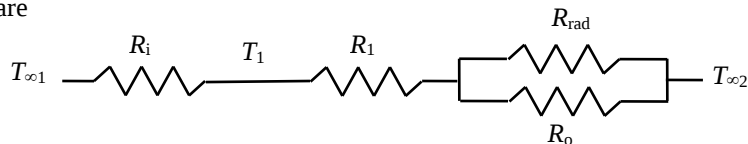
Analysis (a) The inner and the outer surface areas of sphere are

$$A_i = \pi D_i^2 = \pi (8 \text{ m})^2 = 201.06 \text{ m}^2 \quad A_o = \pi D_o^2 = \pi (8.03 \text{ m})^2 = 202.57 \text{ m}^2$$

We assume the outer surface temperature T_2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surfaces of the tank. With this assumption, the radiation heat transfer coefficient can be determined from

$$\begin{aligned} h_{rad} &= \varepsilon \sigma (T_2^2 + T_{surr}^2)(T_2 + T_{surr}) \\ &= 1(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(273 + 5 \text{ K})^2 + (273 + 25 \text{ K})^2](273 + 25 \text{ K})(273 + 5 \text{ K}) \\ &= 5.424 \text{ W/m}^2 \cdot \text{K} \end{aligned}$$

The individual thermal resistances are



$$R_{conv,i} = \frac{1}{h_i A} = \frac{1}{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(201.06 \text{ m}^2)} = 0.000062 \text{ }^\circ\text{C/W}$$

$$R_1 = R_{sphere} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(4.015 - 4.0) \text{ m}}{4\pi (15 \text{ W/m}\cdot^\circ\text{C})(4.015 \text{ m})(4.0 \text{ m})} = 0.000005 \text{ }^\circ\text{C/W}$$

$$R_{conv,o} = \frac{1}{h_o A} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(202.57 \text{ m}^2)} = 0.000494 \text{ }^\circ\text{C/W}$$

$$R_{rad} = \frac{1}{h_{rad} A} = \frac{1}{(5.424 \text{ W/m}^2 \cdot ^\circ\text{C})(202.57 \text{ m}^2)} = 0.000910 \text{ }^\circ\text{C/W}$$

$$\frac{1}{R_{eqv}} = \frac{1}{R_{conv,o}} + \frac{1}{R_{rad}} = \frac{1}{0.000494} + \frac{1}{0.000910} \rightarrow R_{eqv} = 0.000320 \text{ }^\circ\text{C/W}$$

$$R_{total} = R_{conv,i} + R_1 + R_{eqv} = 0.000062 + 0.000005 + 0.000320 = 0.000387 \text{ }^\circ\text{C/W}$$

Then the steady rate of heat transfer to the iced water becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(25 - 0)^\circ\text{C}}{0.000387 \text{ }^\circ\text{C/W}} = \mathbf{64,600 \text{ W}}$$

(b) The total amount of heat transfer during a 24-hour period and the amount of ice that will melt during this period are

$$Q = \dot{Q} \Delta t = (64,600 \text{ kJ/s})(24 \times 3600 \text{ s}) = 5.581 \times 10^6 \text{ kJ}$$

$$m_{ice} = \frac{Q}{h_{if}} = \frac{5.581 \times 10^6 \text{ kJ}}{333.7 \text{ kJ/kg}} = \mathbf{16,730 \text{ kg}}$$

Check: The outer surface temperature of the tank is

$$\begin{aligned} \dot{Q} &= h_{conv+rad} A_o (T_{\infty 1} - T_s) \\ \rightarrow T_s &= T_{\infty 1} - \frac{\dot{Q}}{h_{conv+rad} A_o} = 25^\circ\text{C} - \frac{64,600 \text{ W}}{(10 + 5.424 \text{ W/m}^2 \cdot ^\circ\text{C})(202.57 \text{ m}^2)} = 4.3^\circ\text{C} \end{aligned}$$

which is very close to the assumed temperature of 5°C for the outer surface temperature used in the evaluation of the radiation heat transfer coefficient. Therefore, there is no need to repeat the calculations.

10-72 A steam pipe covered with 10-cm thick glass wool insulation is subjected to convection on its surfaces. The rate of heat transfer per unit length and the temperature drops across the pipe and the insulation are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. 3 Thermal conductivities are constant. 4 The thermal contact resistance at the interface is negligible.

Properties The thermal conductivities are given to be $k = 15 \text{ W/m} \cdot ^\circ\text{C}$ for steel and $k = 0.038 \text{ W/m} \cdot ^\circ\text{C}$ for glass wool insulation

Analysis The inner and the outer surface areas of the insulated pipe per unit length are

$$A_i = \pi D_i L = \pi(0.05 \text{ m})(1 \text{ m}) = 0.157 \text{ m}^2$$

$$A_o = \pi D_o L = \pi(0.055 + 0.06 \text{ m})(1 \text{ m}) = 0.361 \text{ m}^2$$

The individual thermal resistances are



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.157 \text{ m}^2)} = 0.08 ^\circ\text{C/W}$$

$$R_1 = R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k_1 L} = \frac{\ln(2.75 / 2.5)}{2\pi(15 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} = 0.00101 ^\circ\text{C/W}$$

$$R_2 = R_{\text{insulation}} = \frac{\ln(r_3 / r_2)}{2\pi k_2 L} = \frac{\ln(5.75 / 2.75)}{2\pi(0.038 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} = 3.089 ^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(15 \text{ W/m}^2 \cdot ^\circ\text{C})(0.361 \text{ m}^2)} = 0.1847 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_1 + R_2 + R_o = 0.08 + 0.00101 + 3.089 + 0.1847 = 3.355 ^\circ\text{C/W}$$

Then the steady rate of heat loss from the steam per m. pipe length becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(320 - 5) ^\circ\text{C}}{3.355 ^\circ\text{C/W}} = \mathbf{93.9 \text{ W}}$$

The temperature drops across the pipe and the insulation are

$$\Delta T_{\text{pipe}} = \dot{Q} R_{\text{pipe}} = (93.9 \text{ W})(0.00101 ^\circ\text{C/W}) = \mathbf{0.095 ^\circ\text{C}}$$

$$\Delta T_{\text{insulation}} = \dot{Q} R_{\text{insulation}} = (93.9 \text{ W})(3.089 ^\circ\text{C/W}) = \mathbf{290 ^\circ\text{C}}$$

10-73 EES Prob. 10-72 is reconsidered. The effect of the thickness of the insulation on the rate of heat loss from the steam and the temperature drop across the insulation layer are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$T_{\text{infinity}_1}=320 \text{ [C]}; \quad T_{\text{infinity}_2}=5 \text{ [C]}$$

$$k_{\text{steel}}=15 \text{ [W/m-C]}$$

$$D_i=0.05 \text{ [m]}; \quad D_o=0.055 \text{ [m]}$$

$$r_1=D_i/2; \quad r_2=D_o/2$$

$$t_{\text{ins}}=3 \text{ [cm]}$$

$$k_{\text{ins}}=0.038 \text{ [W/m-C]}$$

$$h_o=15 \text{ [W/m}^2\text{-C]}$$

$$h_i=80 \text{ [W/m}^2\text{-C]}$$

$$L=1 \text{ [m]}$$

"ANALYSIS"

$$A_i=\pi D_i L$$

$$A_o=\pi(D_o+2t_{\text{ins}}\text{Convert(cm, m)})L$$

$$R_{\text{conv}_i}=1/(h_i A_i)$$

$$R_{\text{pipe}}=\ln(r_2/r_1)/(2\pi k_{\text{steel}}L)$$

$$R_{\text{ins}}=\ln(r_3/r_2)/(2\pi k_{\text{ins}}L)$$

$$r_3=r_2+t_{\text{ins}}\text{Convert(cm, m)} \quad \text{"t_{ins} is in cm"}$$

$$R_{\text{conv}_o}=1/(h_o A_o)$$

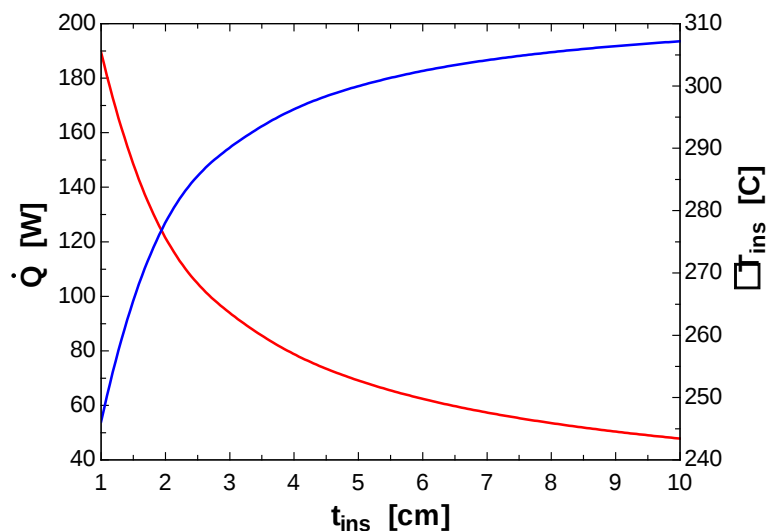
$$R_{\text{total}}=R_{\text{conv}_i}+R_{\text{pipe}}+R_{\text{ins}}+R_{\text{conv}_o}$$

$$\dot{Q}_{\text{dot}}=(T_{\text{infinity}_1}-T_{\text{infinity}_2})/R_{\text{total}}$$

$$\Delta T_{\text{pipe}}=\dot{Q}_{\text{dot}} R_{\text{pipe}}$$

$$\Delta T_{\text{ins}}=\dot{Q}_{\text{dot}} R_{\text{ins}}$$

T_{ins} [cm]	$\dot{Q}$ [W]	ΔT_{ins} [C]
1	189.5	246.1
2	121.5	278.1
3	93.91	290.1
4	78.78	296.3
5	69.13	300
6	62.38	302.4
7	57.37	304.1
8	53.49	305.4
9	50.37	306.4
10	47.81	307.2



10-74 CD EES A 50-m long section of a steam pipe passes through an open space at 15°C. The rate of heat loss from the steam pipe, the annual cost of this heat loss, and the thickness of fiberglass insulation needed to save 90 percent of the heat lost are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal conductivity is constant. **4** The thermal contact resistance at the interface is negligible. **5** The pipe temperature remains constant at about 150°C with or without insulation. **6** The combined heat transfer coefficient on the outer surface remains constant even after the pipe is insulated.

Properties The thermal conductivity of fiberglass insulation is given to be $k = 0.035 \text{ W/m}\cdot^\circ\text{C}$.

Analysis (a) The rate of heat loss from the steam pipe is

$$A_o = \pi DL = \pi(0.1 \text{ m})(50 \text{ m}) = 15.71 \text{ m}^2$$

$$\dot{Q}_{\text{bare}} = h_o A(T_s - T_{\text{air}}) = (20 \text{ W/m}^2 \cdot ^\circ\text{C})(15.71 \text{ m}^2)(150 - 15)^\circ\text{C} = \mathbf{42,412 \text{ W}}$$



(b) The amount of heat loss per year is

$$Q = \dot{Q}\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10^9 \text{ kJ/yr}$$

The amount of gas consumption from the natural gas furnace that has an efficiency of 75% is

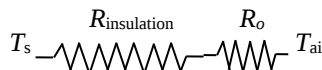
$$Q_{\text{gas}} = \frac{1.337 \times 10^9 \text{ kJ/yr}}{0.75} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 16,903 \text{ therms/yr}$$

The annual cost of this energy lost is

$$\begin{aligned} \text{Energy cost} &= (\text{Energy used})(\text{Unit cost of energy}) \\ &= (16,903 \text{ therms/yr})(\$0.52 / \text{therm}) = \mathbf{\$8790/\text{yr}} \end{aligned}$$

(c) In order to save 90% of the heat loss and thus to reduce it to $0.1 \times 42,412 = 4241 \text{ W}$, the thickness of insulation needed is determined from

$$\dot{Q}_{\text{insulated}} = \frac{T_s - T_{\text{air}}}{R_o + R_{\text{insulation}}} = \frac{T_s - T_{\text{air}}}{\frac{1}{h_o A_o} + \frac{\ln(r_2 / r_1)}{2\pi k L}}$$



Substituting and solving for r_2 , we get

$$4241 \text{ W} = \frac{(150 - 15)^\circ\text{C}}{\frac{1}{(20 \text{ W/m}^2 \cdot ^\circ\text{C})[(2\pi r_2)(50 \text{ m})]} + \frac{\ln(r_2 / 0.05)}{2\pi(0.035 \text{ W/m}\cdot^\circ\text{C})(50 \text{ m})}} \longrightarrow r_2 = 0.0692 \text{ m}$$

Then the thickness of insulation becomes

$$t_{\text{insulation}} = r_2 - r_1 = 6.92 - 5 = \mathbf{1.92 \text{ cm}}$$

10-75 An electric hot water tank is made of two concentric cylindrical metal sheets with foam insulation in between. The fraction of the hot water cost that is due to the heat loss from the tank and the payback period of the do-it-yourself insulation kit are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. 3 Thermal conductivities are constant. 4 The thermal resistances of the water tank and the outer thin sheet metal shell are negligible. 5 Heat loss from the top and bottom surfaces is negligible.

Properties The thermal conductivities are given to be $k = 0.03 \text{ W/m} \cdot ^\circ\text{C}$ for foam insulation and $k = 0.035 \text{ W/m} \cdot ^\circ\text{C}$ for fiber glass insulation

Analysis We consider only the side surfaces of the water heater for simplicity, and disregard the top and bottom surfaces (it will make difference of about 10 percent). The individual thermal resistances are

$$A_i = \pi D_i L = \pi(0.40 \text{ m})(2 \text{ m}) = 2.51 \text{ m}^2$$

$$R_o = \frac{1}{h_i A_i} = \frac{1}{(50 \text{ W/m}^2 \cdot ^\circ\text{C})(2.51 \text{ m}^2)} = 0.008 ^\circ\text{C/W}$$

$$A_o = \pi D_o L = \pi(0.46 \text{ m})(2 \text{ m}) = 2.89 \text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(12 \text{ W/m}^2 \cdot ^\circ\text{C})(2.89 \text{ m}^2)} = 0.029 ^\circ\text{C/W}$$

$$R_{\text{foam}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(23 / 20)}{2\pi(0.03 \text{ W/m}^2 \cdot ^\circ\text{C})(2 \text{ m})} = 0.37 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_o + R_{\text{foam}} = 0.008 + 0.029 + 0.37 = 0.407 ^\circ\text{C/W}$$

The rate of heat loss from the hot water tank is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{\text{total}}} = \frac{(55 - 27)^\circ\text{C}}{0.407 ^\circ\text{C/W}} = 68.8 \text{ W}$$

The amount and cost of heat loss per year are

$$Q = \dot{Q} \Delta t = (0.0688 \text{ kW})(365 \times 24 \text{ h/yr}) = 602.7 \text{ kWh/yr}$$

$$\text{Cost of Energy} = (\text{Amount of energy})(\text{Unit cost}) = (602.7 \text{ kWh})(\$0.08 / \text{kWh}) = \$48.22$$

$$f = \frac{\$48.22}{\$280} = 0.1722 = \mathbf{17.2\%}$$

If 3 cm thick fiber glass insulation is used to wrap the entire tank, the individual resistances becomes

$$A_o = \pi D_o L = \pi(0.52 \text{ m})(2 \text{ m}) = 3.267 \text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(12 \text{ W/m}^2 \cdot ^\circ\text{C})(3.267 \text{ m}^2)} = 0.026 ^\circ\text{C/W}$$

$$R_{\text{foam}} = \frac{\ln(r_2 / r_1)}{2\pi k_1 L} = \frac{\ln(23 / 20)}{2\pi(0.03 \text{ W/m}^2 \cdot ^\circ\text{C})(2 \text{ m})} = 0.371 ^\circ\text{C/W}$$

$$R_{\text{fiberglass}} = \frac{\ln(r_3 / r_2)}{2\pi k_2 L} = \frac{\ln(26 / 23)}{2\pi(0.035 \text{ W/m}^2 \cdot ^\circ\text{C})(2 \text{ m})} = 0.279 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_o + R_{\text{foam}} + R_{\text{fiberglass}} = 0.008 + 0.026 + 0.371 + 0.279 = 0.684 ^\circ\text{C/W}$$

The rate of heat loss from the hot water heater in this case is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{\text{total}}} = \frac{(55 - 27)^\circ\text{C}}{0.684 ^\circ\text{C/W}} = 40.94 \text{ W}$$

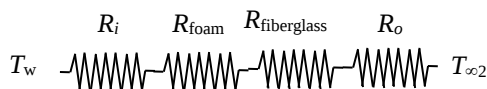
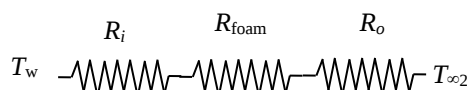
The energy saving is

$$\text{saving} = 70 - 40.94 = 29.06 \text{ W}$$

The time necessary for this additional insulation to pay for its cost of \$30 is then determined to be

$$\text{Cost} = (0.02906 \text{ kW})(\text{Time period})(\$0.08 / \text{kWh}) = \$30$$

Then, Time period = 12,904 hours = **538 days $\approx$ 1.5 years**



10-76 EES Prob. 10-75 is reconsidered. The fraction of energy cost of hot water due to the heat loss from the tank as a function of the hot-water temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

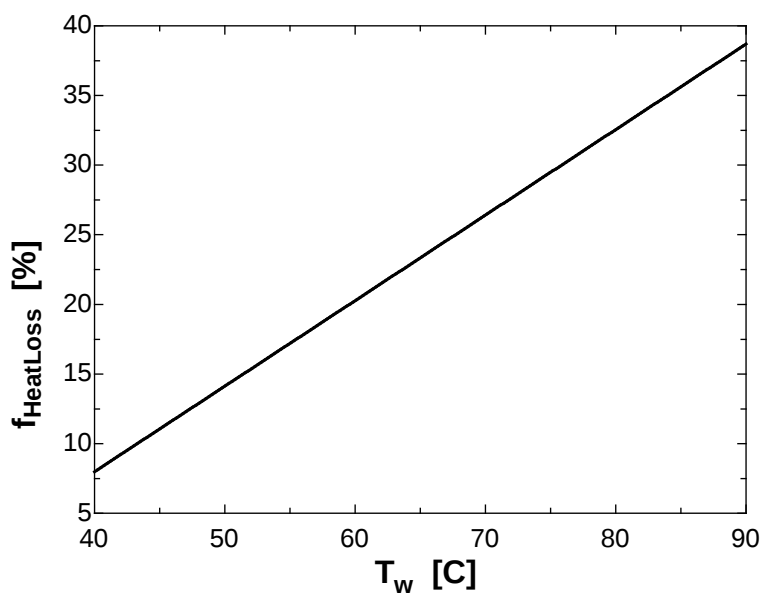
"GIVEN"

L=2 [m]
D_i=0.40 [m]
D_o=0.46 [m]
r_1=D_i/2
r_2=D_o/2
T_w=55 [C]
T_infinity_2=27 [C]
h_i=50 [W/m^2-C]
h_o=12 [W/m^2-C]
k_ins=0.03 [W/m-C]
Price_electric=0.08 [\$/kWh]
Cost_heating=280 [\$/year]

"ANALYSIS"

A_i=pi*D_i*L
A_o=pi*D_o*L
R_conv_i=1/(h_i*A_i)
R_ins=ln(r_2/r_1)/(2*pi*k_ins*L)
R_conv_o=1/(h_o*A_o)
R_total=R_conv_i+R_ins+R_conv_o
Q_dot=(T_w-T_infinity_2)/R_total
Q=(Q_dot*Convert(W, kW))*time
time=365*24 [h/year]
Cost_HeatLoss=Q*Price_electric
f_HeatLoss=Cost_HeatLoss/Cost_heating*Convert(, %)

T _w [C]	f _{HeatLoss} [%]
40	7.984
45	11.06
50	14.13
55	17.2
60	20.27
65	23.34
70	26.41
75	29.48
80	32.55
85	35.62
90	38.69



10-77 A cold aluminum canned drink that is initially at a uniform temperature of 3°C is brought into a room air at 25°C. The time it will take for the average temperature of the drink to rise to 15°C with and without rubber insulation is to be determined.

Assumptions 1 The drink is at a uniform temperature at all times. 2 The thermal resistance of the can and the internal convection resistance are negligible so that the can is at the same temperature as the drink inside. 3 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 4 Thermal properties are constant. 5 The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity of rubber insulation is given to be $k = 0.13 \text{ W/m} \cdot ^\circ\text{C}$. For the drink, we use the properties of water at room temperature, $\rho = 1000 \text{ kg/m}^3$ and $c_p = 4180 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis This is a transient heat conduction, and the rate of heat transfer will decrease as the drink warms up and the temperature difference between the drink and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of $(3+15)/2 = 9.5^\circ\text{C}$ during the process. Then the average rate of heat transfer into the drink is

$$A_o = \pi D_o L + 2 \frac{\pi D^2}{4} = \pi(0.06 \text{ m})(0.125 \text{ m}) + 2 \frac{\pi(0.06 \text{ m})^2}{4} = 0.02922 \text{ m}^2$$

$$\dot{Q}_{\text{bare,ave}} = h_o A(T_{\text{air}} - T_{\text{can,ave}}) = (10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02922 \text{ m}^2)(25 - 9.5)^\circ\text{C} = 4.529 \text{ W}$$

The amount of heat that must be supplied to the drink to raise its temperature to 15°C is

$$m = \rho V = \rho \pi^2 L = (1000 \text{ kg/m}^3) \pi (0.03 \text{ m})^2 (0.125 \text{ m}) = 0.3534 \text{ kg}$$

$$Q = mc_p \Delta T = (0.3534 \text{ kg})(4180 \text{ J/kg})(15 - 3)^\circ\text{C} = 16,250 \text{ J}$$

Then the time required for this much heat transfer to take place is

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16,250 \text{ J}}{4.529 \text{ J/s}} = 3588 \text{ s} = \mathbf{59.8 \text{ min}}$$

We now repeat calculations after wrapping the can with 1-cm thick rubber insulation, except the top surface. The rate of heat transfer from the top surface is

$$\dot{Q}_{\text{top,ave}} = h_o A_{\text{top}} (T_{\text{air}} - T_{\text{can,ave}}) = (10 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.03 \text{ m})^2](25 - 9.5)^\circ\text{C} = 0.44 \text{ W}$$

Heat transfer through the insulated side surface is

$$A_o = \pi D_o L = \pi(0.08 \text{ m})(0.125 \text{ m}) = 0.03142 \text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03142 \text{ m}^2)} = 3.183^\circ\text{C/W}$$

$$R_{\text{insulation,side}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(4/3)}{2\pi(0.13 \text{ W/m}^2 \cdot ^\circ\text{C})(0.125 \text{ m})} = 2.818^\circ\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} = 3.183 + 2.818 = 6.001^\circ\text{C/W}$$

$$\dot{Q}_{\text{side}} = \frac{T_{\text{air}} - T_{\text{can,ave}}}{R_{\text{conv,o}}} = \frac{(25 - 9.5)^\circ\text{C}}{6.001^\circ\text{C/W}} = 2.58 \text{ W}$$



The ratio of bottom to the side surface areas is $(\pi r^2)/(2\pi r L) = r/(2L) = 3/(2 \times 12.5) = 0.12$. Therefore, the effect of heat transfer through the bottom surface can be accounted for approximately by increasing the heat transfer from the side surface by 12%. Then,

$$\dot{Q}_{\text{insulated}} = \dot{Q}_{\text{side+bottom}} + \dot{Q}_{\text{top}} = 1.12 \times 2.58 + 0.44 = 3.33 \text{ W}$$

Then the time of heating becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16,250 \text{ J}}{3.33 \text{ J/s}} = 4880 \text{ s} = \mathbf{81.3 \text{ min}}$$

10-78 A cold aluminum canned drink that is initially at a uniform temperature of 3°C is brought into a room air at 25°C. The time it will take for the average temperature of the drink to rise to 10°C with and without rubber insulation is to be determined.

Assumptions 1 The drink is at a uniform temperature at all times. 2 The thermal resistance of the can and the internal convection resistance are negligible so that the can is at the same temperature as the drink inside. 3 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 4 Thermal properties are constant. 5 The thermal contact resistance at the interface is to be considered.

Properties The thermal conductivity of rubber insulation is given to be $k = 0.13 \text{ W/m} \cdot ^\circ\text{C}$. For the drink, we use the properties of water at room temperature, $\rho = 1000 \text{ kg/m}^3$ and $c_p = 4180 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis This is a transient heat conduction, and the rate of heat transfer will decrease as the drink warms up and the temperature difference between the drink and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of $(3+10)/2 = 6.5^\circ\text{C}$ during the process. Then the average rate of heat transfer into the drink is

$$A_o = \pi D_o L + 2 \frac{\pi D^2}{4} = \pi(0.06 \text{ m})(0.125 \text{ m}) + 2 \frac{\pi(0.06 \text{ m})^2}{4} = 0.02922 \text{ m}^2$$

$$\dot{Q}_{\text{bare,ave}} = h_o A(T_{\text{air}} - T_{\text{can,ave}}) = (10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02922 \text{ m}^2)(25 - 9.5)^\circ\text{C} = 4.529 \text{ W}$$

The amount of heat that must be supplied to the drink to raise its temperature to 15°C is

$$m = \rho V = \rho \pi^2 L = (1000 \text{ kg/m}^3) \pi (0.03 \text{ m})^2 (0.125 \text{ m}) = 0.3534 \text{ kg}$$

$$Q = mc_p \Delta T = (0.3534 \text{ kg})(4180 \text{ J/kg})(15 - 4)^\circ\text{C} = 16,250 \text{ J}$$

Then the time required for this much heat transfer to take place is

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16,250 \text{ J}}{4.529 \text{ J/s}} = 3588 \text{ s} = \mathbf{59.8 \text{ min}}$$

We now repeat calculations after wrapping the can with 1-cm thick rubber insulation, except the top surface. The rate of heat transfer from the top surface is

$$\dot{Q}_{\text{top,ave}} = h_o A_{\text{top}} (T_{\text{air}} - T_{\text{can,ave}}) = (10 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.03 \text{ m})^2](25 - 9.5)^\circ\text{C} = 0.44 \text{ W}$$

Heat transfer through the insulated side surface is

$$A_o = \pi D_o L = \pi(0.08 \text{ m})(0.125 \text{ m}) = 0.03142 \text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(10 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03142 \text{ m}^2)} = 3.183^\circ\text{C/W}$$

$$R_{\text{insulation,side}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(4/3)}{2\pi(0.13 \text{ W/m}^2 \cdot ^\circ\text{C})(0.125 \text{ m})} = 2.818^\circ\text{C/W}$$

$$R_{\text{contact}} = \frac{0.00008 \text{ m}^2 \cdot ^\circ\text{C/W}}{\pi(0.06 \text{ m})(0.125 \text{ m})} = 0.0034^\circ\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} + R_{\text{contact}} = 3.183 + 2.818 + 0.0034 = 6.004^\circ\text{C/W}$$

$$\dot{Q}_{\text{side}} = \frac{T_{\text{air}} - T_{\text{can,ave}}}{R_{\text{conv,o}}} = \frac{(25 - 9.5)^\circ\text{C}}{6.004^\circ\text{C/W}} = 2.58 \text{ W}$$



The ratio of bottom to the side surface areas is $(\pi r^2)/(2\pi r L) = r/(2L) = 3/(2 \times 12.5) = 0.12$. Therefore, the effect of heat transfer through the bottom surface can be accounted for approximately by increasing the heat transfer from the side surface by 12%. Then,

$$\dot{Q}_{\text{insulated}} = \dot{Q}_{\text{side+bottom}} + \dot{Q}_{\text{top}} = 1.12 \times 2.58 + 0.44 = 3.33 \text{ W}$$

Then the time of heating becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16,250 \text{ J}}{3.33 \text{ J/s}} = 4880 \text{ s} = \mathbf{81.3 \text{ min}}$$

Discussion The thermal contact resistance did not have any effect on heat transfer.

10-79E A steam pipe covered with 2-in thick fiberglass insulation is subjected to convection on its surfaces. The rate of heat loss from the steam per unit length and the error involved in neglecting the thermal resistance of the steel pipe in calculations are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. 3 Thermal conductivities are constant. 4 The thermal contact resistance at the interface is negligible.

Properties The thermal conductivities are given to be $k = 8.7 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for steel and $k = 0.020 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for fiberglass insulation.

Analysis The inner and outer surface areas of the insulated pipe are

$$A_i = \pi D_i L = \pi (3.5 / 12 \text{ ft})(1 \text{ ft}) = 0.916 \text{ ft}^2$$

$$A_o = \pi D_o L = \pi (8 / 12 \text{ ft})(1 \text{ ft}) = 2.094 \text{ ft}^2$$

The diagram shows a thermal resistance network with four resistors in series: R_i , R_{pipe} , $R_{\text{insulation}}$, and R_o . The network is connected between $T_{\infty 1}$ on the left and $T_{\infty 2}$ on the right.

The individual resistances are

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(30 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.916 \text{ ft}^2)} = 0.036 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_1 = R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k_1 L} = \frac{\ln(2 / 1.75)}{2\pi (8.7 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1 \text{ ft})} = 0.002 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_2 = R_{\text{insulation}} = \frac{\ln(r_3 / r_2)}{2\pi k_2 L} = \frac{\ln(4 / 2)}{2\pi (0.020 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1 \text{ ft})} = 5.516 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(2.094 \text{ ft}^2)} = 0.096 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{total}} = R_i + R_1 + R_2 + R_o = 0.036 + 0.002 + 5.516 + 0.096 = 5.65 \text{ h}\cdot^\circ\text{F/Btu}$$

Then the steady rate of heat loss from the steam per ft. pipe length becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(450 - 55)^\circ\text{F}}{5.65 \text{ h}\cdot^\circ\text{F/Btu}} = \mathbf{69.91 \text{ Btu/h}}$$

If the thermal resistance of the steel pipe is neglected, the new value of total thermal resistance will be

$$R_{\text{total}} = R_i + R_2 + R_o = 0.036 + 5.516 + 0.096 = 5.648 \text{ h}\cdot^\circ\text{F/Btu}$$

Then the percentage error involved in calculations becomes

$$\text{error}\% = \frac{(5.65 - 5.648) \text{ h}\cdot^\circ\text{F/Btu}}{5.65 \text{ h}\cdot^\circ\text{F/Btu}} \times 100 = \mathbf{0.035\%}$$

which is insignificant.

10-80 Hot water is flowing through a 15-m section of a cast iron pipe. The pipe is exposed to cold air and surfaces in the basement. The rate of heat loss from the hot water and the average velocity of the water in the pipe as it passes through the basement are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. 3 Thermal properties are constant.

Properties The thermal conductivity and emissivity of cast iron are given to be $k = 52 \text{ W/m}\cdot^\circ\text{C}$ and $\varepsilon = 0.7$.

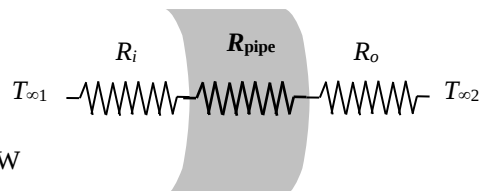
Analysis The individual resistances are

$$A_i = \pi D_i L = \pi(0.04 \text{ m})(15 \text{ m}) = 1.885 \text{ m}^2$$

$$A_o = \pi D_o L = \pi(0.046 \text{ m})(15 \text{ m}) = 2.168 \text{ m}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(120 \text{ W/m}^2\cdot^\circ\text{C})(1.885 \text{ m}^2)} = 0.00442 \text{ }^\circ\text{C/W}$$

$$R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k_1 L} = \frac{\ln(2.3 / 2)}{2\pi(52 \text{ W/m}\cdot^\circ\text{C})(15 \text{ m})} = 0.00003 \text{ }^\circ\text{C/W}$$



The outer surface temperature of the pipe will be somewhat below the water temperature. Assuming the outer surface temperature of the pipe to be 60°C (we will check this assumption later), the radiation heat transfer coefficient is determined to be

$$\begin{aligned} h_{\text{rad}} &= \varepsilon \sigma (T_2^2 + T_{\text{surr}}^2)(T_2 + T_{\text{surr}}) \\ &= (0.7)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(333 \text{ K})^2 + (283 \text{ K})^2](333 + 283) = 4.67 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

Since the surrounding medium and surfaces are at the same temperature, the radiation and convection heat transfer coefficients can be added and the result can be taken as the combined heat transfer coefficient. Then,

$$h_{\text{combined}} = h_{\text{rad}} + h_{\text{conv},2} = 4.67 + 15 = 19.67 \text{ W/m}^2\cdot^\circ\text{C}$$

$$R_o = \frac{1}{h_{\text{combined}} A_o} = \frac{1}{(19.67 \text{ W/m}^2\cdot^\circ\text{C})(2.168 \text{ m}^2)} = 0.02345 \text{ }^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{pipe}} + R_o = 0.00442 + 0.00003 + 0.02345 = 0.0279 \text{ }^\circ\text{C/W}$$

The rate of heat loss from the hot water pipe then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(70 - 10)^\circ\text{C}}{0.0279 \text{ }^\circ\text{C/W}} = \mathbf{2151 \text{ W}}$$

For a temperature drop of 3°C , the mass flow rate of water and the average velocity of water must be

$$\dot{Q} = \dot{m} c_p \Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p \Delta T} = \frac{2151 \text{ J/s}}{(4180 \text{ J/kg}\cdot^\circ\text{C})(3^\circ\text{C})} = 0.172 \text{ kg/s}$$

$$\dot{m} = \rho V A_c \longrightarrow V = \frac{\dot{m}}{\rho A_c} = \frac{0.172 \text{ kg/s}}{(1000 \text{ kg/m}^3) \frac{\pi(0.04 \text{ m})^2}{4}} = \mathbf{0.136 \text{ m/s}}$$

Discussion The outer surface temperature of the pipe is

$$\dot{Q} = \frac{T_{\infty 1} - T_s}{R_i + R_{\text{pipe}}} \rightarrow 2151 \text{ W} = \frac{(70 - T_s)^\circ\text{C}}{(0.00442 + 0.00003)^\circ\text{C/W}} \rightarrow T_s = 60.4^\circ\text{C}$$

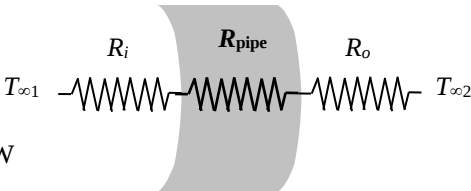
which is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Therefore, there is no need to repeat the calculations.

10-81 Hot water is flowing through a 15 m section of a copper pipe. The pipe is exposed to cold air and surfaces in the basement. The rate of heat loss from the hot water and the average velocity of the water in the pipe as it passes through the basement are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant.

Properties The thermal conductivity and emissivity of copper are given to be $k = 386 \text{ W/m}\cdot^\circ\text{C}$ and $\varepsilon = 0.7$.

Analysis The individual resistances are

$$\begin{aligned}
 A_i &= \pi D_i L = \pi (0.04 \text{ m})(15 \text{ m}) = 1.885 \text{ m}^2 \\
 A_o &= \pi D_o L = \pi (0.046 \text{ m})(15 \text{ m}) = 2.168 \text{ m}^2 \\
 R_i &= \frac{1}{h_i A_i} = \frac{1}{(120 \text{ W/m}^2 \cdot ^\circ\text{C})(1.885 \text{ m}^2)} = 0.00442 \text{ }^\circ\text{C/W} \\
 R_{\text{pipe}} &= \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(2.3 / 2)}{2\pi (386 \text{ W/m}\cdot^\circ\text{C})(15 \text{ m})} = 0.0000038 \text{ }^\circ\text{C/W}
 \end{aligned}$$


The outer surface temperature of the pipe will be somewhat below the water temperature. Assuming the outer surface temperature of the pipe to be 60°C (we will check this assumption later), the radiation heat transfer coefficient is determined to be

$$\begin{aligned}
 h_{\text{rad}} &= \varepsilon \sigma (T_2^2 + T_{\text{surr}}^2)(T_2 + T_{\text{surr}}) \\
 &= (0.7)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(333 \text{ K})^2 + (283 \text{ K})^2](333 + 283) = 4.67 \text{ W/m}^2 \cdot \text{K}
 \end{aligned}$$

Since the surrounding medium and surfaces are at the same temperature, the radiation and convection heat transfer coefficients can be added and the result can be taken as the combined heat transfer coefficient. Then,

$$\begin{aligned}
 h_{\text{combined}} &= h_{\text{rad}} + h_{\text{conv},2} = 4.67 + 15 = 19.67 \text{ W/m}^2 \cdot ^\circ\text{C} \\
 R_o &= \frac{1}{h_{\text{combined}} A_o} = \frac{1}{(19.67 \text{ W/m}^2 \cdot ^\circ\text{C})(2.168 \text{ m}^2)} = 0.02345 \text{ }^\circ\text{C/W} \\
 R_{\text{total}} &= R_i + R_{\text{pipe}} + R_o = 0.00442 + 0.0000038 + 0.02345 = 0.02787 \text{ }^\circ\text{C/W}
 \end{aligned}$$

The rate of heat loss from the hot water pipe then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(70 - 10)^\circ\text{C}}{0.02787 \text{ }^\circ\text{C/W}} = \mathbf{2153 \text{ W}}$$

For a temperature drop of 3°C , the mass flow rate of water and the average velocity of water must be

$$\begin{aligned}
 \dot{Q} &= \dot{m} c_p \Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p \Delta T} = \frac{2153 \text{ J/s}}{(4180 \text{ J/kg}\cdot^\circ\text{C})(3^\circ\text{C})} = 0.172 \text{ kg/s} \\
 \dot{m} &= \rho V A_c \longrightarrow V = \frac{\dot{m}}{\rho A_c} = \frac{0.186 \text{ kg/s}}{(1000 \text{ kg/m}^3) \frac{\pi (0.04 \text{ m})^2}{4}} = \mathbf{0.137 \text{ m/s}}
 \end{aligned}$$

Discussion The outer surface temperature of the pipe is

$$\dot{Q} = \frac{T_{\infty 1} - T_s}{R_i + R_{\text{pipe}}} \rightarrow 2153 \text{ W} = \frac{(70 - T_s)^\circ\text{C}}{(0.00442 + 0.0000038)^\circ\text{C/W}} \rightarrow T_s = 60.5^\circ\text{C}$$

which is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Therefore, there is no need to repeat the calculations.

10-82E Steam exiting the turbine of a steam power plant at 100°F is to be condensed in a large condenser by cooling water flowing through copper tubes. For specified heat transfer coefficients, the length of the tube required to condense steam at a rate of 120 lbm/h is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal properties are constant. **4** Heat transfer coefficients are constant and uniform over the surfaces.

Properties The thermal conductivity of copper tube is given to be $k = 223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$. The heat of vaporization of water at 100°F is given to be 1037 Btu/lbm.

Analysis The individual resistances are

$$A_i = \pi D_i L = \pi (0.4 / 12 \text{ ft})(1 \text{ ft}) = 0.105 \text{ ft}^2$$

$$A_o = \pi D_o L = \pi (0.6 / 12 \text{ ft})(1 \text{ ft}) = 0.157 \text{ ft}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(35 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.105 \text{ ft}^2)} = 0.27211 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(3 / 2)}{2\pi (223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1 \text{ ft})} = 0.00029 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(1500 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.157 \text{ ft}^2)} = 0.00425 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{total}} = R_i + R_{\text{pipe}} + R_o = 0.27211 + 0.00029 + 0.00425 = 0.27665 \text{ h}\cdot^\circ\text{F/Btu}$$

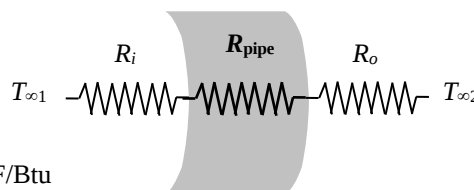
The heat transfer rate per ft length of the tube is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(100 - 70)^\circ\text{F}}{0.27665 \text{ h}\cdot^\circ\text{F/Btu}} = 108.44 \text{ Btu/h}$$

The total rate of heat transfer required to condense steam at a rate of 120 lbm/h and the length of the tube required is determined to be

$$\dot{Q}_{\text{total}} = \dot{m} h_{fg} = (120 \text{ lbm/h})(1037 \text{ Btu/lbm}) = 124,440 \text{ Btu/h}$$

$$\text{Tube length} = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{124,440}{108.44} = \mathbf{1148 \text{ ft}}$$



10-83E Steam exiting the turbine of a steam power plant at 100°F is to be condensed in a large condenser by cooling water flowing through copper tubes. For specified heat transfer coefficients and 0.01-in thick scale build up on the inner surface, the length of the tube required to condense steam at a rate of 120 lbm/h is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant. 4 Heat transfer coefficients are constant and uniform over the surfaces.

Properties The thermal conductivities are given to be $k = 223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for copper tube and be $k = 0.5 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for the mineral deposit. The heat of vaporization of water at 100°F is given to be 1037 Btu/lbm.

Analysis When a 0.01-in thick layer of deposit forms on the inner surface of the pipe, the inner diameter of the pipe will reduce from 0.4 in to 0.38 in. The individual thermal resistances are

$$T_{\infty 1} \quad R_i \quad R_{\text{deposit}} \quad R_{\text{pipe}} \quad R_o \quad T_{\infty 2}$$

$$A_i = \pi D_i L = \pi (0.38 / 12 \text{ ft})(1 \text{ ft}) = 0.099 \text{ ft}^2$$

$$A_o = \pi D_o L = \pi (0.6 / 12 \text{ ft})(1 \text{ ft}) = 0.157 \text{ ft}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(35 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.099 \text{ ft}^2)} = 0.2886 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(3 / 2)}{2\pi (223 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1 \text{ ft})} = 0.00029 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{deposit}} = \frac{\ln(r_1 / r_{\text{dep}})}{2\pi k_2 L} = \frac{\ln(0.2 / 0.19)}{2\pi (0.5 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1 \text{ ft})} = 0.01633 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(1500 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.157 \text{ ft}^2)} = 0.00425 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{total}} = R_i + R_{\text{pipe}} + R_{\text{deposit}} + R_o = 0.2886 + 0.00029 + 0.01633 + 0.00425 = 0.3095 \text{ h}\cdot^\circ\text{F/Btu}$$

The heat transfer rate per ft length of the tube is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(100 - 70)^\circ\text{F}}{0.3095 \text{ h}\cdot^\circ\text{F/Btu}} = 96.9 \text{ Btu/h}$$

The total rate of heat transfer required to condense steam at a rate of 120 lbm/h and the length of the tube required can be determined to be

$$\dot{Q}_{\text{total}} = \dot{m} h_{fg} = (120 \text{ lbm/h})(1037 \text{ Btu/lbm}) = 124,440 \text{ Btu/h}$$

$$\text{Tube length} = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{124,440}{96.9} = \mathbf{1284 \text{ ft}}$$

10-84E EES Prob. 10-82E is reconsidered. The effects of the thermal conductivity of the pipe material and the outer diameter of the pipe on the length of the tube required are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

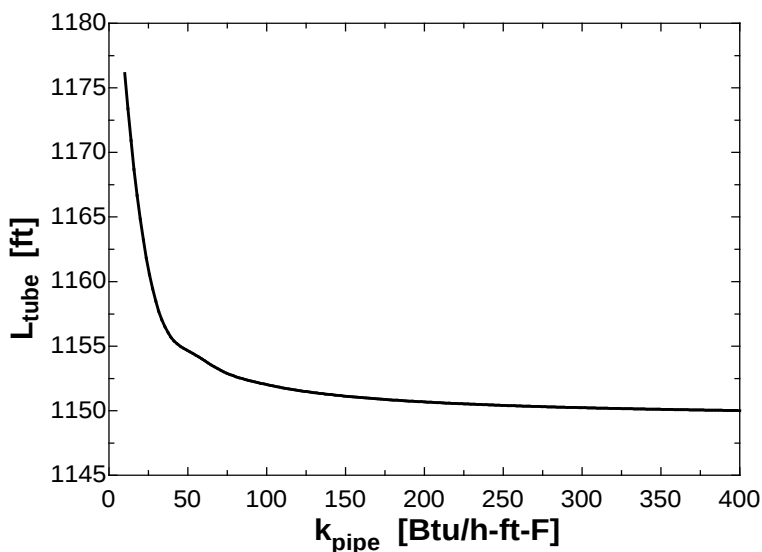
"GIVEN"

T_infinity_1=100 [F]
 T_infinity_2=70 [F]
 k_pipe=223 [Btu/h-ft-F]
 D_i=0.4 [in]
 D_o=0.6 [in]
 r_1=D_i/2
 r_2=D_o/2
 h_fg=1037 [Btu/lbm]
 h_o=1500 [Btu/h-ft^2-F]
 h_i=35 [Btu/h-ft^2-F]
 m_dot=120 [lbm/h]

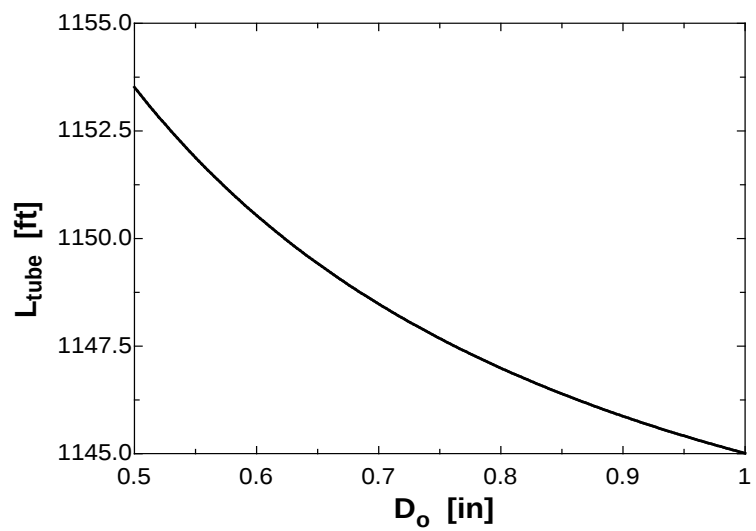
"ANALYSIS"

L=1 [ft] "for 1 ft length of the tube"
 A_i=pi*(D_i*Convert(in, ft))*L
 A_o=pi*(D_o*Convert(in, ft))*L
 R_conv_i=1/(h_i*A_i)
 R_pipe=ln(r_2/r_1)/(2*pi*k_pipe*L)
 R_conv_o=1/(h_o*A_o)
 R_total=R_conv_i+R_pipe+R_conv_o
 Q_dot=(T_infinity_1-T_infinity_2)/R_total
 Q_dot_total=m_dot*h_fg
 L_tube=Q_dot_total/Q_dot

k _{pipe} [Btu/h.ft.F]	L _{tube} [ft]
10	1176
30.53	1158
51.05	1155
71.58	1153
92.11	1152
112.6	1152
133.2	1151
153.7	1151
174.2	1151
194.7	1151
215.3	1151
235.8	1150
256.3	1150
276.8	1150
297.4	1150
317.9	1150
338.4	1150
358.9	1150
379.5	1150
400	1150



D_o[in]	L_{tube} [ft]
0.5	1154
0.525	1153
0.55	1152
0.575	1151
0.6	1151
0.625	1150
0.65	1149
0.675	1149
0.7	1148
0.725	1148
0.75	1148
0.775	1147
0.8	1147
0.825	1147
0.85	1146
0.875	1146
0.9	1146
0.925	1146
0.95	1145
0.975	1145
1	1145



10-85 A spherical tank filled with liquid nitrogen at 1 atm and -196°C is exposed to convection and radiation with the surrounding air and surfaces. The rate of evaporation of liquid nitrogen in the tank as a result of the heat gain from the surroundings for the cases of no insulation, 5-cm thick fiberglass insulation, and 2-cm thick superinsulation are to be determined.

Assumptions 1 Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. 3 The combined heat transfer coefficient is constant and uniform over the entire surface. 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the nitrogen inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The heat of vaporization and density of liquid nitrogen at 1 atm are given to be 198 kJ/kg and 810 kg/m^3 , respectively. The thermal conductivities are given to be $k = 0.035\text{ W/m}\cdot^{\circ}\text{C}$ for fiberglass insulation and $k = 0.00005\text{ W/m}\cdot^{\circ}\text{C}$ for super insulation.

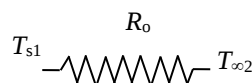
Analysis (a) The heat transfer rate and the rate of evaporation of the liquid without insulation are

$$A = \pi D^2 = \pi (3\text{ m})^2 = 28.27\text{ m}^2$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2\cdot^{\circ}\text{C})(28.27\text{ m}^2)} = 0.00101^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_o} = \frac{[15 - (-196)]^{\circ}\text{C}}{0.00101^{\circ}\text{C/W}} = 208,910\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{208.910\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{1.055\text{ kg/s}}$$



(b) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are

$$A = \pi D^2 = \pi (3.1\text{ m})^2 = 30.19\text{ m}^2$$

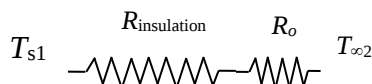
$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2\cdot^{\circ}\text{C})(30.19\text{ m}^2)} = 0.000946^{\circ}\text{C/W}$$

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.55 - 1.5)\text{ m}}{4\pi (0.035\text{ W/m}\cdot^{\circ}\text{C})(1.55\text{ m})(1.5\text{ m})} = 0.0489^{\circ}\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} = 0.000946 + 0.0489 = 0.0498^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[15 - (-196)]^{\circ}\text{C}}{0.0498^{\circ}\text{C/W}} = 4233\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{4.233\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{0.0214\text{ kg/s}}$$



(c) The heat transfer rate and the rate of evaporation of the liquid with 2-cm thick layer of superinsulation is

$$A = \pi D^2 = \pi (3.04\text{ m})^2 = 29.03\text{ m}^2$$

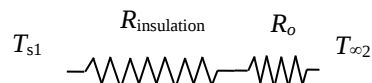
$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2\cdot^{\circ}\text{C})(29.03\text{ m}^2)} = 0.000984^{\circ}\text{C/W}$$

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.52 - 1.5)\text{ m}}{4\pi (0.00005\text{ W/m}\cdot^{\circ}\text{C})(1.52\text{ m})(1.5\text{ m})} = 13.96^{\circ}\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} = 0.000984 + 13.96 = 13.96^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[15 - (-196)]^{\circ}\text{C}}{13.96^{\circ}\text{C/W}} = 15.11\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{0.01511\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{0.000076\text{ kg/s}}$$



10-86 A spherical tank filled with liquid oxygen at 1 atm and -183°C is exposed to convection and radiation with the surrounding air and surfaces. The rate of evaporation of liquid oxygen in the tank as a result of the heat gain from the surroundings for the cases of no insulation, 5-cm thick fiberglass insulation, and 2-cm thick superinsulation are to be determined.

Assumptions 1 Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. 3 The combined heat transfer coefficient is constant and uniform over the entire surface. 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the oxygen inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The heat of vaporization and density of liquid oxygen at 1 atm are given to be 213 kJ/kg and 1140 kg/m^3 , respectively. The thermal conductivities are given to be $k = 0.035\text{ W/m}\cdot^{\circ}\text{C}$ for fiberglass insulation and $k = 0.00005\text{ W/m}\cdot^{\circ}\text{C}$ for super insulation.

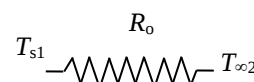
Analysis (a) The heat transfer rate and the rate of evaporation of the liquid without insulation are

$$A = \pi D^2 = \pi (3\text{ m})^2 = 28.27\text{ m}^2$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2\cdot^{\circ}\text{C})(28.27\text{ m}^2)} = 0.00101^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_o} = \frac{[15 - (-183)]^{\circ}\text{C}}{0.00101^{\circ}\text{C/W}} = 196,040\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{196.040\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{0.920\text{ kg/s}}$$



(b) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are

$$A = \pi D^2 = \pi (3.1\text{ m})^2 = 30.19\text{ m}^2$$

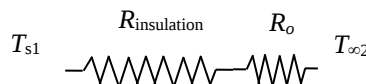
$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2\cdot^{\circ}\text{C})(30.19\text{ m}^2)} = 0.000946^{\circ}\text{C/W}$$

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.55 - 1.5)\text{ m}}{4\pi (0.035\text{ W/m}\cdot^{\circ}\text{C})(1.55\text{ m})(1.5\text{ m})} = 0.0489^{\circ}\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} = 0.000946 + 0.0489 = 0.0498^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[15 - (-183)]^{\circ}\text{C}}{0.0498^{\circ}\text{C/W}} = 3976\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{3.976\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{0.0187\text{ kg/s}}$$



(c) The heat transfer rate and the rate of evaporation of the liquid with a 2-cm superinsulation is

$$A = \pi D^2 = \pi (3.04\text{ m})^2 = 29.03\text{ m}^2$$

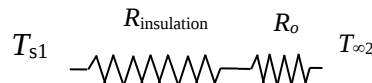
$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2\cdot^{\circ}\text{C})(29.03\text{ m}^2)} = 0.000984^{\circ}\text{C/W}$$

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.52 - 1.5)\text{ m}}{4\pi (0.00005\text{ W/m}\cdot^{\circ}\text{C})(1.52\text{ m})(1.5\text{ m})} = 13.96^{\circ}\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} = 0.000984 + 13.96 = 13.96^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[15 - (-183)]^{\circ}\text{C}}{13.96^{\circ}\text{C/W}} = 14.18\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{0.01418\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{0.000067\text{ kg/s}}$$



Critical Radius of Insulation

10-87C In a cylindrical pipe or a spherical shell, the additional insulation increases the conduction resistance of insulation, but decreases the convection resistance of the surface because of the increase in the outer surface area. Due to these opposite effects, a critical radius of insulation is defined as the outer radius that provides maximum rate of heat transfer. For a cylindrical layer, it is defined as $r_{cr} = k/h$ where k is the thermal conductivity of insulation and h is the external convection heat transfer coefficient.

10-88C It will decrease.

10-89C Yes, the measurements can be right. If the radius of insulation is less than critical radius of insulation of the pipe, the rate of heat loss will increase.

10-90C No.

10-91C For a cylindrical pipe, the critical radius of insulation is defined as $r_{cr} = k/h$. On windy days, the external convection heat transfer coefficient is greater compared to calm days. Therefore critical radius of insulation will be greater on calm days.

10-92 An electric wire is tightly wrapped with a 1-mm thick plastic cover. The interface temperature and the effect of doubling the thickness of the plastic cover on the interface temperature are to be determined.

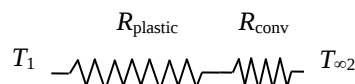
Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant. 4 The thermal contact resistance at the interface is negligible. 5 Heat transfer coefficient accounts for the radiation effects, if any.

Properties The thermal conductivity of plastic cover is given to be $k = 0.15 \text{ W/m}\cdot^\circ\text{C}$.

Analysis In steady operation, the rate of heat transfer from the wire is equal to the heat generated within the wire,

$$\dot{Q} = \dot{W}_e = \mathbf{VI} = (8 \text{ V})(13 \text{ A}) = 104 \text{ W}$$

The total thermal resistance is



$$R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(24 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.0042 \text{ m})(10 \text{ m})]} = 0.3158 \text{ }^\circ\text{C/W}$$

$$R_{\text{plastic}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(2.1/1.1)}{2\pi(0.15 \text{ W/m}\cdot^\circ\text{C})(10 \text{ m})} = 0.0686 \text{ }^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{plastic}} = 0.3158 + 0.0686 = 0.3844 \text{ }^\circ\text{C/W}$$

Then the interface temperature becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty} + \dot{Q}R_{\text{total}} = 30^\circ\text{C} + (104 \text{ W})(0.3844 \text{ }^\circ\text{C/W}) = \mathbf{70.0^\circ\text{C}}$$

The critical radius of plastic insulation is

$$r_{cr} = \frac{k}{h} = \frac{0.15 \text{ W/m}\cdot^\circ\text{C}}{24 \text{ W/m}^2 \cdot ^\circ\text{C}} = 0.00625 \text{ m} = 6.25 \text{ mm}$$

Doubling the thickness of the plastic cover will increase the outer radius of the wire to 3 mm, which is less than the critical radius of insulation. Therefore, doubling the thickness of plastic cover will increase the rate of heat loss and decrease the interface temperature.

10-93E An electrical wire is covered with 0.02-in thick plastic insulation. It is to be determined if the plastic insulation on the wire will increase or decrease heat transfer from the wire.

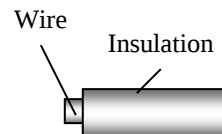
Assumptions 1 Heat transfer from the wire is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant. 4 The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity of plastic cover is given to be $k = 0.075 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$.

Analysis The critical radius of plastic insulation is

$$r_{cr} = \frac{k}{h} = \frac{0.075 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{2.5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}} = 0.03 \text{ ft} = 0.36 \text{ in} > r_2 (= 0.0615 \text{ in})$$

Since the outer radius of the wire with insulation is smaller than critical radius of insulation, plastic insulation will **increase** heat transfer from the wire.



10-94E An electrical wire is covered with 0.02-in thick plastic insulation. By considering the effect of thermal contact resistance, it is to be determined if the plastic insulation on the wire will increase or decrease heat transfer from the wire.

Assumptions 1 Heat transfer from the wire is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant

Properties The thermal conductivity of plastic cover is given to be $k = 0.075 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$.

Analysis Without insulation, the total thermal resistance is (per ft length of the wire)



$$R_{tot} = R_{conv} = \frac{1}{h_o A_o} = \frac{1}{(2.5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})[\pi(0.083/12 \text{ ft})(1 \text{ ft})]} = 18.4 \text{ h}\cdot^\circ\text{F/Btu}$$

With insulation, the total thermal resistance is

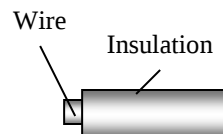
$$R_{conv} = \frac{1}{h_o A_o} = \frac{1}{(2.5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})[\pi(0.123/12 \text{ ft})(1 \text{ ft})]} = 12.42 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{plastic} = \frac{\ln(r_2/r_1)}{2\pi k L} = \frac{\ln(0.123/0.083)}{2\pi(0.075 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1 \text{ ft})} = 0.835 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{interface} = \frac{h_c}{A_c} = \frac{0.001 \text{ h}\cdot\text{ft}^2\cdot^\circ\text{F/Btu}}{[\pi(0.083/12 \text{ ft})(1 \text{ ft})]} = 0.046 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{total} = R_{conv} + R_{plastic} + R_{interface} = 12.42 + 0.835 + 0.046 = 13.30 \text{ h}\cdot^\circ\text{F/Btu}$$

Since the total thermal resistance decreases after insulation, plastic insulation **will increase** heat transfer from the wire. The thermal contact resistance appears to have negligible effect in this case.



10-95 A spherical ball is covered with 1-mm thick plastic insulation. It is to be determined if the plastic insulation on the ball will increase or decrease heat transfer from it.

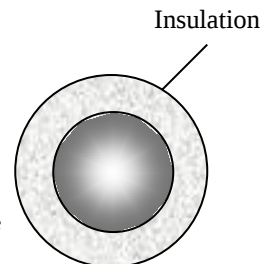
Assumptions **1** Heat transfer from the ball is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Thermal properties are constant. **4** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity of plastic cover is given to be $k = 0.13 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The critical radius of plastic insulation for the spherical ball is

$$r_{cr} = \frac{2k}{h} = \frac{2(0.13 \text{ W/m}\cdot^\circ\text{C})}{20 \text{ W/m}^2\cdot^\circ\text{C}} = 0.013 \text{ m} = 13 \text{ mm} > r_2 (= 7 \text{ mm})$$

Since the outer radius of the ball with insulation is smaller than critical radius of insulation, plastic insulation will **increase** heat transfer from the wire.



10-96 EES Prob. 10-95 is reconsidered. The rate of heat transfer from the ball as a function of the plastic insulation thickness is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

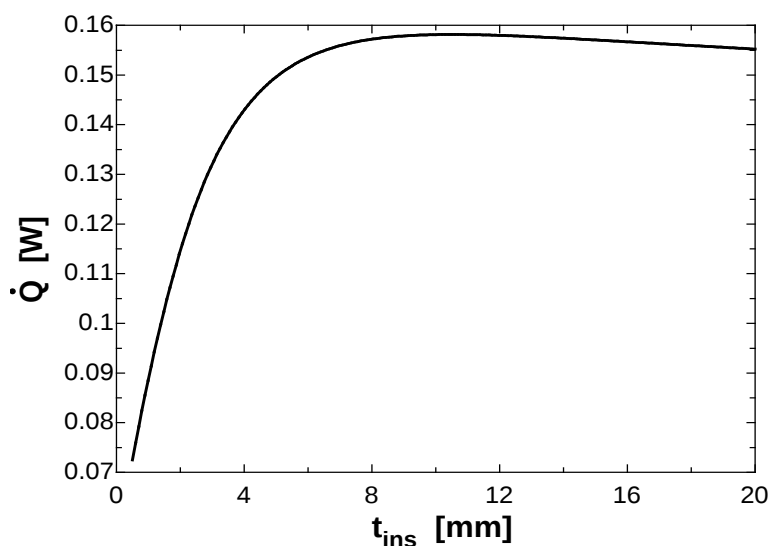
"GIVEN"

D_1=0.005 [m]
t_ins=1 [mm]
k_ins=0.13 [W/m-C]
T_ball=50 [C]
T_infinity=15 [C]
h_o=20 [W/m^2-C]

"ANALYSIS"

D_2=D_1+2*t_ins*Convert(mm, m)
A_o=pi*D_2^2
R_conv_o=1/(h_o*A_o)
R_ins=(r_2-r_1)/(4*pi*r_1*r_2*k_ins)
r_1=D_1/2
r_2=D_2/2
R_total=R_conv_o+R_ins
Q_dot=(T_ball-T_infinity)/R_total

t _{ins} [mm]	Q [W]
0.5	0.07248
1.526	0.1035
2.553	0.1252
3.579	0.139
4.605	0.1474
5.632	0.1523
6.658	0.1552
7.684	0.1569
8.711	0.1577
9.737	0.1581
10.76	0.1581
11.79	0.158
12.82	0.1578
13.84	0.1574
14.87	0.1571
15.89	0.1567
16.92	0.1563
17.95	0.1559
18.97	0.1556
20	0.1552



Heat Transfer from Finned Surfaces

10-97C Increasing the rate of heat transfer from a surface by increasing the heat transfer surface area.

10-98C The fin efficiency is defined as the ratio of actual heat transfer rate from the fin to the ideal heat transfer rate from the fin if the entire fin were at base temperature, and its value is between 0 and 1. Fin effectiveness is defined as the ratio of heat transfer rate from a finned surface to the heat transfer rate from the same surface if there were no fins, and its value is expected to be greater than 1.

10-99C Heat transfer rate will decrease since a fin effectiveness smaller than 1 indicates that the fin acts as insulation.

10-100C Fins enhance heat transfer from a surface by increasing heat transfer surface area for convection heat transfer. However, adding too many fins on a surface can suffocate the fluid and retard convection, and thus it may cause the overall heat transfer coefficient and heat transfer to decrease.

10-101C Effectiveness of a single fin is the ratio of the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the fin base area. The overall effectiveness of a finned surface is defined as the ratio of the total heat transfer from the finned surface to the heat transfer from the same surface if there were no fins.

10-102C Fins should be attached on the air side since the convection heat transfer coefficient is lower on the air side than it is on the water side.

10-103C Fins should be attached to the outside since the heat transfer coefficient inside the tube will be higher due to forced convection. Fins should be added to both sides of the tubes when the convection coefficients at the inner and outer surfaces are comparable in magnitude.

10-104C Welding or tight fitting introduces thermal contact resistance at the interface, and thus retards heat transfer. Therefore, the fins formed by casting or extrusion will provide greater enhancement in heat transfer.

10-105C If the fin is too long, the temperature of the fin tip will approach the surrounding temperature and we can neglect heat transfer from the fin tip. Also, if the surface area of the fin tip is very small compared to the total surface area of the fin, heat transfer from the tip can again be neglected.

10-106C Increasing the length of a fin decreases its efficiency but increases its effectiveness.

10-107C Increasing the diameter of a fin increases its efficiency but decreases its effectiveness.

10-108C The thicker fin has higher efficiency; the thinner one has higher effectiveness.

10-109C The fin with the lower heat transfer coefficient has the higher efficiency and the higher effectiveness.

10-110 A relation is to be obtained for the fin efficiency for a fin of constant cross-sectional area A_c , perimeter p , length L , and thermal conductivity k exposed to convection to a medium at T_∞ with a heat transfer coefficient h . The relation is to be simplified for circular fin of diameter D and for a rectangular fin of thickness t .

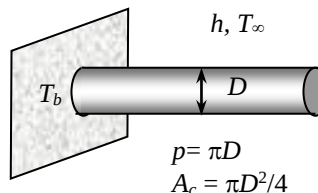
Assumptions 1 The fins are sufficiently long so that the temperature of the fin at the tip is nearly T_∞ . 2 Heat transfer from the fin tips is negligible.

Analysis Taking the temperature of the fin at the base to be T_b and using the heat transfer relation for a long fin, fin efficiency for long fins can be expressed as

$$\eta_{\text{fin}} = \frac{\text{Actual heat transfer rate from the fin}}{\text{Ideal heat transfer rate from the fin}}$$

if the entire fin were at base temperature

$$= \frac{\sqrt{hpkA_c}(T_b - T_\infty)}{hA_{\text{fin}}(T_b - T_\infty)} = \frac{\sqrt{hpkA_c}}{hpL} = \frac{1}{L} \sqrt{\frac{kA_c}{ph}}$$



This relation can be simplified for a circular fin of diameter D and rectangular fin of thickness t and width w to be

$$\eta_{\text{fin,circular}} = \frac{1}{L} \sqrt{\frac{kA_c}{ph}} = \frac{1}{L} \sqrt{\frac{k(\pi D^2/4)}{(\pi D)h}} = \frac{1}{2L} \sqrt{\frac{kD}{h}}$$

$$\eta_{\text{fin,rectangular}} = \frac{1}{L} \sqrt{\frac{kA_c}{ph}} = \frac{1}{L} \sqrt{\frac{k(wt)}{2(w+t)h}} \cong \frac{1}{L} \sqrt{\frac{k(wt)}{2wh}} = \frac{1}{L} \sqrt{\frac{kt}{2h}}$$

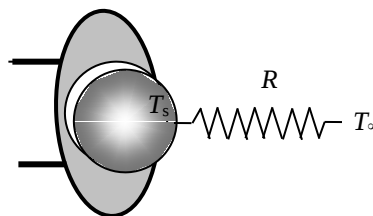
10-111 The maximum power rating of a transistor whose case temperature is not to exceed 80°C is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The transistor case is isothermal at 80°C .

Properties The case-to-ambient thermal resistance is given to be $20^\circ\text{C}/\text{W}$.

Analysis The maximum power at which this transistor can be operated safely is

$$\dot{Q} = \frac{\Delta T}{R_{\text{case-ambient}}} = \frac{T_{\text{case}} - T_\infty}{R_{\text{case-ambient}}} = \frac{(80 - 40)^\circ\text{C}}{25^\circ\text{C}/\text{W}} = \mathbf{1.6\text{ W}}$$



10-112 A fin is attached to a surface. The percent error in the rate of heat transfer from the fin when the infinitely long fin assumption is used instead of the adiabatic fin tip assumption is to be determined.

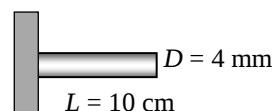
Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 The heat transfer coefficient is constant and uniform over the entire fin surface. 4 The thermal properties of the fins are constant. 5 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum fin is given to be $k = 237 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The expressions for the heat transfer from a fin under infinitely long fin and adiabatic fin tip assumptions are

$$\dot{Q}_{\text{long fin}} = \sqrt{hp k A_c} (T_b - T_\infty)$$

$$\dot{Q}_{\text{ins. tip}} = \sqrt{hp k A_c} (T_b - T_\infty) \tanh(mL)$$



The percent error in using long fin assumption can be expressed as

$$\% \text{ Error} = \frac{\dot{Q}_{\text{long fin}} - \dot{Q}_{\text{ins. tip}}}{\dot{Q}_{\text{ins. tip}}} = \frac{\sqrt{hp k A_c} (T_b - T_\infty) - \sqrt{hp k A_c} (T_b - T_\infty) \tanh(mL)}{\sqrt{hp k A_c} (T_b - T_\infty) \tanh(mL)} = \frac{1}{\tanh(mL)} - 1$$

$$\text{where } m = \sqrt{\frac{hp}{k A_c}} = \sqrt{\frac{(12 \text{ W/m}^2 \cdot ^\circ\text{C}) \pi (0.004 \text{ m})}{(237 \text{ W/m}\cdot^\circ\text{C}) \pi (0.004 \text{ m})^2 / 4}} = 7.116 \text{ m}^{-1}$$

Substituting,

$$\% \text{ Error} = \frac{1}{\tanh(mL)} - 1 = \frac{1}{\tanh[(7.116 \text{ m}^{-1})(0.10 \text{ m})]} - 1 = 0.635 = \mathbf{63.5\%}$$

This result shows that using infinitely long fin assumption may yield results grossly in error.

10-113 A very long fin is attached to a flat surface. The fin temperature at a certain distance from the base and the rate of heat loss from the entire fin are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 The heat transfer coefficient is constant and uniform over the entire fin surface. 4 The thermal properties of the fins are constant. 5 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the fin is given to be $k = 200 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The fin temperature at a distance of 5 cm from the base is determined from

$$m = \sqrt{\frac{hp}{k A_c}} = \sqrt{\frac{(20 \text{ W/m}^2 \cdot ^\circ\text{C})(2 \times 0.05 + 2 \times 0.001) \text{ m}}{(200 \text{ W/m}\cdot^\circ\text{C})(0.05 \times 0.001) \text{ m}^2}} = 14.3 \text{ m}^{-1}$$

$$\frac{T - T_\infty}{T_b - T_\infty} = e^{-mx} \longrightarrow \frac{T - 20}{40 - 20} = e^{-(14.3)(0.05)} \longrightarrow T = \mathbf{29.8^\circ\text{C}}$$



The rate of heat loss from this very long fin is

$$\begin{aligned} \dot{Q}_{\text{long fin}} &= \sqrt{hp k A_c} (T_b - T_\infty) \\ &= \sqrt{(20)(2 \times 0.05 + 2 \times 0.001)(200)(0.05 \times 0.001)} (40 - 20) \\ &= \mathbf{2.9 \text{ W}} \end{aligned}$$

10-114 Circular fins made of copper are considered. The function $\theta(x) = T(x) - T_\infty$ along a fin is to be expressed and the temperature at the middle is to be determined. Also, the rate of heat transfer from each fin, the fin effectiveness, and the total rate of heat transfer from the wall are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 The heat transfer coefficient is constant and uniform over the entire finned and unfinned wall surfaces. 4 The thermal properties of the fins are constant. 5 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the copper fin is given to be $k = 400 \text{ W/m}\cdot^\circ\text{C}$.

Analysis (a) For fin with prescribed tip temperature,

$$\frac{\theta}{\theta_b} = \frac{\theta_L / \theta_b \sinh(mx) + \sinh[m(L-x)]}{\sinh(mL)}$$

With $\theta_b = T_b - T_\infty = T_{s1}$ and $\theta_L = T_L - T_\infty = 0$, the equation becomes

$$\frac{\theta}{\theta_b} = \frac{\sinh[m(L-x)]}{\sinh(mL)} = \frac{\exp[m(L-x)] - \exp[-m(L-x)]}{\exp(mL) - \exp(-mL)}$$

For $x = L/2$:

$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{(100)\pi(0.001)}{(400)\pi(0.001)^2/4}} = 31.6 \text{ m}^{-1}$$

$$\begin{aligned} \theta_{L/2} = T_{L/2} - T_\infty &= \theta_b \frac{\sinh(mL/2)}{\sinh(mL)} = T_{s1} \frac{\exp(mL/2) - \exp(-mL/2)}{\exp(mL) - \exp(-mL)} \\ &= (132) \frac{\exp(31.6 \times 0.0254/2) - \exp(-31.6 \times 0.0254/2)}{\exp(31.6 \times 0.0254) - \exp(-31.6 \times 0.0254)} = 61.6^\circ\text{C} \end{aligned}$$

(b) The rate of heat transfer from a single fin is

$$\begin{aligned} \dot{q}_{\text{one fin}} &= \theta_b \sqrt{hpkA_c} \frac{\cosh(mL)}{\sinh(mL)} \\ &= (132 - 0) \sqrt{(100)\pi(0.001)(400)\pi(0.001)^2/4} \frac{\cosh(31.6 \times 0.0254)}{\sinh(31.6 \times 0.0254)} \\ &= 1.97 \text{ W} \end{aligned}$$

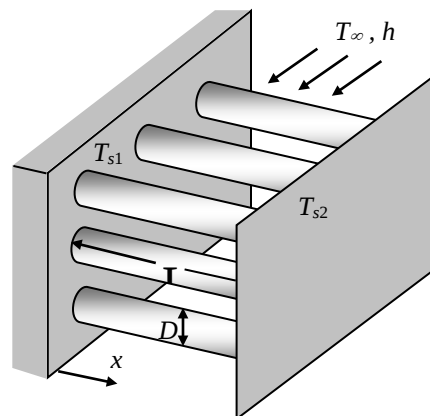
The effectiveness of the fin is

$$\varepsilon = \frac{q_f}{A_c h \theta_b} = \frac{1.97}{0.25\pi(0.001)^2 (100)(132 - 0)} = 190$$

Since $\varepsilon \gg 2$, the fins are well justified.

(c) The total rate of heat transfer is

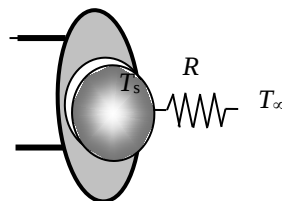
$$\begin{aligned} \dot{q}_{\text{total}} &= \dot{q}_{\text{fins}} + \dot{q}_{\text{base}} \\ &= n_{\text{fin}} \dot{q}_{\text{one fin}} + (A_{\text{wall}} - n_{\text{fin}} A_c) h \theta_b \\ &= (625)(1.97) + [0.1 \times 0.1 - 625 \times 0.25\pi(0.001)^2](100)(132) \\ &= \mathbf{1363 \text{ W}} \end{aligned}$$



10-115 A commercially available heat sink is to be selected to keep the case temperature of a transistor below 90°C in an environment at 20°C.

Assumptions 1 Steady operating conditions exist. 2 The transistor case is isothermal at 90°C. 3 The contact resistance between the transistor and the heat sink is negligible.

Analysis The thermal resistance between the transistor attached to the sink and the ambient air is determined to be



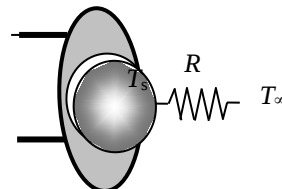
$$\dot{Q} = \frac{\Delta T}{R_{\text{case-ambient}}} \longrightarrow R_{\text{case-ambient}} = \frac{T_{\text{transistor}} - T_{\infty}}{\dot{Q}} = \frac{(90 - 20)^{\circ}\text{C}}{40 \text{ W}} = 1.75^{\circ}\text{C/W}$$

The thermal resistance of the heat sink must be below 1.75°C/W. Table 10-6 reveals that HS6071 in vertical position, HS5030 and HS6115 in both horizontal and vertical position can be selected.

10-116 A commercially available heat sink is to be selected to keep the case temperature of a transistor below 55°C in an environment at 18°C.

Assumptions 1 Steady operating conditions exist. 2 The transistor case is isothermal at 55°C. 3 The contact resistance between the transistor and the heat sink is negligible.

Analysis The thermal resistance between the transistor attached to the sink and the ambient air is determined to be



$$\dot{Q} = \frac{\Delta T}{R_{\text{case-ambient}}} \longrightarrow R_{\text{case-ambient}} = \frac{T_{\text{transistor}} - T_{\infty}}{\dot{Q}} = \frac{(55 - 18)^{\circ}\text{C}}{25 \text{ W}} = 1.5^{\circ}\text{C/W}$$

The thermal resistance of the heat sink must be below 1.5°C/W. Table 10-6 reveals that HS5030 in both horizontal and vertical positions, HS6071 in vertical position, and HS6115 in both horizontal and vertical positions can be selected.

10-117 Circular aluminum fins are to be attached to the tubes of a heating system. The increase in heat transfer from the tubes per unit length as a result of adding fins is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat transfer coefficient is constant and uniform over the entire fin surfaces. 3 Thermal conductivity is constant. 4 Heat transfer by radiation is negligible.

Properties The thermal conductivity of the fins is given to be $k = 186 \text{ W/m} \cdot ^\circ\text{C}$.

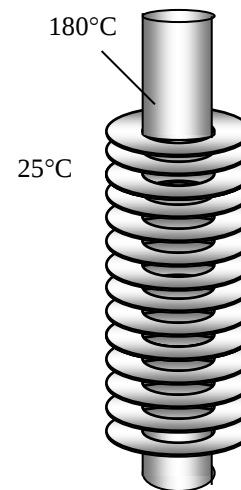
Analysis In case of no fins, heat transfer from the tube per meter of its length is

$$A_{\text{no fin}} = \pi D_1 L = \pi (0.05 \text{ m})(1 \text{ m}) = 0.1571 \text{ m}^2$$

$$\dot{Q}_{\text{no fin}} = h A_{\text{no fin}} (T_b - T_\infty) = (40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1571 \text{ m}^2)(180 - 25)^\circ\text{C} = 974 \text{ W}$$

The efficiency of these circular fins is, from the efficiency curve, Fig. 10-43

$$\left. \begin{aligned} L &= (D_2 - D_1) / 2 = (0.06 - 0.05) / 2 = 0.005 \text{ m} \\ \frac{r_2 + (t/2)}{r_1} &= \frac{0.03 + (0.001/2)}{0.025} = 1.22 \\ L_c^{3/2} \left(\frac{h}{kA_p} \right)^{1/2} &= \left(L + \frac{t}{2} \right) \sqrt{\frac{h}{kt}} \\ &= \left(0.005 + \frac{0.001}{2} \right) \sqrt{\frac{40 \text{ W/m}^2 \cdot ^\circ\text{C}}{(186 \text{ W/m} \cdot ^\circ\text{C})(0.001 \text{ m})}} = 0.08 \end{aligned} \right\} \eta_{\text{fin}} = 0.97$$



Heat transfer from a single fin is

$$A_{\text{fin}} = 2\pi(r_2^2 - r_1^2) + 2\pi r_2 t = 2\pi(0.03^2 - 0.025^2) + 2\pi(0.03)(0.001) = 0.001916 \text{ m}^2$$

$$\dot{Q}_{\text{fin}} = \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}} (T_b - T_\infty)$$

$$= 0.97(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.001916 \text{ m}^2)(180 - 25)^\circ\text{C}$$

$$= 11.53 \text{ W}$$

Heat transfer from a single unfinned portion of the tube is

$$A_{\text{unfin}} = \pi D_1 s = \pi(0.05 \text{ m})(0.003 \text{ m}) = 0.0004712 \text{ m}^2$$

$$\dot{Q}_{\text{unfin}} = h A_{\text{unfin}} (T_b - T_\infty) = (40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0004712 \text{ m}^2)(180 - 25)^\circ\text{C} = 2.92 \text{ W}$$

There are 250 fins and thus 250 interfin spacings per meter length of the tube. The total heat transfer from the finned tube is then determined from

$$\dot{Q}_{\text{total,fin}} = n(\dot{Q}_{\text{fin}} + \dot{Q}_{\text{unfin}}) = 250(11.53 + 2.92) = 3613 \text{ W}$$

Therefore the increase in heat transfer from the tube per meter of its length as a result of the addition of the fins is

$$\dot{Q}_{\text{increase}} = \dot{Q}_{\text{total,fin}} - \dot{Q}_{\text{no fin}} = 3613 - 974 = 2639 \text{ W}$$

10-118E The handle of a stainless steel spoon partially immersed in boiling water extends 7 in. in the air from the free surface of the water. The temperature difference across the exposed surface of the spoon handle is to be determined.

Assumptions **1** The temperature of the submerged portion of the spoon is equal to the water temperature. **2** The temperature in the spoon varies in the axial direction only (along the spoon), $T(x)$. **3** The heat transfer from the tip of the spoon is negligible. **4** The heat transfer coefficient is constant and uniform over the entire spoon surface. **5** The thermal properties of the spoon are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the spoon.

Properties The thermal conductivity of the spoon is given to be $k = 8.7 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$.

Analysis Noting that the cross-sectional area of the spoon is constant and measuring x from the free surface of water, the variation of temperature along the spoon can be expressed as

$$\frac{T(x) - T_\infty}{T_b - T_\infty} = \frac{\cosh m(L - x)}{\cosh mL}$$

where

$$p = 2(0.5/12 \text{ ft} + 0.08/12 \text{ ft}) = 0.0967 \text{ ft}$$

$$A_c = (0.5/12 \text{ ft})(0.08/12 \text{ ft}) = 0.000278 \text{ ft}^2$$

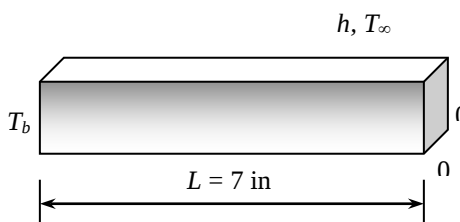
$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{(3 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.0967 \text{ ft})}{(8.7 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.000278 \text{ ft}^2)}} = 10.95 \text{ ft}^{-1}$$

Noting that $x = L = 7/12 = 0.583 \text{ ft}$ at the tip and substituting, the tip temperature of the spoon is determined to be

$$\begin{aligned} T(L) &= T_\infty + (T_b - T_\infty) \frac{\cosh m(L - L)}{\cosh mL} \\ &= 75^\circ\text{F} + (200 - 75) \frac{\cosh 0}{\cosh(10.95 \times 0.583)} = 75^\circ\text{F} + (200 - 75) \frac{1}{296} = 75.4^\circ\text{F} \end{aligned}$$

Therefore, the temperature difference across the exposed section of the spoon handle is

$$\Delta T = T_b - T_{\text{tip}} = (200 - 75.4)^\circ\text{F} = \mathbf{124.6^\circ\text{F}}$$



10-119E The handle of a silver spoon partially immersed in boiling water extends 7 in. in the air from the free surface of the water. The temperature difference across the exposed surface of the spoon handle is to be determined.

Assumptions **1** The temperature of the submerged portion of the spoon is equal to the water temperature. **2** The temperature in the spoon varies in the axial direction only (along the spoon), $T(x)$. **3** The heat transfer from the tip of the spoon is negligible. **4** The heat transfer coefficient is constant and uniform over the entire spoon surface. **5** The thermal properties of the spoon are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the spoon..

Properties The thermal conductivity of the spoon is given to be $k = 247 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$.

Analysis Noting that the cross-sectional area of the spoon is constant and measuring x from the free surface of water, the variation of temperature along the spoon can be expressed as

$$\frac{T(x) - T_\infty}{T_b - T_\infty} = \frac{\cosh m(L - x)}{\cosh mL}$$

where

$$p = 2(0.5/12 \text{ ft} + 0.08/12 \text{ ft}) = 0.0967 \text{ ft}$$

$$A_c = (0.5/12 \text{ ft})(0.08/12 \text{ ft}) = 0.000278 \text{ ft}^2$$

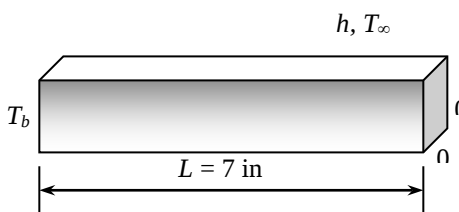
$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{(3 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.0967 \text{ ft})}{(247 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(0.000278 \text{ ft}^2)}} = 2.055 \text{ ft}^{-1}$$

Noting that $x = L = 0.7/12 = 0.583 \text{ ft}$ at the tip and substituting, the tip temperature of the spoon is determined to be

$$\begin{aligned} T(L) &= T_\infty + (T_b - T_\infty) \frac{\cosh m(L - L)}{\cosh mL} \\ &= 75^\circ\text{F} + (200 - 75) \frac{\cosh 0}{\cosh(2.055 \times 0.583)} = 75^\circ\text{F} + (200 - 75) \frac{1}{1.81} = \mathbf{144.1^\circ\text{F}} \end{aligned}$$

Therefore, the temperature difference across the exposed section of the spoon handle is

$$\Delta T = T_b - T_{\text{tip}} = (200 - 144.1)^\circ\text{C} = \mathbf{55.9^\circ\text{F}}$$



10-120E EES Prob. 10-118E is reconsidered. The effects of the thermal conductivity of the spoon material and the length of its extension in the air on the temperature difference across the exposed surface of the spoon handle are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

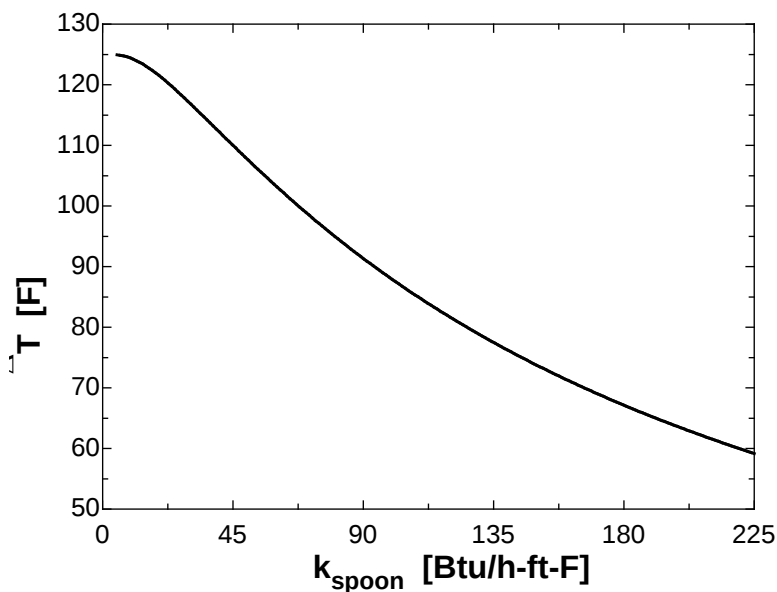
"GIVEN"

$k_{\text{spoon}} = 8.7 \text{ [Btu/h-ft-F]}$
 $T_w = 200 \text{ [F]}$
 $T_{\text{infinity}} = 75 \text{ [F]}$
 $A_c = (0.08/12 * 0.5/12) \text{ [ft}^2\text{]}$
 $L = 7 \text{ [in]}$
 $h = 3 \text{ [Btu/h-ft}^2\text{-F]}$

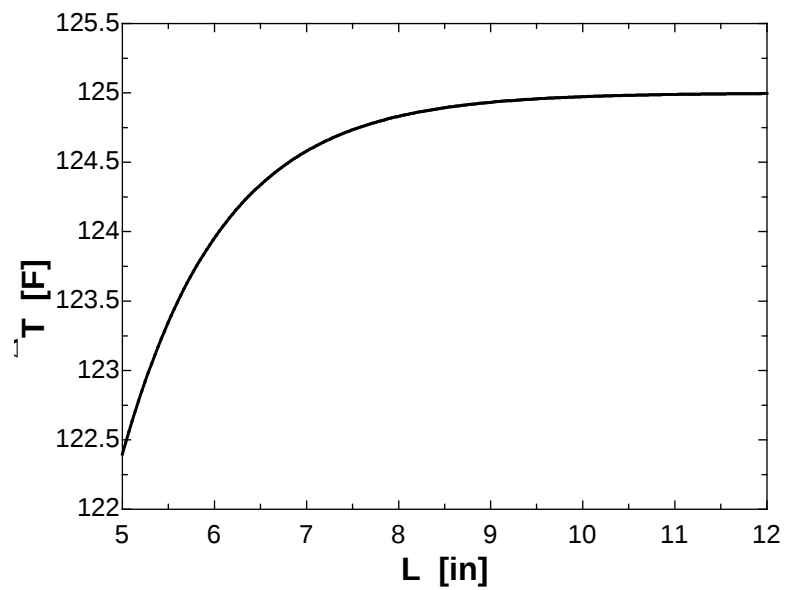
"ANALYSIS"

$p = 2 * (0.08/12 + 0.5/12)$
 $a = \sqrt{(h * p) / (k_{\text{spoon}} * A_c)}$
 $(T_{\text{tip}} - T_{\text{infinity}}) / (T_w - T_{\text{infinity}}) = \cosh(a * (L - x) * \text{Convert(in, ft)}) / \cosh(a * L * \text{Convert(in, ft)})$
 $x = L$ **"for tip temperature"**
 $\text{DELTAT} = T_w - T_{\text{tip}}$

k_{spoon} [Btu/h.ft.F]	ΔT [F]
5	124.9
16.58	122.6
28.16	117.8
39.74	112.5
51.32	107.1
62.89	102
74.47	97.21
86.05	92.78
97.63	88.69
109.2	84.91
120.8	81.42
132.4	78.19
143.9	75.19
155.5	72.41
167.1	69.82
178.7	67.4
190.3	65.14
201.8	63.02
213.4	61.04
225	59.17



L [in]	ΔT [F]
5	122.4
5.5	123.4
6	124
6.5	124.3
7	124.6
7.5	124.7
8	124.8
8.5	124.9
9	124.9
9.5	125
10	125
10.5	125
11	125
11.5	125
12	125



10-121 A circuit board houses 80 logic chips on one side, dissipating 0.04 W each through the back side of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 864 aluminum pin fins on the back surface.

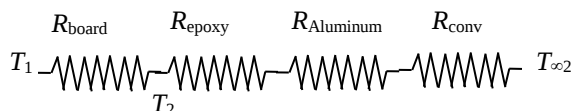
Assumptions 1 Steady operating conditions exist. 2 The temperature in the board and along the fins varies in one direction only (normal to the board). 3 All the heat generated in the chips is conducted across the circuit board, and is dissipated from the back side of the board. 4 Heat transfer from the fin tips is negligible. 5 The heat transfer coefficient is constant and uniform over the entire fin surface. 6 The thermal properties of the fins are constant. 7 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivities are given to be $k = 30 \text{ W/m}\cdot^\circ\text{C}$ for the circuit board, $k = 237 \text{ W/m}\cdot^\circ\text{C}$ for the aluminum plate and fins, and $k = 1.8 \text{ W/m}\cdot^\circ\text{C}$ for the epoxy adhesive.

Analysis (a) The total rate of heat transfer dissipated by the chips is

$$\dot{Q} = 80 \times (0.04 \text{ W}) = 3.2 \text{ W}$$

The individual resistances are



$$A = (0.12 \text{ m})(0.18 \text{ m}) = 0.0216 \text{ m}^2$$

$$R_{\text{board}} = \frac{L}{kA} = \frac{0.003 \text{ m}}{(30 \text{ W/m}\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 0.00463^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(40 \text{ W/m}^2\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 1.1574^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.00463 + 1.1574 = 1.1620^\circ\text{C/W}$$

The temperatures on the two sides of the circuit board are

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q}R_{\text{total}} = 40^\circ\text{C} + (3.2 \text{ W})(1.1620^\circ\text{C/W}) = 43.7^\circ\text{C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 43.7^\circ\text{C} - (3.2 \text{ W})(0.00463^\circ\text{C/W}) = 43.7 - 0.015 \cong 43.7^\circ\text{C}$$

Therefore, the board is nearly isothermal.

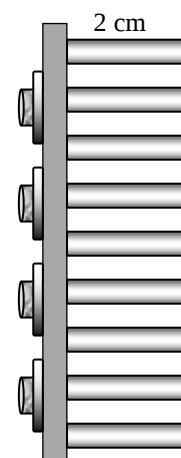
(b) Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(40 \text{ W/m}^2\cdot^\circ\text{C})}{(237 \text{ W/m}\cdot^\circ\text{C})(0.0025 \text{ m})}} = 16.43 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(16.43 \text{ m}^{-1} \times 0.02 \text{ m})}{16.43 \text{ m}^{-1} \times 0.02 \text{ m}} = 0.965$$

The fins can be assumed to be at base temperature provided that the fin area is modified by multiplying it by 0.965. Then the various thermal resistances are

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0002 \text{ m}}{(1.8 \text{ W/m}\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 0.0051^\circ\text{C/W}$$



$$R_{Al} = \frac{L}{kA} = \frac{0.002 \text{ m}}{(237 \text{ W/m} \cdot ^\circ\text{C})(0.0216 \text{ m}^2)} = 0.00039 ^\circ\text{C/W}$$

$$A_{\text{finned}} = \eta_{\text{fin}} n \pi D L = 0.965 \times 864 \pi (0.0025 \text{ m})(0.02 \text{ m}) = 0.131 \text{ m}^2$$

$$A_{\text{unfinned}} = 0.0216 - 864 \frac{\pi D^2}{4} = 0.0216 - 864 \times \frac{\pi (0.0025)^2}{4} = 0.0174 \text{ m}^2$$

$$A_{\text{total, with fins}} = A_{\text{finned}} + A_{\text{unfinned}} = 0.131 + 0.0174 = 0.148 \text{ m}^2$$

$$R_{\text{conv}} = \frac{1}{hA_{\text{total, with fins}}} = \frac{1}{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.148 \text{ m}^2)} = 0.1689 ^\circ\text{C/W}$$

$$\begin{aligned} R_{\text{total}} &= R_{\text{board}} + R_{\text{epoxy}} + R_{\text{aluminum}} + R_{\text{conv}} \\ &= 0.00463 + 0.0051 + 0.00039 + 0.1689 = 0.1790 ^\circ\text{C/W} \end{aligned}$$

Then the temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q} R_{\text{total}} = 40^\circ\text{C} + (3.2 \text{ W})(0.1790 ^\circ\text{C/W}) = \mathbf{40.6^\circ\text{C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q} R_{\text{board}} = 40.6^\circ\text{C} - (3.2 \text{ W})(0.00463 ^\circ\text{C/W}) = 40.6 - 0.015 \cong \mathbf{40.6^\circ\text{C}}$$

10-122 A circuit board houses 80 logic chips on one side, dissipating 0.04 W each through the back side of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 864 copper pin fins on the back surface.

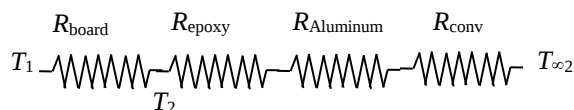
Assumptions 1 Steady operating conditions exist. 2 The temperature in the board and along the fins varies in one direction only (normal to the board). 3 All the heat generated in the chips is conducted across the circuit board, and is dissipated from the back side of the board. 4 Heat transfer from the fin tips is negligible. 5 The heat transfer coefficient is constant and uniform over the entire fin surface. 6 The thermal properties of the fins are constant. 7 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivities are given to be $k = 20 \text{ W/m}\cdot^\circ\text{C}$ for the circuit board, $k = 386 \text{ W/m}\cdot^\circ\text{C}$ for the copper plate and fins, and $k = 1.8 \text{ W/m}\cdot^\circ\text{C}$ for the epoxy adhesive.

Analysis (a) The total rate of heat transfer dissipated by the chips is

$$\dot{Q} = 80 \times (0.04 \text{ W}) = 3.2 \text{ W}$$

The individual resistances are



$$A = (0.12 \text{ m})(0.18 \text{ m}) = 0.0216 \text{ m}^2$$

$$R_{\text{board}} = \frac{L}{kA} = \frac{0.003 \text{ m}}{(30 \text{ W/m}\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 0.00463^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(40 \text{ W/m}^2\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 1.1574^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.00463 + 1.1574 = 1.1620^\circ\text{C/W}$$

The temperatures on the two sides of the circuit board are

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q}R_{\text{total}} = 40^\circ\text{C} + (3.2 \text{ W})(1.1620^\circ\text{C/W}) = 43.7^\circ\text{C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 43.7^\circ\text{C} - (3.2 \text{ W})(0.00463^\circ\text{C/W}) = 43.7 - 0.015 \cong 43.7^\circ\text{C}$$

Therefore, the board is nearly isothermal.

(b) Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

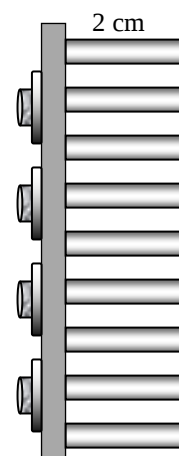
$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(40 \text{ W/m}^2\cdot^\circ\text{C})}{(386 \text{ W/m}\cdot^\circ\text{C})(0.0025 \text{ m})}} = 12.88 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(12.88 \text{ m}^{-1} \times 0.02 \text{ m})}{12.88 \text{ m}^{-1} \times 0.02 \text{ m}} = 0.978$$

The fins can be assumed to be at base temperature provided that the fin area is modified by multiplying it by 0.978. Then the various thermal resistances are

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0002 \text{ m}}{(1.8 \text{ W/m}\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 0.0051^\circ\text{C/W}$$

$$R_{\text{Cu}} = \frac{L}{kA} = \frac{0.002 \text{ m}}{(386 \text{ W/m}\cdot^\circ\text{C})(0.0216 \text{ m}^2)} = 0.00024^\circ\text{C/W}$$



$$A_{\text{finned}} = \eta_{\text{fin}} n \pi D L = 0.978 \times 864 \pi (0.0025 \text{ m})(0.02 \text{ m}) = 0.133 \text{ m}^2$$

$$A_{\text{unfinned}} = 0.0216 - 864 \frac{\pi D^2}{4} = 0.0216 - 864 \times \frac{\pi (0.0025)^2}{4} = 0.0174 \text{ m}^2$$

$$A_{\text{total, with fins}} = A_{\text{finned}} + A_{\text{unfinned}} = 0.133 + 0.0174 = 0.150 \text{ m}^2$$

$$R_{\text{conv}} = \frac{1}{h A_{\text{total, with fins}}} = \frac{1}{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.150 \text{ m}^2)} = 0.1667 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{epoxy}} + R_{\text{aluminum}} + R_{\text{conv}} \\ = 0.00463 + 0.0051 + 0.00024 + 0.1667 = 0.1767 ^\circ\text{C/W}$$

Then the temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q} R_{\text{total}} = 40^\circ\text{C} + (3.2 \text{ W})(0.1767 ^\circ\text{C/W}) = \mathbf{40.6^\circ\text{C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q} R_{\text{board}} = 40.6^\circ\text{C} - (3.2 \text{ W})(0.00463 ^\circ\text{C/W}) = 40.6 - 0.015 \cong \mathbf{40.6^\circ\text{C}}$$

10-123 A hot plate is to be cooled by attaching aluminum pin fins on one side. The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 Heat transfer from the fin tips is negligible. 4 The heat transfer coefficient is constant and uniform over the entire fin surface. 5 The thermal properties of the fins are constant. 6 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum plate and fins is given to be $k = 237 \text{ W/m}\cdot^\circ\text{C}$.

Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(35 \text{ W/m}^2\cdot^\circ\text{C})}{(237 \text{ W/m}\cdot^\circ\text{C})(0.0025 \text{ m})}} = 15.37 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(15.37 \text{ m}^{-1} \times 0.03 \text{ m})}{15.37 \text{ m}^{-1} \times 0.03 \text{ m}} = 0.935$$

The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are

$$n = \frac{1 \text{ m}^2}{(0.006 \text{ m})(0.006 \text{ m})} = 27,777$$

$$A_{\text{fin}} = 27777 \left[\pi DL + \frac{\pi D^2}{4} \right] = 27777 \left[\pi(0.0025)(0.03) + \frac{\pi(0.0025)^2}{4} \right] = 6.68 \text{ m}^2$$

$$A_{\text{unfinned}} = 1 - 27777 \left(\frac{\pi D^2}{4} \right) = 1 - 27777 \left[\frac{\pi(0.0025)^2}{4} \right] = 0.86 \text{ m}^2$$

$$\begin{aligned} \dot{Q}_{\text{finned}} &= \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}} (T_b - T_\infty) \\ &= 0.935(35 \text{ W/m}^2\cdot^\circ\text{C})(6.68 \text{ m}^2)(100 - 30)^\circ\text{C} \\ &= 15,300 \text{ W} \end{aligned}$$

$$\begin{aligned} \dot{Q}_{\text{unfinned}} &= h A_{\text{unfinned}} (T_b - T_\infty) = (35 \text{ W/m}^2\cdot^\circ\text{C})(0.86 \text{ m}^2)(100 - 30)^\circ\text{C} \\ &= 2107 \text{ W} \end{aligned}$$

Then the total heat transfer from the finned plate becomes

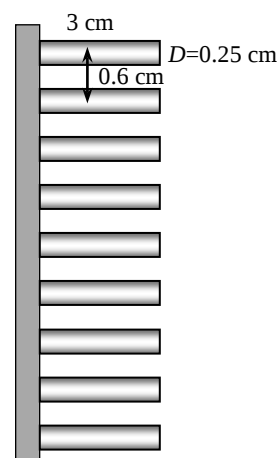
$$\dot{Q}_{\text{total,fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}} = 15,300 + 2107 = 1.74 \times 10^4 \text{ W} = \mathbf{17.4 \text{ kW}}$$

The rate of heat transfer if there were no fin attached to the plate would be

$$\begin{aligned} A_{\text{no fin}} &= (1 \text{ m})(1 \text{ m}) = 1 \text{ m}^2 \\ \dot{Q}_{\text{no fin}} &= h A_{\text{no fin}} (T_b - T_\infty) = (35 \text{ W/m}^2\cdot^\circ\text{C})(1 \text{ m}^2)(100 - 30)^\circ\text{C} = 2450 \text{ W} \end{aligned}$$

Then the fin effectiveness becomes

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} = \frac{17,400}{2450} = \mathbf{7.10}$$



10-124 A hot plate is to be cooled by attaching copper pin fins on one side. The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 Heat transfer from the fin tips is negligible. 4 The heat transfer coefficient is constant and uniform over the entire fin surface. 5 The thermal properties of the fins are constant. 6 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the copper plate and fins is given to be $k = 386 \text{ W/m}\cdot^\circ\text{C}$.

Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(35 \text{ W/m}^2\cdot^\circ\text{C})}{(386 \text{ W/m}\cdot^\circ\text{C})(0.0025 \text{ m})}} = 12.04 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(12.04 \text{ m}^{-1} \times 0.03 \text{ m})}{12.04 \text{ m}^{-1} \times 0.03 \text{ m}} = 0.959$$

The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are

$$n = \frac{1 \text{ m}^2}{(0.006 \text{ m})(0.006 \text{ m})} = 27777$$

$$A_{\text{fin}} = 27777 \left[\pi DL + \frac{\pi D^2}{4} \right] = 27777 \left[\pi(0.0025)(0.03) + \frac{\pi(0.0025)^2}{4} \right] = 6.68 \text{ m}^2$$

$$A_{\text{unfinned}} = 1 - 27777 \left(\frac{\pi D^2}{4} \right) = 1 - 27777 \left[\frac{\pi(0.0025)^2}{4} \right] = 0.86 \text{ m}^2$$

$$\begin{aligned} \dot{Q}_{\text{finned}} &= \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}} (T_b - T_\infty) \\ &= 0.959(35 \text{ W/m}^2\cdot^\circ\text{C})(6.68 \text{ m}^2)(100 - 30)^\circ\text{C} \\ &= 15,700 \text{ W} \end{aligned}$$

$$\dot{Q}_{\text{unfinned}} = h A_{\text{unfinned}} (T_b - T_\infty) = (35 \text{ W/m}^2\cdot^\circ\text{C})(0.86 \text{ m}^2)(100 - 30)^\circ\text{C} = 2107 \text{ W}$$

Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}} = 15,700 + 2107 = 1.78 \times 10^4 \text{ W} = \mathbf{17.8 \text{ kW}}$$

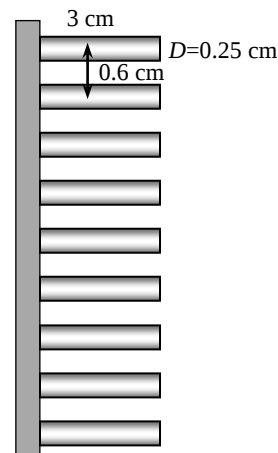
The rate of heat transfer if there were no fin attached to the plate would be

$$A_{\text{no fin}} = (1 \text{ m})(1 \text{ m}) = 1 \text{ m}^2$$

$$\dot{Q}_{\text{no fin}} = h A_{\text{no fin}} (T_b - T_\infty) = (35 \text{ W/m}^2\cdot^\circ\text{C})(1 \text{ m}^2)(100 - 30)^\circ\text{C} = 2450 \text{ W}$$

Then the fin effectiveness becomes

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} = \frac{17800}{2450} = \mathbf{7.27}$$



10-125 EES Prob. 10-123 is reconsidered. The effect of the center-to center distance of the fins on the rate of heat transfer from the surface and the overall effectiveness of the fins is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

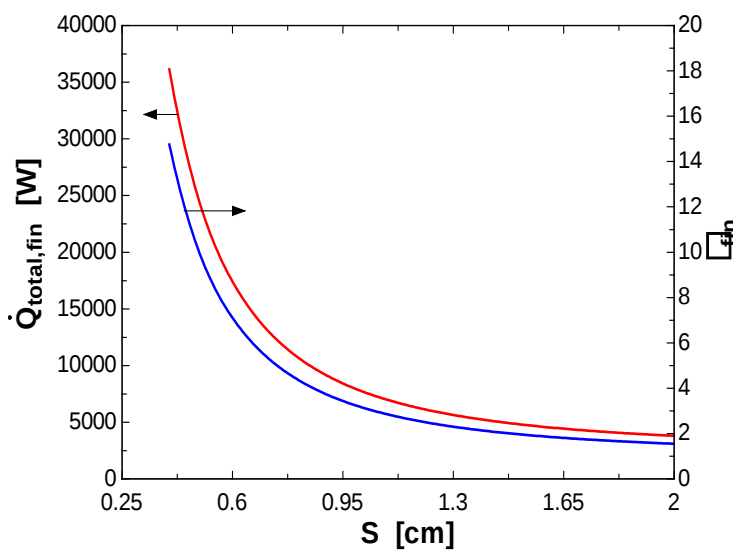
"GIVEN"

T_b=100 [C]
 L=0.03 [m]
 D=0.0025 [m]
 k=237 [W/m-C]
 S=0.6 [cm]
 T_{infinity}=30 [C]
 h=35 [W/m^2-C]
 A_{surface}=1*1 [m^2]

"ANALYSIS"

p=pi*D
 A_c=pi*D^2/4
 a=sqrt((h*p)/(k*A_c))
 eta_{fin}=tanh(a*L)/(a*L)
 n=A_{surface}/(S^2*Convert(cm^2, m^2)) **"number of fins"**
 A_{fin}=n*(pi*D*L+pi*D^2/4)
 A_{unfinned}=A_{surface}-n*(pi*D^2/4)
 Q_{dot}_{finned}=eta_{fin}*h*A_{fin}*(T_b-T_{infinity})
 Q_{dot}_{unfinned}=h*A_{unfinned}*(T_b-T_{infinity})
 Q_{dot}_{total}_{fin}=Q_{dot}_{finned}+Q_{dot}_{unfinned}
 Q_{dot}_{nofin}=h*A_{surface}*(T_b-T_{infinity})
 epsilon_{fin}=Q_{dot}_{total}_{fin}/Q_{dot}_{nofin}

S [cm]	Q _{total fin} [W]	ε _{fin}
0.4	36123	14.74
0.5	24001	9.796
0.6	17416	7.108
0.7	13445	5.488
0.8	10868	4.436
0.9	9101	3.715
1	7838	3.199
1.1	6903	2.817
1.2	6191	2.527
1.3	5638	2.301
1.4	5199	2.122
1.5	4845	1.977
1.6	4555	1.859
1.7	4314	1.761
1.8	4113	1.679
1.9	3942	1.609
2	3797	1.55

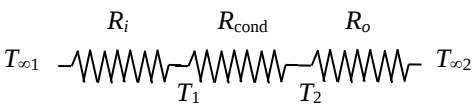


10-126 Two cast iron steam pipes are connected to each other through two 1-cm thick flanges exposed to cold ambient air. The average outer surface temperature of the pipe, the fin efficiency, the rate of heat transfer from the flanges, and the equivalent pipe length of the flange for heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the flanges (fins) varies in one direction only (normal to the pipe). 3 The heat transfer coefficient is constant and uniform over the entire fin surface. 4 The thermal properties of the fins are constant. 5 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the cast iron is given to be $k = 52 \text{ W/m}\cdot^\circ\text{C}$.

Analysis (a) We treat the flanges as fins. The individual thermal resistances are

$$\begin{aligned}
 A_i &= \pi D_i L = \pi (0.092 \text{ m})(6 \text{ m}) = 1.73 \text{ m}^2 \\
 A_o &= \pi D_o L = \pi (0.1 \text{ m})(6 \text{ m}) = 1.88 \text{ m}^2 \\
 R_i &= \frac{1}{h_i A_i} = \frac{1}{(180 \text{ W/m}^2 \cdot ^\circ\text{C})(1.73 \text{ m}^2)} = 0.0032 \text{ }^\circ\text{C/W} \\
 R_{\text{cond}} &= \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(5 / 4.6)}{2\pi (52 \text{ W/m}\cdot^\circ\text{C})(6 \text{ m})} = 0.00004 \text{ }^\circ\text{C/W} \\
 R_o &= \frac{1}{h_o A_o} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(1.88 \text{ m}^2)} = 0.0213 \text{ }^\circ\text{C/W} \\
 R_{\text{total}} &= R_i + R_{\text{cond}} + R_o = 0.0032 + 0.00004 + 0.0213 = 0.0245 \text{ }^\circ\text{C/W}
 \end{aligned}$$


The rate of heat transfer and average outer surface temperature of the pipe are

$$\begin{aligned}
 \dot{Q} &= \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(200 - 12)^\circ\text{C}}{0.0245 \text{ }^\circ\text{C}} = 7673 \text{ W} \\
 \dot{Q} &= \frac{T_2 - T_{\infty 2}}{R_o} \longrightarrow T_2 = T_{\infty 2} + \dot{Q} R_o = 12^\circ\text{C} + (7673 \text{ W})(0.0213 \text{ }^\circ\text{C/W}) = \mathbf{175.4^\circ\text{C}}
 \end{aligned}$$

(b) The fin efficiency can be determined from (Fig. 10-43)

$$\left. \begin{aligned}
 \frac{r_2 + \frac{t}{2}}{r_1} &= \frac{0.1 + \frac{0.02}{2}}{0.05} = 2.2 \\
 \xi &= L_c^{3/2} \left(\frac{h}{k A_p} \right)^{1/2} = \left(L + \frac{t}{2} \right) \sqrt{\frac{h}{k t}} = \left(0.05 \text{ m} + \frac{0.02}{2} \text{ m} \right) \sqrt{\frac{25 \text{ W/m}^2 \cdot ^\circ\text{C}}{(52 \text{ W/m}\cdot^\circ\text{C})(0.02 \text{ m})}} = 0.29
 \end{aligned} \right\} \eta_{\text{fin}} = 0.88$$

$$A_{\text{fin}} = 2\pi(r_2^2 - r_1^2) + 2\pi r_2 t = 2\pi[(0.1 \text{ m})^2 - (0.05 \text{ m})^2] + 2\pi(0.1 \text{ m})(0.02 \text{ m}) = 0.0597 \text{ m}^2$$

The heat transfer rate from the flanges is

$$\begin{aligned}
 \dot{Q}_{\text{finned}} &= \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}} (T_b - T_{\infty}) \\
 &= 0.88(25 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0597 \text{ m}^2)(175.4 - 12)^\circ\text{C} = \mathbf{215 \text{ W}}
 \end{aligned}$$

(c) A 6-m long section of the steam pipe is losing heat at a rate of 7673 W or 7673/6 = 1279 W per m length. Then for heat transfer purposes the flange section is equivalent to

$$\text{Equivalent length} = \frac{215 \text{ W}}{1279 \text{ W/m}} = 0.168 \text{ m} = \mathbf{16.8 \text{ cm}}$$

Therefore, the flange acts like a fin and increases the heat transfer by $16.8/2 = 8.4$ times.

Heat Transfer in Common Configurations

10-127C Under steady conditions, the rate of heat transfer between two surfaces is expressed as $\dot{Q} = Sk(T_1 - T_2)$ where S is the conduction shape factor. It is related to the thermal resistance by $S = 1/(kR)$.

10-128C It provides an easy way of calculating the steady rate of heat transfer between two isothermal surfaces in common configurations.

10-129 The hot water pipe of a district heating system is buried in the soil. The rate of heat loss from the pipe is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the soil is constant.

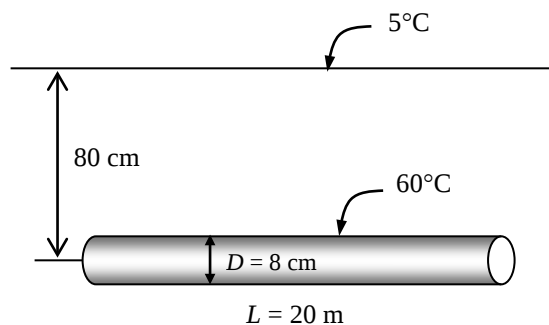
Properties The thermal conductivity of the soil is given to be $k = 0.9 \text{ W/m}\cdot^\circ\text{C}$.

Analysis Since $z > 1.5D$, the shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi L}{\ln(4z/D)} = \frac{2\pi(20 \text{ m})}{\ln[4(0.8 \text{ m})/(0.08 \text{ m})]} = 34.07 \text{ m}$$

Then the steady rate of heat transfer from the pipe becomes

$$\dot{Q} = Sk(T_1 - T_2) = (34.07 \text{ m})(0.9 \text{ W/m}\cdot^\circ\text{C})(60 - 5)^\circ\text{C} = \mathbf{1686 \text{ W}}$$



10-130 EES Prob. 10-129 is reconsidered. The rate of heat loss from the pipe as a function of the burial depth is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$L=20 \text{ [m]}$$

$$D=0.08 \text{ [m]}$$

$$z=0.80 \text{ [m]}$$

$$T_1=60 \text{ [C]}$$

$$T_2=5 \text{ [C]}$$

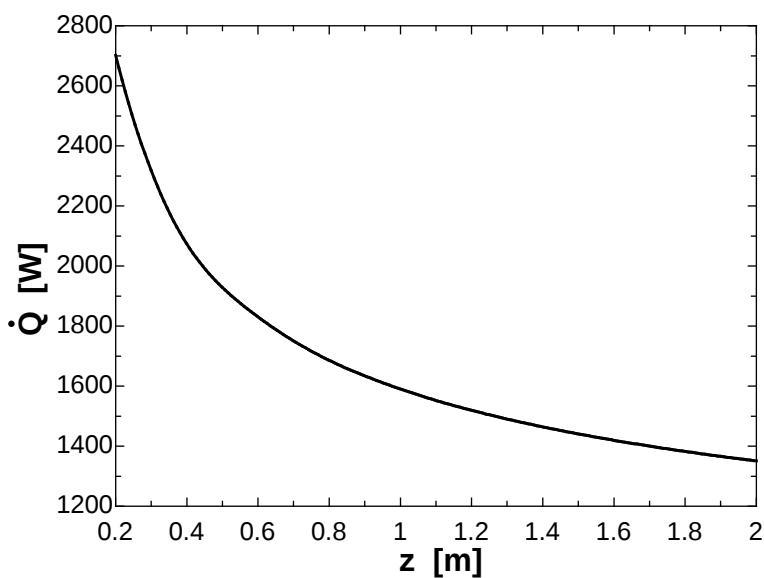
$$k=0.9 \text{ [W/m-C]}$$

"ANALYSIS"

$$S=(2*\pi*L)/\ln(4*z/D)$$

$$\dot{Q}_{\text{dot}}=S*k*(T_1-T_2)$$

z [m]	Q [W]
0.2	2701
0.38	2113
0.56	1867
0.74	1723
0.92	1625
1.1	1552
1.28	1496
1.46	1450
1.64	1412
1.82	1379
2	1351



10-131 Hot and cold water pipes run parallel to each other in a thick concrete layer. The rate of heat transfer between the pipes is to be determined.

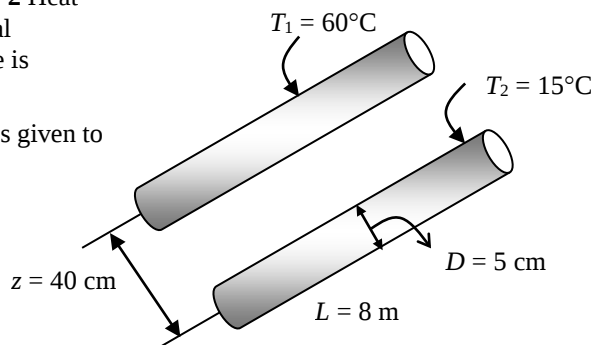
Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the concrete is constant.

Properties The thermal conductivity of concrete is given to be $k = 0.75 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi L}{\cosh^{-1}\left(\frac{4z^2 - D_1^2 - D_2^2}{2D_1D_2}\right)}$$

$$= \frac{2\pi(8 \text{ m})}{\cosh^{-1}\left(\frac{4(0.4 \text{ m})^2 - (0.05 \text{ m})^2 - (0.05 \text{ m})^2}{2(0.05 \text{ m})(0.05 \text{ m})}\right)} = 9.078 \text{ m}$$



Then the steady rate of heat transfer between the pipes becomes

$$\dot{Q} = Sk(T_1 - T_2) = (9.078 \text{ m})(0.75 \text{ W/m}\cdot^\circ\text{C})(60 - 15)^\circ\text{C} = \mathbf{306 \text{ W}}$$

10-132 EES Prob. 10-131 is reconsidered. The rate of heat transfer between the pipes as a function of the distance between the centerlines of the pipes is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

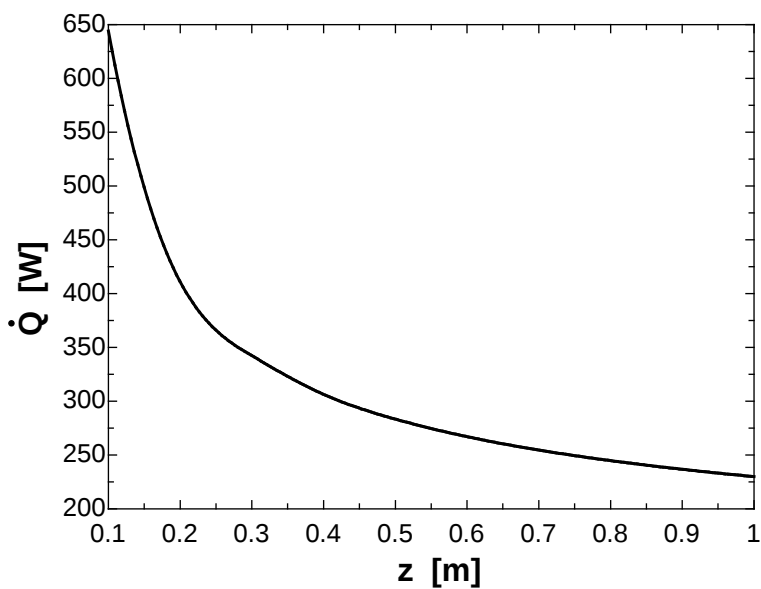
"GIVEN"

L=8 [m]
D_1=0.05 [m]
D_2=D_1
z=0.40 [m]
T_1=60 [C]
T_2=15 [C]
k=0.75 [W/m-C]

"ANALYSIS"

$S = (2\pi k L) / (\text{arccosh}((4z^2 - D_1^2 - D_2^2) / (2D_1 D_2)))$
 $\dot{Q} = S k (T_1 - T_2)$

z [m]	Q [W]
0.1	644.1
0.2	411.1
0.3	342.3
0.4	306.4
0.5	283.4
0.6	267
0.7	254.7
0.8	244.8
0.9	236.8
1	230



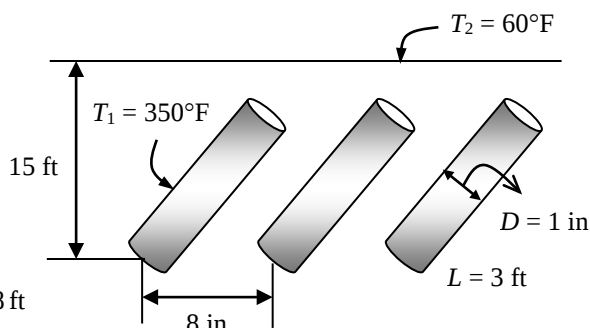
10-133E A row of used uranium fuel rods are buried in the ground parallel to each other. The rate of heat transfer from the fuel rods to the atmosphere through the soil is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the soil is constant.

Properties The thermal conductivity of the soil is given to be $k = 0.6 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$S_{\text{total}} = 4 \times \frac{2\pi L}{\ln\left(\frac{2w}{\pi D} \sinh \frac{2\pi z}{w}\right)} = 4 \times \frac{2\pi(3 \text{ ft})}{\ln\left(\frac{2(8/12 \text{ ft})}{\pi(1/12 \text{ ft})} \sinh \frac{2\pi(15 \text{ ft})}{(8/12 \text{ ft})}\right)} = 0.5298 \text{ ft}$$



Then the steady rate of heat transfer from the fuel rods becomes

$$\dot{Q} = S_{\text{total}} k (T_1 - T_2) = (0.5298 \text{ ft})(0.6 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(350 - 60)^\circ\text{F} = \mathbf{92.2 \text{ Btu/h}}$$

10-134 Hot water flows through a 5-m long section of a thin walled hot water pipe that passes through the center of a 14-cm thick wall filled with fiberglass insulation. The rate of heat transfer from the pipe to the air in the rooms and the temperature drop of the hot water as it flows through the pipe are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the fiberglass insulation is constant. 4 The pipe is at the same temperature as the hot water.

Properties The thermal conductivity of fiberglass insulation is given to be $k = 0.035 \text{ W/m}\cdot^\circ\text{C}$.

Analysis (a) The shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi L}{\ln\left(\frac{8z}{\pi D}\right)} = \frac{2\pi(5 \text{ m})}{\ln\left[\frac{8(0.07 \text{ m})}{\pi(0.025 \text{ m})}\right]} = 16 \text{ m}$$

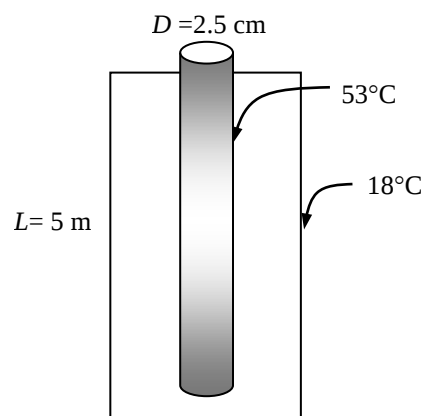
Then the steady rate of heat transfer from the pipe becomes

$$\dot{Q} = Sk(T_1 - T_2) = (16 \text{ m})(0.035 \text{ W/m}\cdot^\circ\text{C})(53 - 18)^\circ\text{C} = \mathbf{19.6 \text{ W}}$$

(b) Using the water properties at the room temperature, the temperature drop of the hot water as it flows through this 5-m section of the wall becomes

$$\dot{Q} = \dot{m} c_p \Delta T$$

$$\Delta T = \frac{\dot{Q}}{\dot{m} c_p} = \frac{\dot{Q}}{\rho \dot{V} c_p} = \frac{\dot{Q}}{\rho V A_c c_p} = \frac{19.6 \text{ J/s}}{(1000 \text{ kg/m}^3)(0.4 \text{ m/s}) \left[\frac{\pi(0.025 \text{ m})^2}{4} \right] (4180 \text{ J/kg}\cdot^\circ\text{C})} = \mathbf{0.024^\circ\text{C}}$$



10-135 Hot water is flowing through a pipe that extends 2 m in the ambient air and continues in the ground before it enters the next building. The surface of the ground is covered with snow at 0°C. The total rate of heat loss from the hot water and the temperature drop of the hot water in the pipe are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the ground is constant. 4 The pipe is at the same temperature as the hot water.

Properties The thermal conductivity of the ground is given to be $k = 1.5 \text{ W/m}\cdot^\circ\text{C}$.

Analysis (a) We assume that the surface temperature of the tube is equal to the temperature of the water. Then the heat loss from the part of the tube that is on the ground is

$$A_s = \pi DL = \pi(0.05 \text{ m})(2 \text{ m}) = 0.3142 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$= (22 \text{ W/m}^2\cdot^\circ\text{C})(0.3142 \text{ m}^2)(80 - 8)^\circ\text{C} = 498 \text{ W}$$

Considering the shape factor, the heat loss for vertical part of the tube can be determined from

$$S = \frac{2\pi L}{\ln\left(\frac{4L}{D}\right)} = \frac{2\pi(3 \text{ m})}{\ln\left[\frac{4(3 \text{ m})}{(0.05 \text{ m})}\right]} = 3.44 \text{ m}$$

$$\dot{Q} = Sk(T_1 - T_2) = (3.44 \text{ m})(1.5 \text{ W/m}\cdot^\circ\text{C})(80 - 0)^\circ\text{C} = 413 \text{ W}$$

The shape factor, and the rate of heat loss on the horizontal part that is in the ground are

$$S = \frac{2\pi L}{\ln\left(\frac{4z}{D}\right)} = \frac{2\pi(20 \text{ m})}{\ln\left[\frac{4(3 \text{ m})}{(0.05 \text{ m})}\right]} = 22.9 \text{ m}$$

$$\dot{Q} = Sk(T_1 - T_2) = (22.9 \text{ m})(1.5 \text{ W/m}\cdot^\circ\text{C})(80 - 0)^\circ\text{C} = 2748 \text{ W}$$

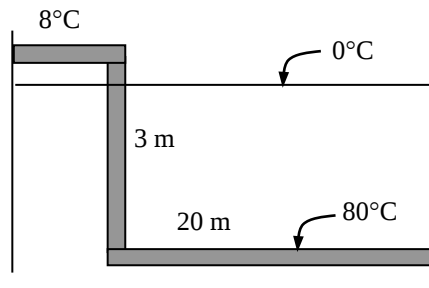
and the total rate of heat loss from the hot water becomes

$$\dot{Q}_{\text{total}} = 498 + 413 + 2748 = \mathbf{3659 \text{ W}}$$

(b) Using the water properties at the room temperature, the temperature drop of the hot water as it flows through this 25-m section of the wall becomes

$$\dot{Q} = \dot{m}c_p\Delta T$$

$$\Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{\dot{Q}}{(\rho\dot{V})c_p} = \frac{\dot{Q}}{(\rho VA_c)c_p} = \frac{3659 \text{ J/s}}{(1000 \text{ kg/m}^3)(1.5 \text{ m/s})\left[\frac{\pi(0.05 \text{ m})^2}{4}\right](4180 \text{ J/kg}\cdot^\circ\text{C})} = \mathbf{0.32^\circ\text{C}}$$



10-136 The walls and the roof of the house are made of 20-cm thick concrete, and the inner and outer surfaces of the house are maintained at specified temperatures. The rate of heat loss from the house through its walls and the roof is to be determined, and the error involved in ignoring the edge and corner effects is to be assessed.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer at the edges and corners is two- or three-dimensional. 3 Thermal conductivity of the concrete is constant. 4 The edge effects of adjoining surfaces on heat transfer are to be considered.

Properties The thermal conductivity of the concrete is given to be $k = 0.75 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The rate of heat transfer excluding the edges and corners is first determined to be

$$A_{\text{total}} = (12 - 0.4)(12 - 0.4) + 4(12 - 0.4)(6 - 0.2) = 403.7 \text{ m}^2$$

$$\dot{Q} = \frac{kA_{\text{total}}}{L}(T_1 - T_2) = \frac{(0.75 \text{ W/m}\cdot^\circ\text{C})(403.7 \text{ m}^2)}{0.2 \text{ m}}(15 - 3)^\circ\text{C} = 18,167 \text{ W}$$

The heat transfer rate through the edges can be determined using the shape factor relations in Table 10-7,

$$S_{\text{corners+edges}} = 4 \times \text{corners} + 4 \times \text{edges} = 4 \times 0.15L + 4 \times 0.54w$$

$$= 4 \times 0.15(0.2 \text{ m}) + 4 \times 0.54(12 \text{ m}) = 26.04 \text{ m}$$

$$\dot{Q}_{\text{corners+edges}} = S_{\text{corners+edges}} k(T_1 - T_2) = (26.04 \text{ m})(0.75 \text{ W/m}\cdot^\circ\text{C})(15 - 3)^\circ\text{C} = 234 \text{ W}$$

and $\dot{Q}_{\text{total}} = 18,167 + 234 = 1.840 \times 10^4 \text{ W} = \mathbf{18.4 \text{ kW}}$

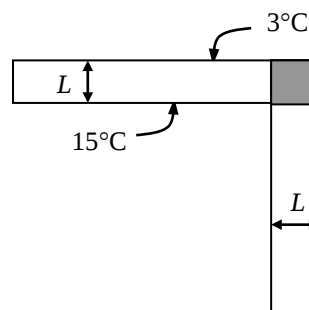
Ignoring the edge effects of adjoining surfaces, the rate of heat transfer is determined from

$$A_{\text{total}} = (12)(12) + 4(12)(6) = 432 \text{ m}^2$$

$$\dot{Q} = \frac{kA_{\text{total}}}{L}(T_1 - T_2) = \frac{(0.75 \text{ W/m}\cdot^\circ\text{C})(432 \text{ m}^2)}{0.2 \text{ m}}(15 - 3)^\circ\text{C} = 1.94 \times 10^4 = 19.4 \text{ kW}$$

The percentage error involved in ignoring the effects of the edges then becomes

$$\% \text{error} = \frac{19.4 - 18.4}{18.4} \times 100 = \mathbf{5.4\%}$$



10-137 The inner and outer surfaces of a long thick-walled concrete duct are maintained at specified temperatures. The rate of heat transfer through the walls of the duct is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the concrete is constant.

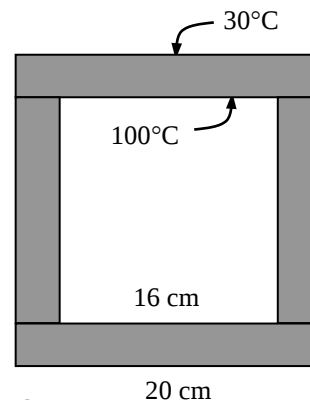
Properties The thermal conductivity of concrete is given to be $k = 0.75 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$\frac{a}{b} = \frac{20}{16} = 1.25 < 1.41 \longrightarrow S = \frac{2\pi L}{0.785 \ln\left(\frac{a}{b}\right)} = \frac{2\pi(25 \text{ m})}{0.785 \ln 1.25} = 896.7 \text{ m}$$

Then the steady rate of heat transfer through the walls of the duct becomes

$$\dot{Q} = Sk(T_1 - T_2) = (896.7 \text{ m})(0.75 \text{ W/m}\cdot^\circ\text{C})(100 - 30)^\circ\text{C} = 4.71 \times 10^4 \text{ W} = \mathbf{47.1 \text{ kW}}$$



10-138 A spherical tank containing some radioactive material is buried in the ground. The tank and the ground surface are maintained at specified temperatures. The rate of heat transfer from the tank is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the ground is constant.

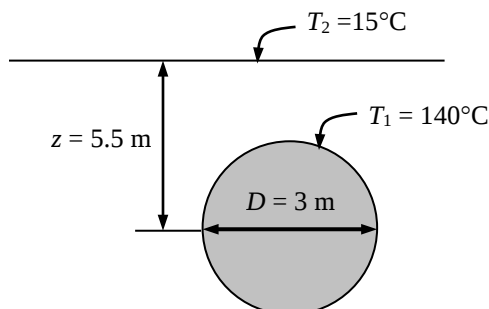
Properties The thermal conductivity of the ground is given to be $k = 1.4 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi D}{1 - 0.25 \frac{D}{z}} = \frac{2\pi(3 \text{ m})}{1 - 0.25 \frac{3 \text{ m}}{5.5 \text{ m}}} = 21.83 \text{ m}$$

Then the steady rate of heat transfer from the tank becomes

$$\dot{Q} = Sk(T_1 - T_2) = (21.83 \text{ m})(1.4 \text{ W/m} \cdot ^\circ\text{C})(140 - 15)^\circ\text{C} = \mathbf{3820 \text{ W}}$$



10-139 EES Prob. 10-138 is reconsidered. The rate of heat transfer from the tank as a function of the tank diameter is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$D=3 \text{ [m]}$

$k=1.4 \text{ [W/m-C]}$

$h=4 \text{ [m]}$

$T_1=140 \text{ [C]}$

$T_2=15 \text{ [C]}$

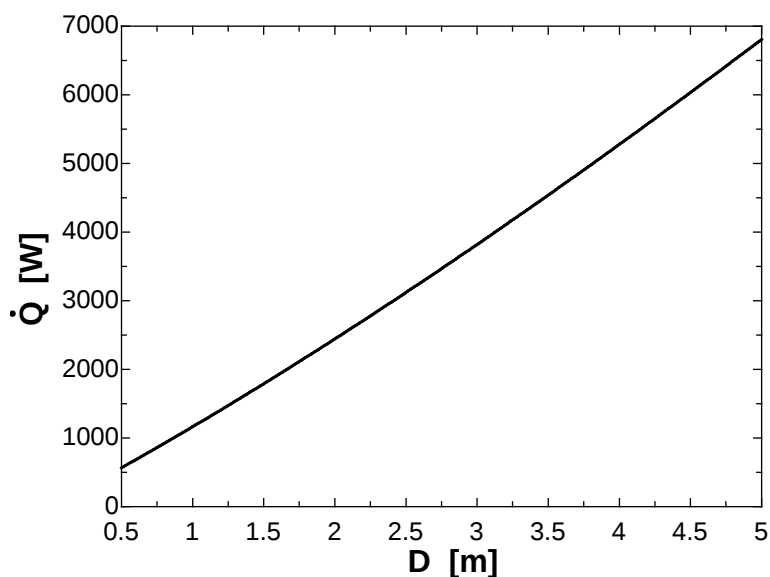
"ANALYSIS"

$z=h+D/2$

$S=(2*\pi*D)/(1-0.25*D/z)$

$Q_dot=S*k*(T_1-T_2)$

D [m]	Q [W]
0.5	566.4
1	1164
1.5	1791
2	2443
2.5	3120
3	3820
3.5	4539
4	5278
4.5	6034
5	6807



10-140 Hot water passes through a row of 8 parallel pipes placed vertically in the middle of a concrete wall whose surfaces are exposed to a medium at 20°C with a heat transfer coefficient of 8 W/m²·°C. The rate of heat loss from the hot water, and the surface temperature of the wall are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of concrete is constant.

Properties The thermal conductivity of concrete is given to be $k = 0.75 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi L}{\ln\left(\frac{8z}{\pi D}\right)} = \frac{2\pi(4 \text{ m})}{\ln\left(\frac{8(0.075 \text{ m})}{\pi(0.03 \text{ m})}\right)} = 13.58 \text{ m}$$

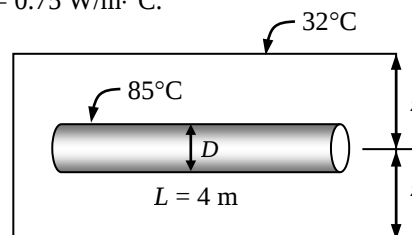
Then rate of heat loss from the hot water in 8 parallel pipes becomes

$$\dot{Q} = 8Sk(T_1 - T_2) = 8(13.58 \text{ m})(0.75 \text{ W/m} \cdot ^\circ\text{C})(85 - 32)^\circ\text{C} = \mathbf{4318 \text{ W}}$$

The surface temperature of the wall can be determined from

$$A_s = 2(4 \text{ m})(8 \text{ m}) = 64 \text{ m}^2 \quad (\text{from both sides})$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 32^\circ\text{C} + \frac{4318 \text{ W}}{(12 \text{ W/m}^2 \cdot ^\circ\text{C})(64 \text{ m}^2)} = \mathbf{37.6^\circ\text{C}}$$



Review Problems

10-141E Steam is produced in copper tubes by heat transferred from another fluid condensing outside the tubes at a high temperature. The rate of heat transfer per foot length of the tube when a 0.01 in thick layer of limestone is formed on the inner surface of the tube is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant. 4 Heat transfer coefficients are constant and uniform over the surfaces.

Properties The thermal conductivities are given to be $k = 223$ Btu/h·ft·°F for copper tubes and $k = 1.7$ Btu/h·ft·°F for limestone.

Analysis The total thermal resistance of the new heat exchanger is

$$\dot{Q}_{\text{new}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total, new}}} \longrightarrow R_{\text{total, new}} = \frac{T_{\infty 1} - T_{\infty 2}}{\dot{Q}_{\text{new}}} = \frac{(350 - 250)^{\circ}\text{F}}{2 \times 10^4 \text{ Btu/h}} = 0.005 \text{ h} \cdot ^{\circ}\text{F/Btu}$$

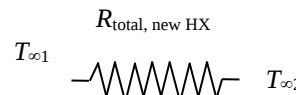
After 0.01 in thick layer of limestone forms, the new value of thermal resistance and heat transfer rate are determined to be

$$R_{\text{limestone, i}} = \frac{\ln(r_1 / r_i)}{2\pi k L} = \frac{\ln(0.5 / 0.49)}{2\pi (1.7 \text{ Btu/h} \cdot \text{ft} \cdot ^{\circ}\text{F})(1 \text{ ft})} = 0.00189 \text{ h} \cdot ^{\circ}\text{F/Btu}$$

$$R_{\text{total, w/lime}} = R_{\text{total, new}} + R_{\text{limestone, i}} = 0.005 + 0.00189 = 0.00689 \text{ h} \cdot ^{\circ}\text{F/Btu}$$

$$\dot{Q}_{\text{w/lime}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total, w/lime}}} = \frac{(350 - 250)^{\circ}\text{F}}{0.00689 \text{ h} \cdot ^{\circ}\text{F/Btu}} = \mathbf{1.45 \times 10^4 \text{ Btu/h}} \quad (\text{a decline of } 27\%)$$

Discussion Note that the limestone layer will change the inner surface area of the pipe and thus the internal convection resistance slightly, but this effect should be negligible.

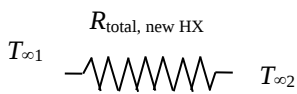


10-142E Steam is produced in copper tubes by heat transferred from another fluid condensing outside the tubes at a high temperature. The rate of heat transfer per foot length of the tube when a 0.01 in thick layer of limestone is formed on the inner and outer surfaces of the tube is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties are constant. 4 Heat transfer coefficients are constant and uniform over the surfaces.

Properties The thermal conductivities are given to be $k = 223$ Btu/h·ft·°F for copper tubes and $k = 1.7$ Btu/h·ft·°F for limestone.

Analysis The total thermal resistance of the new heat exchanger is

$$\dot{Q}_{\text{new}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total, new}}} \longrightarrow R_{\text{total, new}} = \frac{T_{\infty 1} - T_{\infty 2}}{\dot{Q}_{\text{new}}} = \frac{(350 - 250)^{\circ}\text{F}}{2 \times 10^4 \text{ Btu/h}} = 0.005 \text{ h} \cdot ^{\circ}\text{F/Btu}$$


After 0.01 in thick layer of limestone forms, the new value of thermal resistance and heat transfer rate are determined to be

$$R_{\text{limestone, i}} \quad R_{\text{total, new HX}} \quad R_{\text{limestone, o}}$$


$$R_{\text{limestone, i}} = \frac{\ln(r_1 / r_i)}{2\pi kL} = \frac{\ln(0.5 / 0.49)}{2\pi(1.7 \text{ Btu/h} \cdot \text{ft} \cdot ^{\circ}\text{F})(1 \text{ ft})} = 0.00189 \text{ h} \cdot ^{\circ}\text{F/Btu}$$

$$R_{\text{limestone, o}} = \frac{\ln(r_o / r_2)}{2\pi kL} = \frac{\ln(0.66 / 0.65)}{2\pi(1.7 \text{ Btu/h} \cdot \text{ft} \cdot ^{\circ}\text{F})(1 \text{ ft})} = 0.00143 \text{ h} \cdot ^{\circ}\text{F/Btu}$$

$$R_{\text{total, w/lime}} = R_{\text{total, new}} + R_{\text{limestone, i}} + R_{\text{limestone, o}} = 0.005 + 0.00189 + 0.00143 = 0.00832 \text{ h} \cdot ^{\circ}\text{F/Btu}$$

$$\dot{Q}_{\text{w/lime}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total, w/lime}}} = \frac{(350 - 250)^{\circ}\text{F}}{0.00832 \text{ h} \cdot ^{\circ}\text{F/Btu}} = 1.20 \times 10^4 \text{ Btu/h} \quad (\text{a decline of } 40\%)$$

Discussion Note that the limestone layer will change the inner surface area of the pipe and thus the internal convection resistance slightly, but this effect should be negligible.

10-143 A cylindrical tank filled with liquid propane at 1 atm is exposed to convection and radiation. The time it will take for the propane to evaporate completely as a result of the heat gain from the surroundings for the cases of no insulation and 5-cm thick glass wool insulation are to be determined.

Assumptions 1 Heat transfer is steady. 2 Heat transfer is one-dimensional. 3 The combined heat transfer coefficient is constant and uniform over the entire surface. 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the propane inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The heat of vaporization and density of liquid propane at 1 atm are given to be 425 kJ/kg and 581 kg/m³, respectively. The thermal conductivity of glass wool insulation is given to be $k = 0.038$ W/m·°C.

Analysis (a) If the tank is not insulated, the heat transfer rate is determined to be

$$A_{\text{tank}} = \pi DL + 2\pi(\pi D^2 / 4) = \pi(1.2 \text{ m})(6 \text{ m}) + 2\pi(1.2 \text{ m})^2 / 4 = 24.88 \text{ m}^2$$

$$\dot{Q} = hA_{\text{tank}}(T_{\infty 1} - T_{\infty 2}) = (25 \text{ W/m}^2 \cdot ^\circ\text{C})(24.88 \text{ m}^2)[30 - (-42)]^\circ\text{C} = 44,787 \text{ W}$$

The volume of the tank and the mass of the propane are

$$V = \pi r^2 L = \pi(0.6 \text{ m})^2 (6 \text{ m}) = 6.786 \text{ m}^3$$

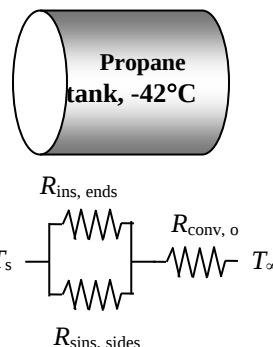
$$m = \rho V = (581 \text{ kg/m}^3)(6.786 \text{ m}^3) = 3942.6 \text{ kg}$$

The rate of vaporization of propane is

$$\dot{Q} = \dot{m}h_{fg} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{44,787 \text{ kJ/s}}{425 \text{ kJ/kg}} = 0.1054 \text{ kg/s}$$

Then the time period for the propane tank to empty becomes

$$\Delta t = \frac{m}{\dot{m}} = \frac{3942.6 \text{ kg}}{0.1054 \text{ kg/s}} = 37,413 \text{ s} = \mathbf{10.4 \text{ hours}}$$



(b) We now repeat calculations for the case of insulated tank with 5-cm thick insulation.

$$A_o = \pi DL + 2\pi(\pi D^2 / 4) = \pi(1.3 \text{ m})(6 \text{ m}) + 2\pi(1.3 \text{ m})^2 / 4 = 27.16 \text{ m}^2$$

$$R_{\text{conv},o} = \frac{1}{h_o A_o} = \frac{1}{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(27.16 \text{ m}^2)} = 0.001473 ^\circ\text{C/W}$$

$$R_{\text{insulation},\text{side}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(65 / 60)}{2\pi(0.038 \text{ W/m} \cdot ^\circ\text{C})(6 \text{ m})} = 0.05587 ^\circ\text{C/W}$$

$$R_{\text{insulation},\text{ends}} = 2 \frac{L}{k A_{\text{avg}}} = \frac{2 \times 0.05 \text{ m}}{(0.038 \text{ W/m} \cdot ^\circ\text{C})[\pi(1.25 \text{ m})^2 / 4]} = 2.1444 ^\circ\text{C/W}$$

Noting that the insulation on the side surface and the end surfaces are in parallel, the equivalent resistance for the insulation is determined to be

$$R_{\text{insulation}} = \left(\frac{1}{R_{\text{insulation},\text{side}}} + \frac{1}{R_{\text{insulation},\text{ends}}} \right)^{-1} = \left(\frac{1}{0.05587 ^\circ\text{C/W}} + \frac{1}{2.1444 ^\circ\text{C/W}} \right)^{-1} = 0.05445 ^\circ\text{C/W}$$

Then the total thermal resistance and the heat transfer rate become

$$R_{\text{total}} = R_{\text{conv},o} + R_{\text{insulation}} = 0.001473 + 0.05445 = 0.05592 ^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty} - T_s}{R_{\text{total}}} = \frac{[30 - (-42)]^\circ\text{C}}{0.05592 ^\circ\text{C/W}} = 1288 \text{ W}$$

Then the time period for the propane tank to empty becomes

$$\dot{Q} = \dot{m}h_{fg} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{1.288 \text{ kJ/s}}{425 \text{ kJ/kg}} = 0.003031 \text{ kg/s}$$

$$\Delta t = \frac{m}{\dot{m}} = \frac{3942.6 \text{ kg}}{0.003031 \text{ kg/s}} = 1.301 \times 10^6 \text{ s} = 361.4 \text{ hours} = \mathbf{15.1 \text{ days}}$$

10-144 Hot water is flowing through a 15-m section of a cast iron pipe. The pipe is exposed to cold air and surfaces in the basement, and it experiences a 3°C-temperature drop. The combined convection and radiation heat transfer coefficient at the outer surface of the pipe is to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any significant change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no significant variation in the axial direction. 3 Thermal properties are constant.

Properties The thermal conductivity of cast iron is given to be $k = 52 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis Using water properties at room temperature, the mass flow rate of water and rate of heat transfer from the water are determined to be

$$\dot{m} = \rho \dot{V}_c = \rho V A_c = (1000 \text{ kg/m}^3)(1.5 \text{ m/s})[\pi(0.03)^2 / 4] \text{ m}^2 = 1.06 \text{ kg/s}$$

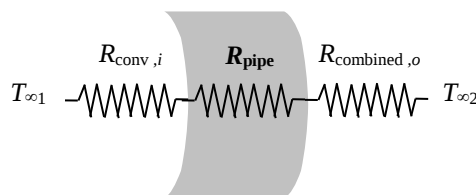
$$\dot{Q} = \dot{m} c_p \Delta T = (1.06 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C})(70 - 67)^\circ\text{C} = 13,296 \text{ W}$$

The thermal resistances for convection in the pipe and the pipe itself are

$$R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k L}$$

$$= \frac{\ln(1.75 / 1.5)}{2\pi(52 \text{ W/m} \cdot ^\circ\text{C})(15 \text{ m})} = 0.000031^\circ\text{C/W}$$

$$R_{\text{conv},i} = \frac{1}{h_i A_i} = \frac{1}{(400 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.03)(15)] \text{ m}^2} = 0.001768^\circ\text{C/W}$$



Using arithmetic mean temperature $(70+67)/2 = 68.5^\circ\text{C}$ for water, the heat transfer can be expressed as

$$\dot{Q} = \frac{T_{\infty,1,ave} - T_{\infty,2}}{R_{\text{total}}} = \frac{T_{\infty,1,ave} - T_{\infty,2}}{R_{\text{conv},i} + R_{\text{pipe}} + R_{\text{combined},o}} = \frac{T_{\infty,1,ave} - T_{\infty,2}}{R_{\text{conv},i} + R_{\text{pipe}} + \frac{1}{h_{\text{combined}} A_o}}$$

Substituting, $13,296 \text{ W} = \frac{(68.5 - 15)^\circ\text{C}}{(0.000031^\circ\text{C/W}) + (0.001768^\circ\text{C/W}) + \frac{1}{h_{\text{combined}}[\pi(0.035)(15)] \text{ m}^2}}$

Solving for the combined heat transfer coefficient gives

$$h_{\text{combined}} = 272.5 \text{ W/m}^2 \cdot ^\circ\text{C}$$

10-145 An 10-m long section of a steam pipe exposed to the ambient is to be insulated to reduce the heat loss through that section of the pipe by 90 percent. The amount of heat loss from the steam in 10 h and the amount of saved per year by insulating the steam pipe.

Assumptions 1 Heat transfer through the pipe is steady and one-dimensional. 2 Thermal conductivities are constant. 3 The furnace operates continuously. 4 The given heat transfer coefficients accounts for the radiation effects. 5 The temperatures of the pipe surface and the surroundings are representative of annual average during operating hours. 6 The plant operates 110 days a year.

Analysis The rate of heat transfer for the uninsulated case is

$$A_o = \pi D_o L = \pi (0.12 \text{ m})(10 \text{ m}) = 3.77 \text{ m}^2$$

$$\dot{Q} = hA_o(T_s - T_{air}) = (35 \text{ W/m}^2 \cdot ^\circ\text{C})(3.77 \text{ m}^2)(82 - 8)^\circ\text{C} = 9764 \text{ W}$$

The amount of heat loss during a 10-hour period is

$$Q = \dot{Q}\Delta t = (9.764 \text{ kJ/s})(10 \times 3600 \text{ s}) = 3.515 \times 10^5 \text{ kJ (per day)}$$

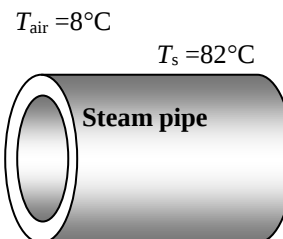
The steam generator has an efficiency of 85%, and steam heating is used for 110 days a year. Then the amount of natural gas consumed per year and its cost are

$$\text{Fuel used} = \frac{3.515 \times 10^5 \text{ kJ}}{0.85} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) (110 \text{ days/yr}) = 431.2 \text{ therms/yr}$$

$$\begin{aligned} \text{Cost of fuel} &= (\text{Amount of fuel})(\text{Unit cost of fuel}) \\ &= (431.2 \text{ therms/yr})(\$1.20/\text{therm}) = \$517.4/\text{yr} \end{aligned}$$

Then the money saved by reducing the heat loss by 90% by insulation becomes

$$\text{Money saved} = 0.9 \times (\text{Cost of fuel}) = 0.9 \times \$517.4/\text{yr} = \mathbf{\$466}$$



10-146 A multilayer circuit board dissipating 27 W of heat consists of 4 layers of copper and 3 layers of epoxy glass sandwiched together. The circuit board is attached to a heat sink from both ends maintained at 35°C. The magnitude and location of the maximum temperature that occurs in the board is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer can be approximated as being one-dimensional. 3 Thermal conductivities are constant. 4 Heat is generated uniformly in the epoxy layers of the board. 5 Heat transfer from the top and bottom surfaces of the board is negligible. 6 The thermal contact resistances at the copper-epoxy interfaces are negligible.

Properties The thermal conductivities are given to be $k = 386 \text{ W/m} \cdot ^\circ\text{C}$ for copper layers and $k = 0.26 \text{ W/m} \cdot ^\circ\text{C}$ for epoxy glass boards.

Analysis The effective conductivity of the multilayer circuit board is first determined to be

$$(kt)_{\text{copper}} = 4[(386 \text{ W/m} \cdot ^\circ\text{C})(0.0002 \text{ m})] = 0.3088 \text{ W/}^\circ\text{C}$$

$$(kt)_{\text{epoxy}} = 3[(0.26 \text{ W/m} \cdot ^\circ\text{C})(0.0015 \text{ m})] = 0.00117 \text{ W/}^\circ\text{C}$$

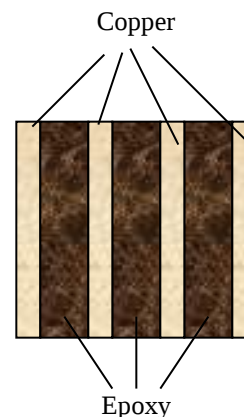
$$k_{\text{eff}} = \frac{(kt)_{\text{copper}} + (kt)_{\text{epoxy}}}{t_{\text{copper}} + t_{\text{epoxy}}} = \frac{(0.3088 + 0.00117) \text{ W/}^\circ\text{C}}{[4(0.0002) + 3(0.0015) \text{ m}]} = 58.48 \text{ W/m} \cdot ^\circ\text{C}$$

The maximum temperature will occur at the midplane of the board that is the farthest to the heat sink. Its value is

$$A = 0.18[4(0.0002) + 3(0.0015)] = 0.000954 \text{ m}^2$$

$$\dot{Q} = \frac{k_{\text{eff}} A}{L} (T_1 - T_2)$$

$$T_{\text{max}} = T_1 = T_2 + \frac{\dot{Q}L}{k_{\text{eff}} A} = 35^\circ\text{C} + \frac{(27/2 \text{ W})(0.18/2 \text{ m})}{(58.48 \text{ W/m} \cdot ^\circ\text{C})(0.000954 \text{ m}^2)} = \mathbf{56.8^\circ\text{C}}$$



10-147 The plumbing system of a house involves some section of a plastic pipe exposed to the ambient air. The pipe is initially filled with stationary water at 0°C. It is to be determined if the water in the pipe will completely freeze during a cold night.

Assumptions 1 Heat transfer is transient, but can be treated as steady since the water temperature remains constant during freezing. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties of water are constant. 4 The water in the pipe is stationary, and its initial temperature is 0°C. 5 The convection resistance inside the pipe is negligible so that the inner surface temperature of the pipe is 0°C.

Properties The thermal conductivity of the pipe is given to be $k = 0.16 \text{ W/m} \cdot ^\circ\text{C}$. The density and latent heat of fusion of water at 0°C are $\rho = 1000 \text{ kg/m}^3$ and $h_{if} = 333.7 \text{ kJ/kg}$ (Table A-15).

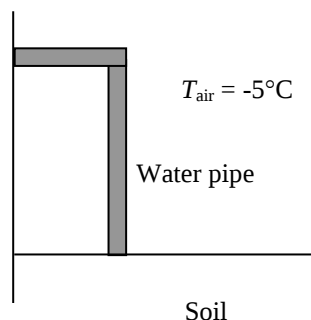
Analysis We assume the inner surface of the pipe to be at 0°C at all times. The thermal resistances involved and the rate of heat transfer are

$$R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(1.2 / 1)}{2\pi (0.16 \text{ W/m} \cdot ^\circ\text{C})(0.5 \text{ m})} = 0.3627 ^\circ\text{C/W}$$

$$R_{\text{conv,o}} = \frac{1}{h_o A} = \frac{1}{(40 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.024 \text{ m})(0.5 \text{ m})]} = 0.6631 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{conv,o}} = 0.3627 + 0.6631 = 1.0258 ^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[0 - (-5)]^\circ\text{C}}{1.0258 ^\circ\text{C/W}} = 4.874 \text{ W}$$



The total amount of heat lost by the water during a 14-h period that night is

$$Q = \dot{Q}\Delta t = (4.874 \text{ J/s})(14 \times 3600 \text{ s}) = 245.7 \text{ kJ}$$

The amount of heat required to freeze the water in the pipe completely is

$$m = \rho V = \rho \pi r^2 L = (1000 \text{ kg/m}^3)\pi(0.01 \text{ m})^2(0.5 \text{ m}) = 0.157 \text{ kg}$$

$$Q = m h_{fg} = (0.157 \text{ kg})(333.7 \text{ kJ/kg}) = 52.4 \text{ kJ}$$

The water in the pipe will **freeze completely** that night since the amount heat loss is greater than the amount it takes to freeze the water completely ($245.7 > 52.4$).

10-148 The plumbing system of a house involves some section of a plastic pipe exposed to the ambient air. The pipe is initially filled with stationary water at 0°C. It is to be determined if the water in the pipe will completely freeze during a cold night.

Assumptions 1 Heat transfer is transient, but can be treated as steady since the water temperature remains constant during freezing. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal properties of water are constant. 4 The water in the pipe is stationary, and its initial temperature is 0°C. 5 The convection resistance inside the pipe is negligible so that the inner surface temperature of the pipe is 0°C.

Properties The thermal conductivity of the pipe is given to be $k = 0.16 \text{ W/m}\cdot^\circ\text{C}$. The density and latent heat of fusion of water at 0°C are $\rho = 1000 \text{ kg/m}^3$ and $h_{if} = 333.7 \text{ kJ/kg}$ (Table A-15).

Analysis We assume the inner surface of the pipe to be at 0°C at all times. The thermal resistances involved and the rate of heat transfer are

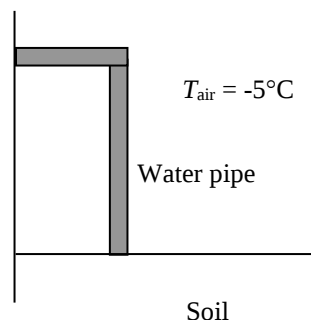
$$R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(1.2 / 1)}{2\pi(0.16 \text{ W/m}\cdot^\circ\text{C})(0.5 \text{ m}^2)} = 0.3627^\circ\text{C/W}$$

$$R_{\text{conv,o}} = \frac{1}{h_o A} = \frac{1}{(10 \text{ W/m}^2\cdot^\circ\text{C})[\pi(0.024 \text{ m})(0.5 \text{ m})]} = 2.6526^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{conv,o}} = 0.3627 + 2.6526 = 3.0153^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[0 - (-5)]^\circ\text{C}}{3.0153^\circ\text{C/W}} = 1.658 \text{ W}$$

$$Q = \dot{Q}\Delta t = (1.658 \text{ J/s})(14 \times 3600 \text{ s}) = 83.57 \text{ kJ}$$



The amount of heat required to freeze the water in the pipe completely is

$$m = \rho V = \rho \pi r^2 L = (1000 \text{ kg/m}^3) \pi (0.01 \text{ m})^2 (0.5 \text{ m}) = 0.157 \text{ kg}$$

$$Q = m h_{fg} = (0.157 \text{ kg})(333.7 \text{ kJ/kg}) = 52.4 \text{ kJ}$$

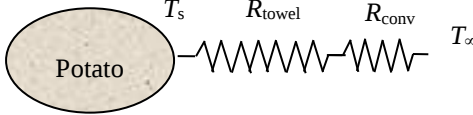
The water in the pipe will **freeze completely** that night since the amount heat loss is greater than the amount it takes to freeze the water completely ($83.57 > 52.4$).

10-149E The surface temperature of a baked potato drops from 300°F to 200°F in 5 minutes in an environment at 70°F. The average heat transfer coefficient and the cooling time of the potato if it is wrapped completely in a towel are to be determined.

Assumptions 1 Thermal properties of potato are constant, and can be taken to be the properties of water. 2 The thermal contact resistance at the interface is negligible. 3 The heat transfer coefficients for wrapped and unwrapped potatoes are the same.

Properties The thermal conductivity of a thick towel is given to be $k = 0.035 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$. We take the properties of potato to be those of water at room temperature, $\rho = 62.2 \text{ lbm/ft}^3$ and $c_p = 0.998 \text{ Btu/lbm}\cdot^\circ\text{F}$.

Analysis This is a transient heat conduction problem, and the rate of heat transfer will decrease as the potato cools down and the temperature difference between the potato and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of $(300+200)/2 = 250^\circ\text{F}$ for the potato during the process. The mass of the potato is

$$\begin{aligned}
 m &= \rho V = \rho \frac{4}{3} \pi r^3 \\
 &= (62.2 \text{ lbm/ft}^3) \frac{4}{3} \pi (1.5/12 \text{ ft})^3 \\
 &= 0.5089 \text{ lbm}
 \end{aligned}$$


The amount of heat lost as the potato is cooled from 300 to 200°F is

$$Q = mc_p \Delta T = (0.5089 \text{ lbm})(0.998 \text{ Btu/lbm}\cdot^\circ\text{F})(300 - 200)^\circ\text{F} = 50.8 \text{ Btu}$$

The rate of heat transfer and the average heat transfer coefficient between the potato and its surroundings are

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{50.8 \text{ Btu}}{(5/60 \text{ h})} = 609.6 \text{ Btu/h}$$

$$\dot{Q} = hA_o(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A_o(T_s - T_\infty)} = \frac{609.6 \text{ Btu/h}}{\pi(3/12 \text{ ft})^2(250 - 70)^\circ\text{F}} = \mathbf{17.2 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}}$$

When the potato is wrapped in a towel, the thermal resistance and heat transfer rate are determined to be

$$R_{\text{towel}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{[(1.5 + 0.12)/12] \text{ ft} - (1.5/12) \text{ ft}}{4\pi(0.035 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})[(1.5 + 0.12)/12] \text{ ft}(1.5/12) \text{ ft}} = 1.3473 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(17.2 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})\pi(3.24/12)^2 \text{ ft}^2} = 0.2539 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{total}} = R_{\text{towel}} + R_{\text{conv}} = 1.3473 + 0.2539 = 1.6012 \text{ h}\cdot^\circ\text{F/Btu}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(250 - 70)^\circ\text{F}}{1.6012 \text{ h}\cdot^\circ\text{F/Btu}} = 112.4 \text{ Btu/h}$$

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{50.8 \text{ Btu}}{112.4 \text{ Btu/h}} = 0.452 \text{ h} = \mathbf{27.1 \text{ min}}$$

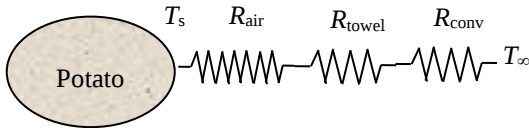
This result is conservative since the heat transfer coefficient will be lower in this case because of the smaller exposed surface temperature.

10-150E The surface temperature of a baked potato drops from 300°F to 200°F in 5 minutes in an environment at 70°F. The average heat transfer coefficient and the cooling time of the potato if it is loosely wrapped completely in a towel are to be determined.

Assumptions 1 Thermal properties of potato are constant, and can be taken to be the properties of water. 2 The heat transfer coefficients for wrapped and unwrapped potatoes are the same.

Properties The thermal conductivity of a thick towel is given to be $k = 0.035 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$. The thermal conductivity of air is given to be $k = 0.015 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$. We take the properties of potato to be those of water at room temperature, $\rho = 62.2 \text{ lbm/ft}^3$ and $c_p = 0.998 \text{ Btu/lbm}\cdot^\circ\text{F}$.

Analysis This is a transient heat conduction problem, and the rate of heat transfer will decrease as the potato cools down and the temperature difference between the potato and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of $(300+200)/2 = 250^\circ\text{F}$ for the potato during the process. The mass of the potato is

$$\begin{aligned}
 m &= \rho V = \rho \frac{4}{3} \pi r^3 \\
 &= (62.2 \text{ lbm/ft}^3) \frac{4}{3} \pi (1.5/12 \text{ ft})^3 \\
 &= 0.5089 \text{ lbm}
 \end{aligned}$$


The amount of heat lost as the potato is cooled from 300 to 200°F is

$$Q = mc_p \Delta T = (0.5089 \text{ lbm})(0.998 \text{ Btu/lbm}\cdot^\circ\text{F})(300 - 200)^\circ\text{F} = 50.8 \text{ Btu}$$

The rate of heat transfer and the average heat transfer coefficient between the potato and its surroundings are

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{50.8 \text{ Btu}}{(5/60 \text{ h})} = 609.6 \text{ Btu/h}$$

$$\dot{Q} = hA_o(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A_o(T_s - T_\infty)} = \frac{609.6 \text{ Btu/h}}{\pi(3/12 \text{ ft})^2(250 - 70)^\circ\text{F}} = \mathbf{17.2 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}}$$

When the potato is wrapped in a towel, the thermal resistance and heat transfer rate are determined to be

$$R_{\text{air}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{[(1.50 + 0.02)/12] \text{ ft} - (1.50/12) \text{ ft}}{4\pi(0.015 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})[(1.50 + 0.02)/12] \text{ ft}(1.50/12) \text{ ft}} = 0.5584 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{towel}} = \frac{r_3 - r_2}{4\pi k r_2 r_3} = \frac{[(1.52 + 0.12)/12] \text{ ft} - (1.52/12) \text{ ft}}{4\pi(0.035 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})[(1.52 + 0.12)/12] \text{ ft}(1.52/12) \text{ ft}} = 1.3134 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(17.2 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})\pi(3.28/12)^2 \text{ ft}^2} = 0.2477 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{\text{total}} = R_{\text{air}} + R_{\text{towel}} + R_{\text{conv}} = 0.5584 + 1.3134 + 0.2477 = 2.1195 \text{ h}\cdot^\circ\text{F/Btu}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(250 - 70)^\circ\text{F}}{2.1195 \text{ h}\cdot^\circ\text{F/Btu}} = 84.9 \text{ Btu/h}$$

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{50.8 \text{ Btu}}{84.9 \text{ Btu/h}} = 0.598 \text{ h} = \mathbf{35.9 \text{ min}}$$

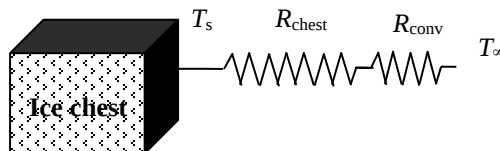
This result is conservative since the heat transfer coefficient will be lower because of the smaller exposed surface temperature.

10-151 An ice chest made of 10-cm thick styrofoam is initially filled with 45 kg of ice at 0°C. The length of time it will take for the ice in the chest to melt completely is to be determined.

Assumptions 1 Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. 2 Heat transfer is one-dimensional. 3 Thermal conductivity is constant. 4 The inner surface temperature of the ice chest can be taken to be 0°C at all times. 5 Heat transfer from the base of the ice chest is negligible.

Properties The thermal conductivity of styrofoam is given to be $k = 0.033 \text{ W/m}\cdot^\circ\text{C}$. The heat of fusion of water at 1 atm is $h_{if} = 333.7 \text{ kJ/kg}$.

Analysis Disregarding any heat loss through the bottom of the ice chest, the total thermal resistance and the heat transfer rate are determined to be



$$A_i = 2(0.3 - 0.03)(0.4 - 0.06) + 2(0.3 - 0.03)(0.5 - 0.06) + (0.4 - 0.06)(0.5 - 0.06) = 0.5708 \text{ m}^2$$

$$A_o = 2(0.3)(0.4) + 2(0.3)(0.5) + (0.4)(0.5) = 0.74 \text{ m}^2$$

$$R_{\text{chest}} = \frac{L}{kA_i} = \frac{0.03 \text{ m}}{(0.033 \text{ W/m}\cdot^\circ\text{C})(0.5708 \text{ m}^2)} = 1.5927^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA_o} = \frac{1}{(18 \text{ W/m}^2\cdot^\circ\text{C})(0.74 \text{ m}^2)} = 0.07508^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{chest}} + R_{\text{conv}} = 1.5927 + 0.07508 = 1.6678^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(28 - 0)^\circ\text{C}}{1.6678^\circ\text{C/W}} = 16.79 \text{ W}$$

The total amount of heat necessary to melt the ice completely is

$$Q = mh_{if} = (50 \text{ kg})(333.7 \text{ kJ/kg}) = 16,685 \text{ kJ}$$

Then the time period to transfer this much heat to the cooler to melt the ice completely becomes

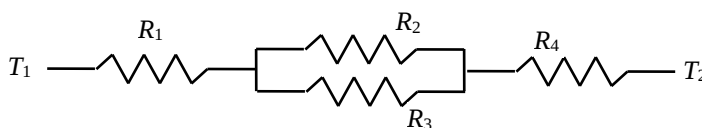
$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16,685,000 \text{ J}}{16.79 \text{ J/s}} = 9.937 \times 10^5 \text{ s} = 276 \text{ h} = \mathbf{11.5 \text{ days}}$$

10-152 A wall is constructed of two large steel plates separated by 1-cm thick steel bars placed 99 cm apart. The remaining space between the steel plates is filled with fiberglass insulation. The rate of heat transfer through the wall is to be determined, and it is to be assessed if the steel bars between the plates can be ignored in heat transfer analysis since they occupy only 1 percent of the heat transfer surface area.

Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2 Heat transfer through the wall can be approximated to be one-dimensional. 3 Thermal conductivities are constant. 4 The surfaces of the wall are maintained at constant temperatures.

Properties The thermal conductivities are given to be $k = 15 \text{ W/m}\cdot^\circ\text{C}$ for steel plates and $k = 0.035 \text{ W/m}\cdot^\circ\text{C}$ for fiberglass insulation.

Analysis We consider 1 m high and 1 m wide portion of the wall which is representative of entire wall. Thermal resistance network and individual resistances are



$$R_1 = R_4 = R_{\text{steel}} = \frac{L}{kA} = \frac{0.02 \text{ m}}{(15 \text{ W/m}\cdot^\circ\text{C})(1 \text{ m}^2)} = 0.00133^\circ\text{C/W}$$

$$R_2 = R_{\text{steel}} = \frac{L}{kA} = \frac{0.2 \text{ m}}{(15 \text{ W/m}\cdot^\circ\text{C})(0.01 \text{ m}^2)} = 1.333^\circ\text{C/W}$$

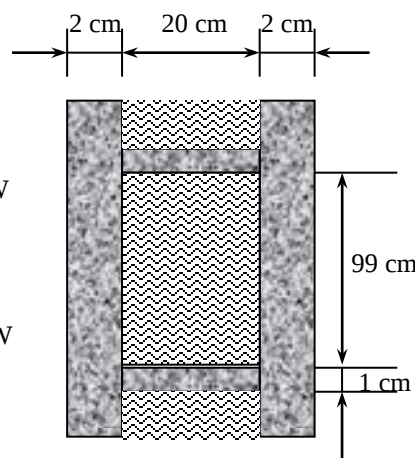
$$R_3 = R_{\text{insulation}} = \frac{L}{kA} = \frac{0.2 \text{ m}}{(0.035 \text{ W/m}\cdot^\circ\text{C})(0.99 \text{ m}^2)} = 5.772^\circ\text{C/W}$$

$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_2} + \frac{1}{R_3} = \frac{1}{1.333} + \frac{1}{5.772} \rightarrow R_{\text{eqv}} = 1.083^\circ\text{C/W}$$

$$R_{\text{total}} = R_1 + R_{\text{eqv}} + R_4 = 0.00133 + 1.083 + 0.00133 = 1.0857^\circ\text{C/W}$$

The rate of heat transfer per m^2 surface area of the wall is

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{22^\circ\text{C}}{1.0857^\circ\text{C/W}} = 20.26 \text{ W}$$



The total rate of heat transfer through the entire wall is then determined to be

$$\dot{Q}_{\text{total}} = (4 \times 6) \dot{Q} = 24(20.26 \text{ W}) = \mathbf{486.2 \text{ W}}$$

If the steel bars were ignored since they constitute only 1% of the wall section, the R_{equiv} would simply be equal to the thermal resistance of the insulation, and the heat transfer rate in this case would be

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{\Delta T}{R_1 + R_{\text{insulation}} + R_4} = \frac{22^\circ\text{C}}{(0.00133 + 5.772 + 0.00133)^\circ\text{C/W}} = 3.81 \text{ W}$$

which is much less than 20.26 W obtained earlier. Therefore, $(20.26 - 3.81)/20.26 = 81.2\%$ of the heat transfer occurs through the steel bars across the wall despite the negligible space that they occupy, and obviously their effect cannot be neglected. The connecting bars are serving as “thermal bridges.”

10-153 A circuit board houses electronic components on one side, dissipating a total of 15 W through the backside of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 20 aluminum fins of rectangular profile on the backside.

Assumptions 1 Steady operating conditions exist. 2 The temperature in the board and along the fins varies in one direction only (normal to the board). 3 All the heat generated in the chips is conducted across the circuit board, and is dissipated from the backside of the board. 4 Heat transfer from the fin tips is negligible. 5 The heat transfer coefficient is constant and uniform over the entire fin surface. 6 The thermal properties of the fins are constant. 7 The heat transfer coefficient accounts for the effect of radiation from the fins.

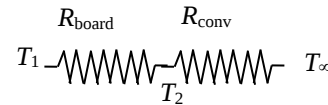
Properties The thermal conductivities are given to be $k = 12 \text{ W/m} \cdot ^\circ\text{C}$ for the circuit board, $k = 237 \text{ W/m} \cdot ^\circ\text{C}$ for the aluminum plate and fins, and $k = 1.8 \text{ W/m} \cdot ^\circ\text{C}$ for the epoxy adhesive.

Analysis (a) The thermal resistance of the board and the convection resistance on the backside of the board are

$$R_{\text{board}} = \frac{L}{kA} = \frac{0.002 \text{ m}}{(12 \text{ W/m} \cdot ^\circ\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 0.011 ^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(45 \text{ W/m} \cdot ^\circ\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 1.481 ^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.011 + 1.481 = 1.492 ^\circ\text{C/W}$$



Then surface temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 37^\circ\text{C} + (15 \text{ W})(1.492 ^\circ\text{C/W}) = \mathbf{59.4^\circ\text{C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 59.4^\circ\text{C} - (15 \text{ W})(0.011 ^\circ\text{C/W}) = \mathbf{59.2^\circ\text{C}}$$

(b) Noting that the cross-sectional areas of the fins are constant, the efficiency of these rectangular fins is determined to be

$$m = \sqrt{\frac{hp}{kA_c}} \cong \sqrt{\frac{h(2w)}{k(tw)}} = \sqrt{\frac{2h}{kt}} = \sqrt{\frac{2(45 \text{ W/m} \cdot ^\circ\text{C})}{(237 \text{ W/m} \cdot ^\circ\text{C})(0.002 \text{ m})}} = 13.78 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(13.78 \text{ m}^{-1} \times 0.02 \text{ m})}{13.78 \text{ m}^{-1} \times 0.02 \text{ m}} = 0.975$$

The finned and unfinned surface areas are

$$A_{\text{finned}} = (20)2w\left(L + \frac{t}{2}\right) = (20)2(0.15)\left(0.02 + \frac{0.002}{2}\right) = 0.126 \text{ m}^2$$

$$A_{\text{unfinned}} = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 \text{ m}^2$$

Then,

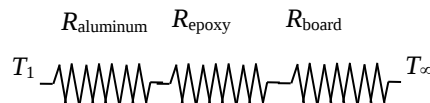
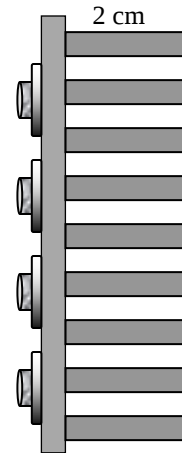
$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}} (T_{\text{base}} - T_\infty)$$

$$\dot{Q}_{\text{unfinned}} = hA_{\text{unfinned}} (T_{\text{base}} - T_\infty)$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{unfinned}} + \dot{Q}_{\text{finned}} = h(T_{\text{base}} - T_\infty)(\eta_{\text{fin}} A_{\text{fin}} + A_{\text{unfinned}})$$

Substituting, the base temperature of the finned surfaces is determined to be

$$\begin{aligned} T_{\text{base}} &= T_\infty + \frac{\dot{Q}_{\text{total}}}{h(\eta_{\text{fin}} A_{\text{fin}} + A_{\text{unfinned}})} \\ &= 37^\circ\text{C} + \frac{15 \text{ W}}{(45 \text{ W/m} \cdot ^\circ\text{C})[(0.975)(0.126 \text{ m}^2) + (0.0090 \text{ m}^2)]} = \mathbf{39.5^\circ\text{C}} \end{aligned}$$



Then the temperatures on both sides of the board are determined using the thermal resistance network to be

$$R_{\text{aluminum}} = \frac{L}{kA} = \frac{0.001 \text{ m}}{(237 \text{ W/m} \cdot ^\circ\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 0.00028 ^\circ\text{C/W}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0003 \text{ m}}{(1.8 \text{ W/m} \cdot ^\circ\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 0.01111 ^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_1 - T_{\text{base}}}{R_{\text{aluminum}} + R_{\text{epoxy}} + R_{\text{board}}} = \frac{(T_1 - 39.5) ^\circ\text{C}}{(0.00028 + 0.01111 + 0.011) ^\circ\text{C/W}}$$

$$\longrightarrow T_1 = 39.5 ^\circ\text{C} + (15 \text{ W})(0.02239 ^\circ\text{C/W}) = \mathbf{39.8 ^\circ\text{C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 39.8 ^\circ\text{C} - (15 \text{ W})(0.011 ^\circ\text{C/W}) = \mathbf{39.6 ^\circ\text{C}}$$

10-154 A circuit board houses electronic components on one side, dissipating a total of 15 W through the backside of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 20 copper fins of rectangular profile on the backside.

Assumptions 1 Steady operating conditions exist. 2 The temperature in the board and along the fins varies in one direction only (normal to the board). 3 All the heat generated in the chips is conducted across the circuit board, and is dissipated from the backside of the board. 4 Heat transfer from the fin tips is negligible. 5 The heat transfer coefficient is constant and uniform over the entire fin surface. 6 The thermal properties of the fins are constant. 7 The heat transfer coefficient accounts for the effect of radiation from the fins.

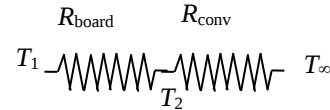
Properties The thermal conductivities are given to be $k = 12 \text{ W/m}\cdot^\circ\text{C}$ for the circuit board, $k = 386 \text{ W/m}\cdot^\circ\text{C}$ for the copper plate and fins, and $k = 1.8 \text{ W/m}\cdot^\circ\text{C}$ for the epoxy adhesive.

Analysis (a) The thermal resistance of the board and the convection resistance on the backside of the board are

$$R_{\text{board}} = \frac{L}{kA} = \frac{0.002 \text{ m}}{(12 \text{ W/m}\cdot^\circ\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 0.011^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(45 \text{ W/m}\cdot^\circ\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 1.481^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.011 + 1.481 = 1.492^\circ\text{C/W}$$



Then surface temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 37^\circ\text{C} + (15 \text{ W})(1.492^\circ\text{C/W}) = \mathbf{59.4^\circ\text{C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 59.4^\circ\text{C} - (15 \text{ W})(0.011^\circ\text{C/W}) = \mathbf{59.2^\circ\text{C}}$$

(b) Noting that the cross-sectional areas of the fins are constant, the efficiency of these rectangular fins is determined to be

$$m = \sqrt{\frac{hp}{kA_c}} \cong \sqrt{\frac{h(2w)}{k(tw)}} = \sqrt{\frac{2h}{kt}} = \sqrt{\frac{2(45 \text{ W/m}\cdot^\circ\text{C})}{(386 \text{ W/m}\cdot^\circ\text{C})(0.002 \text{ m})}} = 10.80 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(10.80 \text{ m}^{-1} \times 0.02 \text{ m})}{10.80 \text{ m}^{-1} \times 0.02 \text{ m}} = 0.985$$

The finned and unfinned surface areas are

$$A_{\text{finned}} = (20)2w\left(L + \frac{t}{2}\right) = (20)2(0.15)\left(0.02 + \frac{0.002}{2}\right) = 0.126 \text{ m}^2$$

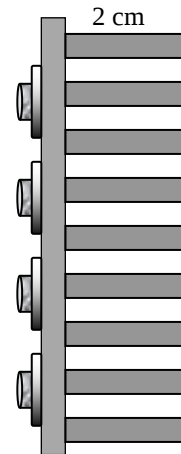
$$A_{\text{unfinned}} = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 \text{ m}^2$$

Then,

$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}} (T_{\text{base}} - T_\infty)$$

$$\dot{Q}_{\text{unfinned}} = hA_{\text{unfinned}} (T_{\text{base}} - T_\infty)$$

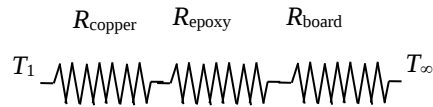
$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{unfinned}} + \dot{Q}_{\text{finned}} = h(T_{\text{base}} - T_\infty)(\eta_{\text{fin}} A_{\text{fin}} + A_{\text{unfinned}})$$



Substituting, the base temperature of the finned surfaces determine to be

$$\begin{aligned}
 T_{\text{base}} &= T_{\infty} + \frac{\dot{Q}_{\text{total}}}{h(\eta_{\text{fin}} A_{\text{fin}} + A_{\text{unfinned}})} \\
 &= 37^{\circ}\text{C} + \frac{15 \text{ W}}{(45 \text{ W/m}^2 \cdot ^{\circ}\text{C})[(0.985)(0.126 \text{ m}^2) + (0.0090 \text{ m}^2)]} = \mathbf{39.5^{\circ}\text{C}}
 \end{aligned}$$

Then the temperatures on both sides of the board are determined using the thermal resistance network to be



$$R_{\text{copper}} = \frac{L}{kA} = \frac{0.001 \text{ m}}{(386 \text{ W/m} \cdot ^{\circ}\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 0.00017^{\circ}\text{C/W}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0003 \text{ m}}{(1.8 \text{ W/m} \cdot ^{\circ}\text{C})(0.1 \text{ m})(0.15 \text{ m})} = 0.01111^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_1 - T_{\text{base}}}{R_{\text{copper}} + R_{\text{epoxy}} + R_{\text{board}}} = \frac{(T_1 - 39.5)^{\circ}\text{C}}{(0.00017 + 0.01111 + 0.011)^{\circ}\text{C/W}}$$

$$\longrightarrow T_1 = 39.5^{\circ}\text{C} + (15 \text{ W})(0.02228^{\circ}\text{C/W}) = \mathbf{39.8^{\circ}\text{C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 39.8^{\circ}\text{C} - (15 \text{ W})(0.011^{\circ}\text{C/W}) = \mathbf{39.6^{\circ}\text{C}}$$

10-155 Steam passes through a row of 10 parallel pipes placed horizontally in a concrete floor exposed to room air at 20°C with a heat transfer coefficient of 12 W/m²·°C. If the surface temperature of the concrete floor is not to exceed 35°C, the minimum burial depth of the steam pipes below the floor surface is to be determined.

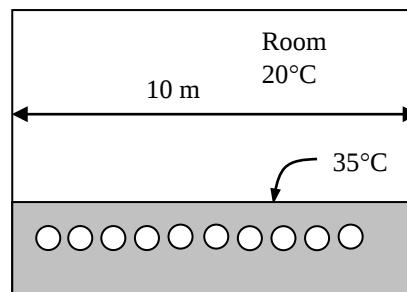
Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the concrete is constant.

Properties The thermal conductivity of concrete is given to be $k = 0.75 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis In steady operation, the rate of heat loss from the steam through the concrete floor by conduction must be equal to the rate of heat transfer from the concrete floor to the room by combined convection and radiation, which is determined to be

$$\begin{aligned}\dot{Q} &= hA_s(T_s - T_\infty) \\ &= (12 \text{ W/m}^2 \cdot ^\circ\text{C})[(10 \text{ m})(5 \text{ m})](35 - 20)^\circ\text{C} = 9000 \text{ W}\end{aligned}$$

Then the depth the steam pipes should be buried can be determined with the aid of shape factor for this configuration from Table 10-7 to be



$$\dot{Q} = nSk(T_1 - T_2) \longrightarrow S = \frac{\dot{Q}}{nk(T_1 - T_2)} = \frac{9000 \text{ W}}{10(0.75 \text{ W/m} \cdot ^\circ\text{C})(145 - 35)^\circ\text{C}} = 10.91 \text{ m (per pipe)}$$

$$w = \frac{a}{n} = \frac{10 \text{ m}}{10} = 1 \text{ m (center - to - center distance of pipes)}$$

$$S = \frac{2\pi L}{\ln\left(\frac{2w}{\pi D} \sinh \frac{2\pi z}{w}\right)}$$

$$10.91 \text{ m} = \frac{2\pi(5 \text{ m})}{\ln\left[\frac{2(1 \text{ m})}{\pi(0.06 \text{ m})} \sinh \frac{2\pi z}{(1 \text{ m})}\right]} \longrightarrow z = 0.205 \text{ m} = \mathbf{20.5 \text{ cm}}$$

10-156 Two persons are wearing different clothes made of different materials with different surface areas. The fractions of heat lost from each person's body by perspiration are to be determined.

Assumptions 1 Heat transfer is steady. 2 Heat transfer is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer by radiation is accounted for in the heat transfer coefficient. 5 The human body is assumed to be cylindrical in shape for heat transfer purposes.

Properties The thermal conductivities of the leather and synthetic fabric are given to be $k = 0.159 \text{ W/m}\cdot^\circ\text{C}$ and $k = 0.13 \text{ W/m}\cdot^\circ\text{C}$, respectively.

Analysis The surface area of each body is first determined from

$$A_1 = \pi DL / 2 = \pi(0.25 \text{ m})(1.7 \text{ m})/2 = 0.6675 \text{ m}^2$$

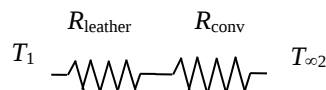
$$A_2 = 2A_1 = 2 \times 0.6675 = 1.335 \text{ m}^2$$

The sensible heat lost from the first person's body is

$$R_{\text{leather}} = \frac{L}{kA} = \frac{0.001 \text{ m}}{(0.159 \text{ W/m}\cdot^\circ\text{C})(0.6675 \text{ m}^2)} = 0.00942 \text{ }^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15 \text{ W/m}^2\cdot^\circ\text{C})(0.6675 \text{ m}^2)} = 0.09988 \text{ }^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00942 + 0.09988 = 0.10930 \text{ }^\circ\text{C/W}$$



The total sensible heat transfer is the sum of heat transferred through the clothes and the skin

$$\dot{Q}_{\text{clothes}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32 - 30)^\circ\text{C}}{0.10930 \text{ }^\circ\text{C/W}} = 18.3 \text{ W}$$

$$\dot{Q}_{\text{skin}} = \frac{T_1 - T_{\infty 2}}{R_{\text{conv}}} = \frac{(32 - 30)^\circ\text{C}}{0.09988 \text{ }^\circ\text{C/W}} = 20.0 \text{ W}$$

$$\dot{Q}_{\text{sensible}} = \dot{Q}_{\text{clothes}} + \dot{Q}_{\text{skin}} = 18.3 + 20 = 38.3 \text{ W}$$

Then the fraction of heat lost by respiration becomes

$$f = \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60 - 38.3}{60} = \mathbf{0.362}$$

Repeating similar calculations for the second person's body

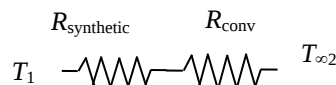
$$R_{\text{synthetic}} = \frac{L}{kA} = \frac{0.001 \text{ m}}{(0.13 \text{ W/m}\cdot^\circ\text{C})(1.335 \text{ m}^2)} = 0.00576 \text{ }^\circ\text{C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15 \text{ W/m}^2\cdot^\circ\text{C})(1.335 \text{ m}^2)} = 0.04994 \text{ }^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00576 + 0.04994 = 0.05570 \text{ }^\circ\text{C/W}$$

$$\dot{Q}_{\text{sensible}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32 - 30)^\circ\text{C}}{0.05570 \text{ }^\circ\text{C/W}} = 35.9 \text{ W}$$

$$f = \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60 - 35.9}{60} = \mathbf{0.402}$$



10-157 A wall constructed of three layers is considered. The rate of heat transfer through the wall and temperature drops across the plaster, brick, covering, and surface-ambient air are to be determined.

Assumptions 1 Heat transfer is steady. 2 Heat transfer is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer by radiation is accounted for in the heat transfer coefficient.

Properties The thermal conductivities of the plaster, brick, and covering are given to be $k = 0.72 \text{ W/m}\cdot^\circ\text{C}$, $k = 0.36 \text{ W/m}\cdot^\circ\text{C}$, $k = 1.40 \text{ W/m}\cdot^\circ\text{C}$, respectively.

Analysis The surface area of the wall and the individual resistances are

$$A = (6 \text{ m}) \times (2.8 \text{ m}) = 16.8 \text{ m}^2$$

$$R_1 = R_{\text{plaster}} = \frac{L_1}{k_1 A} = \frac{0.01 \text{ m}}{(0.36 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.00165^\circ\text{C/W}$$

$$R_2 = R_{\text{brick}} = \frac{L_2}{k_2 A} = \frac{0.20 \text{ m}}{(0.72 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.01653^\circ\text{C/W}$$

$$R_3 = R_{\text{covering}} = \frac{L_3}{k_3 A} = \frac{0.02 \text{ m}}{(1.4 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.00085^\circ\text{C/W}$$

$$R_o = R_{\text{conv},2} = \frac{1}{h_2 A} = \frac{1}{(17 \text{ W/m}^2\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.00350^\circ\text{C/W}$$

$$R_{\text{total}} = R_1 + R_2 + R_3 + R_{\text{conv},2}$$

$$= 0.00165 + 0.01653 + 0.00085 + 0.00350 = 0.02253^\circ\text{C/W}$$

The steady rate of heat transfer through the wall then becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(23 - 8)^\circ\text{C}}{0.02253^\circ\text{C/W}} = \mathbf{665.8 \text{ W}}$$

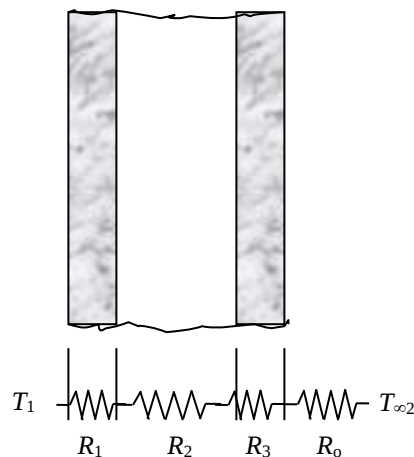
The temperature drops are

$$\Delta T_{\text{plaster}} = \dot{Q} R_{\text{plaster}} = (665.8 \text{ W})(0.00165^\circ\text{C/W}) = \mathbf{1.1^\circ\text{C}}$$

$$\Delta T_{\text{brick}} = \dot{Q} R_{\text{brick}} = (665.8 \text{ W})(0.01653^\circ\text{C/W}) = \mathbf{11.0^\circ\text{C}}$$

$$\Delta T_{\text{covering}} = \dot{Q} R_{\text{covering}} = (665.8 \text{ W})(0.00085^\circ\text{C/W}) = \mathbf{0.6^\circ\text{C}}$$

$$\Delta T_{\text{conv}} = \dot{Q} R_{\text{conv}} = (665.8 \text{ W})(0.00350^\circ\text{C/W}) = \mathbf{2.3^\circ\text{C}}$$



10-158 An insulation is to be added to a wall to decrease the heat loss by 85%. The thickness of insulation and the outer surface temperature of the wall are to be determined for two different insulating materials.

Assumptions 1 Heat transfer is steady. 2 Heat transfer is one-dimensional. 3 Thermal conductivities are constant. 4 Heat transfer by radiation is accounted for in the heat transfer coefficient.

Properties The thermal conductivities of the plaster, brick, covering, polyurethane foam, and glass fiber are given to be $0.72 \text{ W/m}\cdot^\circ\text{C}$, $0.36 \text{ W/m}\cdot^\circ\text{C}$, $1.40 \text{ W/m}\cdot^\circ\text{C}$, $0.025 \text{ W/m}\cdot^\circ\text{C}$, $0.036 \text{ W/m}\cdot^\circ\text{C}$, respectively.

Analysis The surface area of the wall and the individual resistances are

$$A = (6 \text{ m}) \times (2.8 \text{ m}) = 16.8 \text{ m}^2$$

$$R_1 = R_{\text{plaster}} = \frac{L_1}{k_1 A} = \frac{0.01 \text{ m}}{(0.36 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.00165^\circ\text{C/W}$$

$$R_2 = R_{\text{brick}} = \frac{L_2}{k_2 A} = \frac{0.20 \text{ m}}{(0.72 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.01653^\circ\text{C/W}$$

$$R_3 = R_{\text{covering}} = \frac{L_3}{k_3 A} = \frac{0.02 \text{ m}}{(1.4 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.00085^\circ\text{C/W}$$

$$R_o = R_{\text{conv},2} = \frac{1}{h_2 A} = \frac{1}{(17 \text{ W/m}^2\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.00350^\circ\text{C/W}$$

$$\begin{aligned} R_{\text{total, no ins}} &= R_1 + R_2 + R_3 + R_{\text{conv},2} \\ &= 0.00165 + 0.01653 + 0.00085 + 0.00350 \\ &= 0.02253^\circ\text{C/W} \end{aligned}$$

The rate of heat loss without the insulation is

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total, no ins}}} = \frac{(23 - 8)^\circ\text{C}}{0.02253^\circ\text{C/W}} = 666 \text{ W}$$

(a) The rate of heat transfer after insulation is

$$\dot{Q}_{\text{ins}} = 0.15 \dot{Q}_{\text{no ins}} = 0.15 \times 666 = 99.9 \text{ W}$$

The total thermal resistance with the foam insulation is

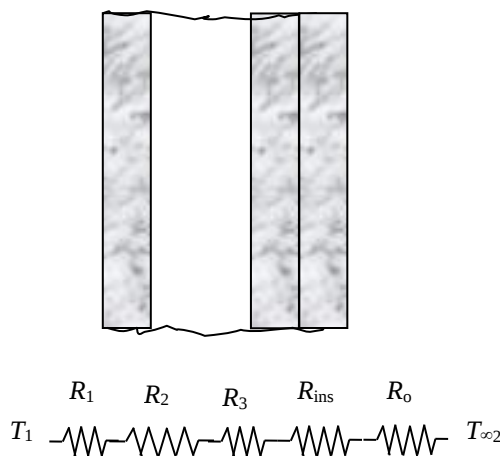
$$\begin{aligned} R_{\text{total}} &= R_1 + R_2 + R_3 + R_{\text{foam}} + R_{\text{conv},2} \\ &= 0.02253^\circ\text{C/W} + \frac{L_4}{(0.025 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} \\ &= 0.02253^\circ\text{C/W} + \frac{L_4}{(0.42 \text{ W}\cdot\text{m}/^\circ\text{C})} \end{aligned}$$

The thickness of insulation is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \rightarrow 99.9 \text{ W} = \frac{(23 - 8)^\circ\text{C}}{0.02253^\circ\text{C/W} + \frac{L_4}{(0.42 \text{ W}\cdot\text{m}/^\circ\text{C})}} \rightarrow L_4 = \mathbf{0.054 \text{ m} = 5.4 \text{ cm}}$$

The outer surface temperature of the wall is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_2 - T_{\infty 2}}{R_{\text{conv}}} \rightarrow 99.9 \text{ W} = \frac{(T_2 - 8)^\circ\text{C}}{0.00350^\circ\text{C/W}} \rightarrow T_2 = \mathbf{8.3^\circ\text{C}}$$



(b) The total thermal resistance with the fiberglass insulation is

$$R_{\text{total}} = R_1 + R_2 + R_3 + R_{\text{fiber glass}} + R_{\text{conv},2}$$

$$= 0.02253^\circ\text{C/W} + \frac{L_4}{(0.036 \text{ W/m}\cdot^\circ\text{C})(16.8 \text{ m}^2)} = 0.02253^\circ\text{C/W} + \frac{L_4}{(0.6048 \text{ W}\cdot\text{m}/^\circ\text{C})}$$

The thickness of insulation is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow 99.9 \text{ W} = \frac{(23 - 8)^\circ\text{C}}{0.02253^\circ\text{C/W} + \frac{L_4}{(0.6048 \text{ W}\cdot\text{m}/^\circ\text{C})}} \longrightarrow L_4 = \mathbf{0.077 \text{ m} = 7.7 \text{ cm}}$$

The outer surface temperature of the wall is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_2 - T_{\infty 2}}{R_{\text{conv}}} \longrightarrow 99.9 = \frac{(T_2 - 8)^\circ\text{C}}{0.00350^\circ\text{C/W}} \longrightarrow T_2 = \mathbf{8.3^\circ\text{C}}$$

Discussion The outer surface temperature is same for both cases since the rate of heat transfer does not change.

10-159 Cold conditioned air is flowing inside a duct of square cross-section. The maximum length of the duct for a specified temperature increase in the duct is to be determined.

Assumptions 1 Heat transfer is steady. 2 Heat transfer is one-dimensional. 3 Thermal conductivities are constant. 4 Steady one-dimensional heat conduction relations can be used due to small thickness of the duct wall. 5 When calculating the conduction thermal resistance of aluminum, the average of inner and outer surface areas will be used.

Properties The thermal conductivity of aluminum is given to be $237 \text{ W/m}\cdot^\circ\text{C}$. The specific heat of air at the given temperature is $c_p = 1006 \text{ J/kg}\cdot^\circ\text{C}$ (Table A-22).

Analysis The inner and the outer surface areas of the duct per unit length and the individual thermal resistances are

$$A_1 = 4a_1 L = 4(0.22 \text{ m})(1 \text{ m}) = 0.88 \text{ m}^2$$

$$A_2 = 4a_2 L = 4(0.25 \text{ m})(1 \text{ m}) = 1.0 \text{ m}^2$$

$$R_i = \frac{1}{h_1 A} = \frac{1}{(75 \text{ W/m}^2\cdot^\circ\text{C})(0.88 \text{ m}^2)} = 0.01515^\circ\text{C/W}$$

$$R_{\text{alum}} = \frac{L}{kA} = \frac{0.015 \text{ m}}{(237 \text{ W/m}\cdot^\circ\text{C})[(0.88 + 1)/2] \text{ m}^2} = 0.00007^\circ\text{C/W}$$

$$R_o = \frac{1}{h_2 A} = \frac{1}{(13 \text{ W/m}^2\cdot^\circ\text{C})(1.0 \text{ m}^2)} = 0.07692^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{alum}} + R_o = 0.01515 + 0.00007 + 0.07692 = 0.09214^\circ\text{C/W}$$

The rate of heat loss from the air inside the duct is

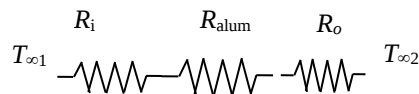
$$\dot{Q} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total}}} = \frac{(33 - 12)^\circ\text{C}}{0.09214^\circ\text{C/W}} = 228 \text{ W}$$

For a temperature rise of 1°C , the air inside the duct should gain heat at a rate of

$$\dot{Q}_{\text{total}} = \dot{m} c_p \Delta T = (0.8 \text{ kg/s})(1006 \text{ J/kg}\cdot^\circ\text{C})(1^\circ\text{C}) = 805 \text{ W}$$

Then the maximum length of the duct becomes

$$L = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{805 \text{ W}}{228 \text{ W}} = \mathbf{3.53 \text{ m}}$$

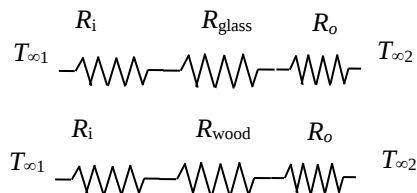


10-160 Heat transfer through a window is considered. The percent error involved in the calculation of heat gain through the window assuming the window consist of glass only is to be determined.

Assumptions 1 Heat transfer is steady. 2 Heat transfer is one-dimensional. 3 Thermal conductivities are constant. 4 Radiation is accounted for in heat transfer coefficients.

Properties The thermal conductivities are given to be $0.7 \text{ W/m}\cdot^\circ\text{C}$ for glass and $0.12 \text{ W/m}\cdot^\circ\text{C}$ for pine wood.

Analysis The surface areas of the glass and the wood and the individual thermal resistances are



$$A_{\text{glass}} = 0.85(1.5 \text{ m})(2 \text{ m}) = 2.55 \text{ m}^2 \quad A_{\text{wood}} = 0.15(1.5 \text{ m})(2 \text{ m}) = 0.45 \text{ m}^2$$

$$R_{i,\text{glass}} = \frac{1}{h_1 A_{\text{glass}}} = \frac{1}{(7 \text{ W/m}^2\cdot^\circ\text{C})(2.55 \text{ m}^2)} = 0.05602^\circ\text{C/W}$$

$$R_{i,\text{wood}} = \frac{1}{h_1 A_{\text{wood}}} = \frac{1}{(7 \text{ W/m}^2\cdot^\circ\text{C})(0.45 \text{ m}^2)} = 0.31746^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{k_{\text{glass}} A_{\text{glass}}} = \frac{0.003 \text{ m}}{(0.7 \text{ W/m}\cdot^\circ\text{C})(2.55 \text{ m}^2)} = 0.00168^\circ\text{C/W}$$

$$R_{\text{wood}} = \frac{L_{\text{wood}}}{k_{\text{wood}} A_{\text{wood}}} = \frac{0.05 \text{ m}}{(0.12 \text{ W/m}\cdot^\circ\text{C})(0.45 \text{ m}^2)} = 0.92593^\circ\text{C/W}$$

$$R_{o,\text{glass}} = \frac{1}{h_2 A_{\text{glass}}} = \frac{1}{(13 \text{ W/m}^2\cdot^\circ\text{C})(2.55 \text{ m}^2)} = 0.03017^\circ\text{C/W}$$

$$R_{o,\text{wood}} = \frac{1}{h_2 A_{\text{wood}}} = \frac{1}{(13 \text{ W/m}^2\cdot^\circ\text{C})(0.45 \text{ m}^2)} = 0.17094^\circ\text{C/W}$$

$$R_{\text{total,glass}} = R_{i,\text{glass}} + R_{\text{glass}} + R_{o,\text{glass}} = 0.05602 + 0.00168 + 0.03017 = 0.08787^\circ\text{C/W}$$

$$R_{\text{total,wood}} = R_{i,\text{wood}} + R_{\text{wood}} + R_{o,\text{wood}} = 0.31746 + 0.92593 + 0.17094 = 1.41433^\circ\text{C/W}$$

The rate of heat gain through the glass and the wood and their total are

$$\dot{Q}_{\text{glass}} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total,glass}}} = \frac{(40 - 24)^\circ\text{C}}{0.08787^\circ\text{C/W}} = 182.1 \text{ W} \quad \dot{Q}_{\text{wood}} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total,wood}}} = \frac{(40 - 24)^\circ\text{C}}{1.41433^\circ\text{C/W}} = 11.3 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{glass}} + \dot{Q}_{\text{wood}} = 182.1 + 11.3 = 193.4 \text{ W}$$

If the window consists of glass only the heat gain through the window is

$$A_{\text{glass}} = (1.5 \text{ m})(2 \text{ m}) = 3.0 \text{ m}^2$$

$$R_{i,\text{glass}} = \frac{1}{h_1 A_{\text{glass}}} = \frac{1}{(7 \text{ W/m}^2\cdot^\circ\text{C})(3.0 \text{ m}^2)} = 0.04762^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{k_{\text{glass}} A_{\text{glass}}} = \frac{0.003 \text{ m}}{(0.7 \text{ W/m}\cdot^\circ\text{C})(3.0 \text{ m}^2)} = 0.00143^\circ\text{C/W}$$

$$R_{o,\text{glass}} = \frac{1}{h_2 A_{\text{glass}}} = \frac{1}{(13 \text{ W/m}^2\cdot^\circ\text{C})(3.0 \text{ m}^2)} = 0.02564^\circ\text{C/W}$$

$$R_{\text{total,glass}} = R_{i,\text{glass}} + R_{\text{glass}} + R_{o,\text{glass}} = 0.04762 + 0.00143 + 0.02564 = 0.07469^\circ\text{C/W}$$

$$\dot{Q}_{\text{glass}} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total,glass}}} = \frac{(40 - 24)^\circ\text{C}}{0.07469^\circ\text{C/W}} = 214.2 \text{ W}$$

Then the percentage error involved in heat gain through the window assuming the window consist of glass only becomes

$$\% \text{ Error} = \frac{\dot{Q}_{\text{glass only}} - \dot{Q}_{\text{with wood}}}{\dot{Q}_{\text{with wood}}} = \frac{214.2 - 193.4}{193.4} \times 100 = \mathbf{10.8\%}$$

10-161 Steam is flowing inside a steel pipe. The thickness of the insulation needed to reduce the heat loss by 95 percent and the thickness of the insulation needed to reduce outer surface temperature to 40°C are to be determined.

Assumptions 1 Heat transfer is steady since there is no indication of any change with time. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. 3 Thermal conductivities are constant. 4 The thermal contact resistance at the interface is negligible.

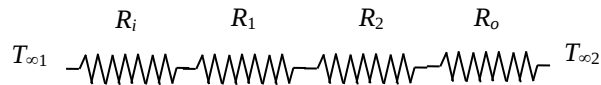
Properties The thermal conductivities are given to be $k = 61 \text{ W/m}\cdot^\circ\text{C}$ for steel and $k = 0.038 \text{ W/m}\cdot^\circ\text{C}$ for insulation.

Analysis (a) Considering a unit length of the pipe, the inner and the outer surface areas of the pipe and the insulation are

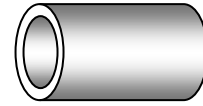
$$A_i = \pi D_i L = \pi(0.10 \text{ m})(1 \text{ m}) = 0.3142 \text{ m}^2$$

$$A_2 = \pi D_o L = \pi(0.12 \text{ m})(1 \text{ m}) = 0.3770 \text{ m}^2$$

$$A_3 = \pi D_3 L = \pi D_3 (1 \text{ m}) = 3.1416 D_3 \text{ m}^2$$



The individual thermal resistances are



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(105 \text{ W/m}^2 \cdot ^\circ\text{C})(0.3142 \text{ m}^2)} = 0.03031 ^\circ\text{C/W}$$

$$R_1 = R_{\text{pipe}} = \frac{\ln(r_2 / r_1)}{2\pi k_1 L} = \frac{\ln(6 / 5)}{2\pi(61 \text{ W/m}\cdot^\circ\text{C})(1 \text{ m})} = 0.00048 ^\circ\text{C/W}$$

$$R_2 = R_{\text{insulation}} = \frac{\ln(r_3 / r_2)}{2\pi k_2 L} = \frac{\ln(D_3 / 0.12)}{2\pi(0.038 \text{ W/m}\cdot^\circ\text{C})(1 \text{ m})} = \frac{\ln(D_3 / 0.12)}{0.23876} ^\circ\text{C/W}$$

$$R_{o,\text{steel}} = \frac{1}{h_o A_o} = \frac{1}{(14 \text{ W/m}^2 \cdot ^\circ\text{C})(0.3770 \text{ m}^2)} = 0.18947 ^\circ\text{C/W}$$

$$R_{o,\text{insulation}} = \frac{1}{h_o A_o} = \frac{1}{(14 \text{ W/m}^2 \cdot ^\circ\text{C})(3.1416 D_3 \text{ m}^2)} = \frac{0.02274}{D_3} ^\circ\text{C/W}$$

$$R_{\text{total, no insulation}} = R_i + R_1 + R_{o,\text{steel}} = 0.03031 + 0.00048 + 0.18947 = 0.22026 ^\circ\text{C/W}$$

$$\begin{aligned} R_{\text{total, insulation}} &= R_i + R_1 + R_2 + R_{o,\text{insulation}} = 0.03031 + 0.00048 + \frac{\ln(D_3 / 0.12)}{0.23876} + \frac{0.02274}{D_3} \\ &= 0.03079 + \frac{\ln(D_3 / 0.12)}{0.23876} + \frac{0.02274}{D_3} ^\circ\text{C/W} \end{aligned}$$

Then the steady rate of heat loss from the steam per meter pipe length for the case of no insulation becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(235 - 20)^\circ\text{C}}{0.22026 ^\circ\text{C/W}} = 976.1 \text{ W}$$

The thickness of the insulation needed in order to save 95 percent of this heat loss can be determined from

$$\dot{Q}_{\text{insulation}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total, insulation}}} \longrightarrow (0.05 \times 976.1) \text{ W} = \frac{(235 - 20)^\circ\text{C}}{\left(0.03079 + \frac{\ln(D_3 / 0.12)}{0.23876} + \frac{0.02274}{D_3}\right) ^\circ\text{C/W}}$$

whose solution is

$$D_3 = 0.3355 \text{ m} \longrightarrow \text{thickness} = \frac{D_3 - D_2}{2} = \frac{33.55 - 12}{2} = \mathbf{10.78 \text{ cm}}$$

(b) The thickness of the insulation needed that would maintain the outer surface of the insulation at a maximum temperature of 40°C can be determined from

$$\dot{Q}_{\text{insulation}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total, insulation}}} = \frac{T_2 - T_{\infty 2}}{R_{\text{o, insulation}}}$$

$$\rightarrow \frac{(235 - 20)^{\circ}\text{C}}{\left(0.03079 + \frac{\ln(D_3 / 0.12)}{0.23876} + \frac{0.02274}{D_3}\right)^{\circ}\text{C/W}} = \frac{(40 - 20)^{\circ}\text{C}}{\frac{0.02274}{D_3}^{\circ}\text{C/W}}$$

whose solution is

$$D_3 = 0.1644 \text{ m} \longrightarrow \text{thickness} = \frac{D_3 - D_2}{2} = \frac{16.44 - 12}{2} = \mathbf{2.22 \text{ cm}}$$

10-162 A 6-m-diameter spherical tank filled with liquefied natural gas (LNG) at -160°C is exposed to ambient air. The time for the LNG temperature to rise to -150°C is to be determined.

Assumptions 1 Heat transfer can be considered to be steady since the specified thermal conditions at the boundaries do not change with time significantly. 2 Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. 3 Radiation is accounted for in the combined heat transfer coefficient. 3 The combined heat transfer coefficient is constant and uniform over the entire surface. 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the LNG inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

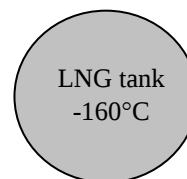
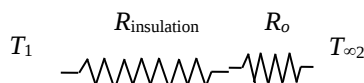
Properties The density and specific heat of LNG are given to be 425 kg/m^3 and $3.475\text{ kJ/kg}\cdot^{\circ}\text{C}$, respectively. The thermal conductivity of super insulation is given to be $k = 0.00008\text{ W/m}\cdot^{\circ}\text{C}$.

Analysis The inner and outer surface areas of the insulated tank and the volume of the LNG are

$$A_1 = \pi D_1^2 = \pi(4\text{ m})^2 = 50.27\text{ m}^2$$

$$A_2 = \pi D_2^2 = \pi(4.10\text{ m})^2 = 52.81\text{ m}^2$$

$$\mathcal{V}_1 = \pi D_1^3 / 6 = \pi(4\text{ m})^3 / 6 = 33.51\text{ m}^3$$



The rate of heat transfer to the LNG is

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(2.05 - 2.0)\text{ m}}{4\pi(0.00008\text{ W/m}\cdot^{\circ}\text{C})(2.0\text{ m})(2.05\text{ m})} = 12.13071^{\circ}\text{C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(22\text{ W/m}^2\cdot^{\circ}\text{C})(52.81\text{ m}^2)} = 0.00086^{\circ}\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{insulation}} = 0.00086 + 12.13071 = 12.13157^{\circ}\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 2} - T_{\text{LNG}}}{R_{\text{total}}} = \frac{[24 - (-155)]^{\circ}\text{C}}{12.13157^{\circ}\text{C/W}} = 14.75\text{ W}$$

We used average LNG temperature in heat transfer rate calculation. The amount of heat transfer to increase the LNG temperature from -160°C to -150°C is

$$m = \rho \mathcal{V}_1 = (425\text{ kg/m}^3)(33.51\text{ m}^3) = 14,242\text{ kg}$$

$$Q = mc_p \Delta T = (14,242\text{ kg})(3.475\text{ kJ/kg}\cdot^{\circ}\text{C})[(-150) - (-160)^{\circ}\text{C}] = 4.95 \times 10^5\text{ kJ}$$

Assuming that heat will be lost from the LNG at an average rate of 15.17 W , the time period for the LNG temperature to rise to -150°C becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{4.95 \times 10^5\text{ kJ}}{0.01475\text{ kW}} = 3.355 \times 10^7\text{ s} = 9320\text{ h} = \mathbf{388\text{ days}}$$

10-163 A hot plate is to be cooled by attaching aluminum fins of square cross section on one side. The number of fins needed to triple the rate of heat transfer is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 Heat transfer from the fin tips is negligible. 4 The heat transfer coefficient is constant and uniform over the entire fin surface. 5 The thermal properties of the fins are constant. 6 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum fins is given to be $k = 237 \text{ W/m}\cdot^\circ\text{C}$.

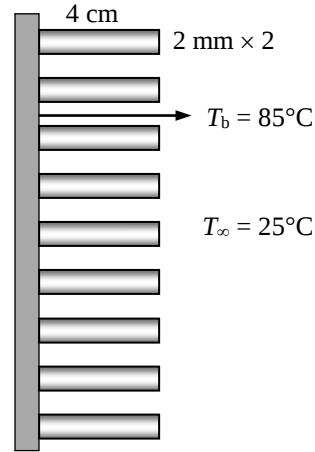
Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the square cross-section fins can be determined to be

$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{4ha}{ka^2}} = \sqrt{\frac{4(20 \text{ W/m}^2\cdot^\circ\text{C})(0.002 \text{ m})}{(237 \text{ W/m}\cdot^\circ\text{C})(0.002 \text{ m})^2}} = 12.99 \text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(12.99 \text{ m}^{-1} \times 0.04 \text{ m})}{12.99 \text{ m}^{-1} \times 0.04 \text{ m}} = 0.919$$

The finned and unfinned surface areas, and heat transfer rates from these areas are

$$\begin{aligned} A_{\text{fin}} &= n_{\text{fin}} \times 4 \times (0.002 \text{ m})(0.04 \text{ m}) = 0.00032 n_{\text{fin}} \text{ m}^2 \\ A_{\text{unfinned}} &= (0.15 \text{ m})(0.20 \text{ m}) - n_{\text{fin}} (0.002 \text{ m})(0.002 \text{ m}) \\ &= 0.03 - 0.000004 n_{\text{fin}} \text{ m}^2 \\ \dot{Q}_{\text{finned}} &= \eta_{\text{fin}} \dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}} (T_b - T_\infty) \\ &= 0.919 (20 \text{ W/m}^2\cdot^\circ\text{C}) (0.00032 n_{\text{fin}} \text{ m}^2) (85 - 25)^\circ\text{C} \\ &= 0.35328 n_{\text{fin}} \text{ W} \\ \dot{Q}_{\text{unfinned}} &= h A_{\text{unfinned}} (T_b - T_\infty) = (20 \text{ W/m}^2\cdot^\circ\text{C}) (0.03 - 0.000004 n_{\text{fin}} \text{ m}^2) (85 - 25)^\circ\text{C} \\ &= 36 - 0.0048 n_{\text{fin}} \text{ W} \end{aligned}$$



Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}} = 0.35328 n_{\text{fin}} + 36 - 0.0048 n_{\text{fin}} \text{ W}$$

The rate of heat transfer if there were no fin attached to the plate would be

$$\begin{aligned} A_{\text{no fin}} &= (0.15 \text{ m})(0.20 \text{ m}) = 0.03 \text{ m}^2 \\ \dot{Q}_{\text{no fin}} &= h A_{\text{no fin}} (T_b - T_\infty) = (20 \text{ W/m}^2\cdot^\circ\text{C}) (0.03 \text{ m}^2) (85 - 25)^\circ\text{C} = 36 \text{ W} \end{aligned}$$

The number of fins can be determined from the overall fin effectiveness equation

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} \longrightarrow 3 = \frac{0.35328 n_{\text{fin}} + 36 - 0.0048 n_{\text{fin}}}{36} \longrightarrow n_{\text{fin}} = \mathbf{207}$$

10-164 EES Prob. 10-163 is reconsidered. The number of fins as a function of the increase in the heat loss by fins relative to no fin case (i.e., overall effectiveness of the fins) is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

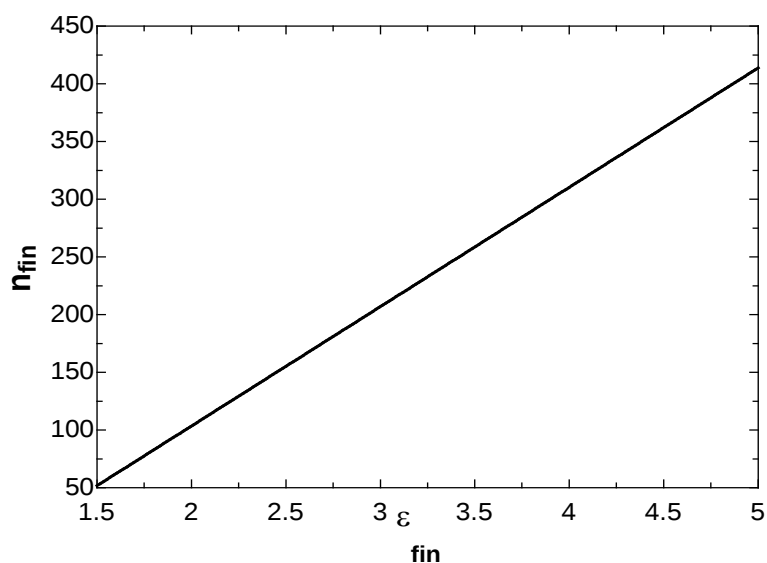
"GIVEN"

$A_{\text{surface}} = 0.15 \times 0.20 \text{ [m}^2\text{]}$
 $T_b = 85 \text{ [C]}; k = 237 \text{ [W/m-C]}$
 $\text{side} = 0.002 \text{ [m]}; L = 0.04 \text{ [m]}$
 $T_{\text{infinity}} = 25 \text{ [C]}$
 $h = 20 \text{ [W/m}^2\text{-C]}$
 $\epsilon_{\text{fin}} = 3$

"ANALYSIS"

$A_c = \text{side}^2$
 $p = 4 \times \text{side}$
 $a = \sqrt{(h \times p) / (k \times A_c)}$
 $\eta_{\text{fin}} = \tanh(a \times L) / (a \times L)$
 $A_{\text{fin}} = n_{\text{fin}} \times 4 \times \text{side} \times L$
 $A_{\text{unfinned}} = A_{\text{surface}} - n_{\text{fin}} \times \text{side}^2$
 $\dot{Q}_{\text{dot finned}} = \eta_{\text{fin}} \times h \times A_{\text{fin}} \times (T_b - T_{\text{infinity}})$
 $\dot{Q}_{\text{dot unfinned}} = h \times A_{\text{unfinned}} \times (T_b - T_{\text{infinity}})$
 $\dot{Q}_{\text{dot total fin}} = \dot{Q}_{\text{dot finned}} + \dot{Q}_{\text{dot unfinned}}$
 $\dot{Q}_{\text{dot nofin}} = h \times A_{\text{surface}} \times (T_b - T_{\text{infinity}})$
 $\epsilon_{\text{fin}} = \dot{Q}_{\text{dot total fin}} / \dot{Q}_{\text{dot nofin}}$

ϵ_{fin}	n_{fin}
1.5	51.72
1.75	77.59
2	103.4
2.25	129.3
2.5	155.2
2.75	181
3	206.9
3.25	232.8
3.5	258.6
3.75	284.5
4	310.3
4.25	336.2
4.5	362.1
4.75	387.9
5	413.8



10-165 A spherical tank containing iced water is buried underground. The rate of heat transfer to the tank is to be determined for the insulated and uninsulated ground surface cases.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the soil is constant. 4 The tank surface is assumed to be at the same temperature as the iced water because of negligible resistance through the steel.

Properties The thermal conductivity of the soil is given to be $k = 0.55 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi D}{1 - 0.25 \frac{D}{z}} = \frac{2\pi(1.4 \text{ m})}{1 - 0.25 \frac{1.4 \text{ m}}{2.4 \text{ m}}} = 10.30 \text{ m}$$

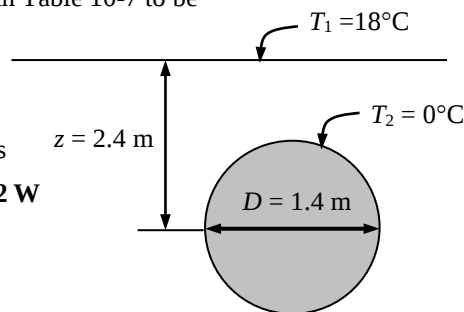
Then the steady rate of heat transfer from the tank becomes

$$\dot{Q} = Sk(T_1 - T_2) = (10.30 \text{ m})(0.55 \text{ W/m}\cdot^\circ\text{C})(18 - 0)^\circ\text{C} = \mathbf{102 \text{ W}}$$

If the ground surface is insulated,

$$S = \frac{2\pi D}{1 + 0.25 \frac{D}{z}} = \frac{2\pi(1.4 \text{ m})}{1 + 0.25 \frac{1.4 \text{ m}}{2.4 \text{ m}}} = 7.68 \text{ m}$$

$$\dot{Q} = Sk(T_1 - T_2) = (7.68 \text{ m})(0.55 \text{ W/m}\cdot^\circ\text{C})(18 - 0)^\circ\text{C} = \mathbf{76 \text{ W}}$$



10-166 A cylindrical tank containing liquefied natural gas (LNG) is placed at the center of a square solid bar. The rate of heat transfer to the tank and the LNG temperature at the end of a one-month period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer is two-dimensional (no change in the axial direction). 3 Thermal conductivity of the bar is constant. 4 The tank surface is at the same temperature as the LNG.

Properties The thermal conductivity of the bar is given to be $k = 0.0002 \text{ W/m}\cdot^\circ\text{C}$. The density and the specific heat of LNG are given to be 425 kg/m^3 and $3.475 \text{ kJ/kg}\cdot^\circ\text{C}$, respectively.

Analysis The shape factor for this configuration is given in Table 10-7 to be

$$S = \frac{2\pi L}{\ln\left(\frac{1.08w}{D}\right)} = \frac{2\pi(1.9 \text{ m})}{\ln\left(1.08 \frac{1.4 \text{ m}}{0.6 \text{ m}}\right)} = 12.92 \text{ m}$$

Then the steady rate of heat transfer to the tank becomes

$$\dot{Q} = Sk(T_1 - T_2) = (12.92 \text{ m})(0.0002 \text{ W/m}\cdot^\circ\text{C})[12 - (-160)]^\circ\text{C} = \mathbf{0.4444 \text{ W}}$$

The mass of LNG is

$$m = \rho V = \rho \pi \frac{D^3}{6} = (425 \text{ kg/m}^3) \pi \frac{(0.6 \text{ m})^3}{6} = 48.07 \text{ kg}$$

The amount heat transfer to the tank for a one-month period is

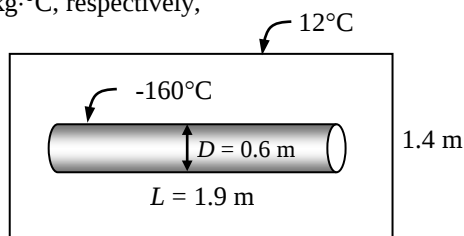
$$Q = \dot{Q} \Delta t = (0.4444 \text{ W})(30 \times 24 \times 3600 \text{ s}) = 1.152 \times 10^6 \text{ J}$$

Then the temperature of LNG at the end of the month becomes

$$Q = mc_p(T_1 - T_2)$$

$$1.152 \times 10^6 \text{ J} = (48.07 \text{ kg})(3475 \text{ J/kg}\cdot^\circ\text{C})[(-160) - T_2]^\circ\text{C}$$

$$T_2 = \mathbf{-153.1^\circ\text{C}}$$



10-167 A typical section of a building wall is considered. The temperature on the interior brick surface is to be determined.

Assumptions 1 Steady operating conditions exist.

Properties The thermal conductivities are given to be $k_{23b} = 50$ W/m·K, $k_{23a} = 0.03$ W/m·K, $k_{12} = 0.5$ W/m·K, $k_{34} = 1.0$ W/m·K.

Analysis We consider 1 m² of wall area. The thermal resistances are

$$R_{12} = \frac{t_{12}}{k_{12}} = \frac{0.01 \text{ m}}{(0.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.02 \text{ m}^2 \cdot ^\circ\text{C/W}$$

$$\begin{aligned} R_{23a} &= t_{23} \frac{L_a}{k_{23a}(L_a + L_b)} \\ &= (0.08 \text{ m}) \frac{0.6 \text{ m}}{(0.03 \text{ W/m} \cdot ^\circ\text{C})(0.6 + 0.005)} = 2.645 \text{ m}^2 \cdot ^\circ\text{C/W} \end{aligned}$$

$$\begin{aligned} R_{23b} &= t_{23} \frac{L_b}{k_{23b}(L_a + L_b)} \\ &= (0.08 \text{ m}) \frac{0.005 \text{ m}}{(50 \text{ W/m} \cdot ^\circ\text{C})(0.6 + 0.005)} = 1.32 \times 10^{-5} \text{ m}^2 \cdot ^\circ\text{C/W} \end{aligned}$$

$$R_{34} = \frac{t_{34}}{k_{34}} = \frac{0.1 \text{ m}}{(1.0 \text{ W/m} \cdot ^\circ\text{C})} = 0.1 \text{ m}^2 \cdot ^\circ\text{C/W}$$

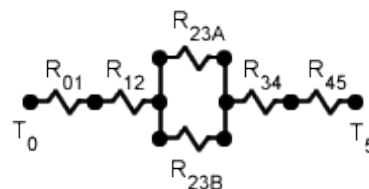
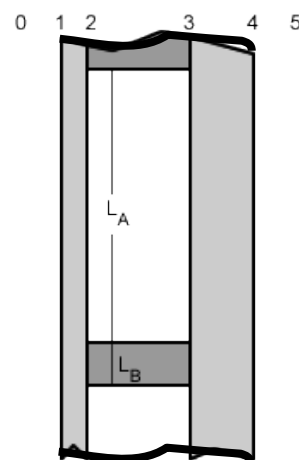
The total thermal resistance and the rate of heat transfer are

$$\begin{aligned} R_{\text{total}} &= R_{12} + \left(\frac{R_{23a} R_{23b}}{R_{23a} + R_{23b}} \right) + R_{34} \\ &= 0.02 + 2.645 \left(\frac{(2.645)(1.32 \times 10^{-5})}{2.645 + 1.32 \times 10^{-5}} \right) + 0.1 = 0.120 \text{ m}^2 \cdot ^\circ\text{C/W} \end{aligned}$$

$$\dot{q} = \frac{T_4 - T_1}{R_{\text{total}}} = \frac{(35 - 20)^\circ\text{C}}{0.120 \text{ m}^2 \cdot ^\circ\text{C/W}} = \mathbf{125 \text{ W/m}^2}$$

The temperature on the interior brick surface is

$$\dot{q} = \frac{T_4 - T_3}{R_{34}} \longrightarrow 125 \text{ W/m}^2 = \frac{(35 - T_3)^\circ\text{C}}{0.1 \text{ m}^2 \cdot ^\circ\text{C/W}} \longrightarrow T_3 = \mathbf{22.5^\circ\text{C}}$$



10-168 Ten rectangular aluminum fins are placed on the outside surface of an electronic device. The rate of heat loss from the electronic device to the surrounding air and the fin effectiveness are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature along the fins varies in one direction only (normal to the plate). 3 The heat transfer coefficient is constant and uniform over the entire fin surface. 4 The thermal properties of the fins are constant. 5 The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum fin is given to be $k = 203 \text{ W/m}\cdot\text{K}$.

Analysis The fin efficiency is to be determined using Fig. 10-42 in the text.

$$\xi = L_c^{3/2} \sqrt{h/(kA_p)} = (L + t/2) \sqrt{h/(kt)} = (0.020 + 0.004/2) \sqrt{\frac{100}{(203)(0.004)}} = 0.244 \longrightarrow \eta_{\text{fin}} = 0.93$$

The rate of heat loss can be determined as follows

$$A_{\text{fin}} = 2 \times 10(0.020 \times 0.100 + 0.004 \times 0.020) = 0.0416 \text{ m}^2$$

$$A_{\text{base}} = 10(0.100 \times 0.004) = 0.004 \text{ m}^2$$

$$\eta_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{fin, max}}} = \frac{\dot{Q}_{\text{fin}}}{hA_{\text{fin}}(T_b - T_{\infty})} \longrightarrow 0.93 = \frac{\dot{Q}_{\text{fin}}}{(100)(0.0416)(60 - 20)} \longrightarrow \dot{Q}_{\text{fin}} = 155 \text{ W}$$

$$\dot{Q}_{\text{base}} = hA_{\text{base}}(T_b - T_{\infty}) = (100)(0.004)(60 - 20) = 16 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{fin}} + \dot{Q}_{\text{base}} = 155 + 16 = \mathbf{171 \text{ W}}$$

The fin effectiveness is

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} = \frac{\dot{Q}_{\text{fin}}}{hA_{\text{base, no fin}}(T_b - T_{\infty})} = \frac{171}{(100)(0.080 \times 0.100)(60 - 20)} = \mathbf{5.3}$$

10-169 One wall of a refrigerated warehouse is made of three layers. The rates of heat transfer across the warehouse without and with the metal bolts, and the percent change in the rate of heat transfer across the wall due to metal bolts are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer coefficients are constant.

Properties The thermal conductivities are given to be $k_{Al} = 200 \text{ W/m}\cdot\text{K}$, $k_{\text{fiberglass}} = 0.038 \text{ W/m}\cdot\text{K}$, $k_{\text{gypsum}} = 0.48 \text{ W/m}\cdot\text{K}$, and $k_{\text{bolts}} = 43 \text{ W/m}\cdot\text{K}$.

Analysis (a) The rate of heat transfer through the warehouse is

$$U_1 = \frac{1}{\frac{1}{h_i} + \frac{L_{Al}}{k_{Al}} + \frac{L_{fg}}{k_{fg}} + \frac{L_{gy}}{k_{gy}} + \frac{1}{h_o}}$$

$$= \frac{1}{\frac{1}{40 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{0.01 \text{ m}}{200 \text{ W/m}\cdot^\circ\text{C}} + \frac{0.08 \text{ m}}{0.038 \text{ W/m}\cdot^\circ\text{C}} + \frac{0.03 \text{ m}}{0.48 \text{ W/m}\cdot^\circ\text{C}} + \frac{1}{40 \text{ W/m}^2 \cdot ^\circ\text{C}}} = 0.451 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q}_1 = U_1 A (T_o - T_i) = (0.451 \text{ W/m}^2 \cdot ^\circ\text{C})(5 \times 10 \text{ m}^2)[20 - (-10)^\circ\text{C}] = \mathbf{676 \text{ W}}$$

(b) The rate of heat transfer with the consideration of metal bolts is

$$\dot{Q}_1 = U_1 A_1 (T_o - T_i) = (0.451)[10 \times 5 - 400 \times 0.25\pi(0.02)^2][20 - (-10)] = 674.8 \text{ W}$$

$$U_2 = \frac{1}{\frac{1}{h_i} + \frac{L_{bolts}}{k_{bolts}} + \frac{1}{h_o}} = \frac{1}{\frac{1}{40 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{0.12 \text{ m}}{43 \text{ W/m}\cdot^\circ\text{C}} + \frac{1}{40 \text{ W/m}^2 \cdot ^\circ\text{C}}} = 18.94 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q}_2 = U_2 A_2 (T_o - T_i) = (18.94 \text{ W/m}^2 \cdot ^\circ\text{C})[400 \times 0.25\pi(0.02)^2 \text{ m}^2][20 - (-10)^\circ\text{C}] = 71.4 \text{ W}$$

$$\dot{Q} = \dot{Q}_1 + \dot{Q}_2 = 674.8 + 71.4 = \mathbf{746 \text{ W}}$$

(c) The percent change in the rate of heat transfer across the wall due to metal bolts is

$$\% \text{ change} = \frac{746 - 676}{676} = 0.103 = \mathbf{10.3\%}$$

10-170 ... 10-173 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 11
TRANSIENT HEAT CONDUCTION

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Lumped System Analysis

11-1C In heat transfer analysis, some bodies are observed to behave like a "lump" whose entire body temperature remains essentially uniform at all times during a heat transfer process. The temperature of such bodies can be taken to be a function of time only. Heat transfer analysis which utilizes this idealization is known as the lumped system analysis. It is applicable when the Biot number (the ratio of conduction resistance within the body to convection resistance at the surface of the body) is less than or equal to 0.1.

11-2C The lumped system analysis is more likely to be applicable for the body cooled naturally since the Biot number is proportional to the convection heat transfer coefficient, which is proportional to the air velocity. Therefore, the Biot number is more likely to be less than 0.1 for the case of natural convection.

11-3C The lumped system analysis is more likely to be applicable for the body allowed to cool in the air since the Biot number is proportional to the convection heat transfer coefficient, which is larger in water than it is in air because of the larger thermal conductivity of water. Therefore, the Biot number is more likely to be less than 0.1 for the case of the solid cooled in the air.

11-4C The temperature drop of the potato during the second minute will be less than 4°C since the temperature of a body approaches the temperature of the surrounding medium asymptotically, and thus it changes rapidly at the beginning, but slowly later on.

11-5C The temperature rise of the potato during the second minute will be less than 5°C since the temperature of a body approaches the temperature of the surrounding medium asymptotically, and thus it changes rapidly at the beginning, but slowly later on.

11-6C Biot number represents the ratio of conduction resistance within the body to convection resistance at the surface of the body. The Biot number is more likely to be larger for poorly conducting solids since such bodies have larger resistances against heat conduction.

11-7C The heat transfer is proportional to the surface area. Two half pieces of the roast have a much larger surface area than the single piece and thus a higher rate of heat transfer. As a result, the two half pieces will cook much faster than the single large piece.

11-8C The cylinder will cool faster than the sphere since heat transfer rate is proportional to the surface area, and the sphere has the smallest area for a given volume.

11-9C The lumped system analysis is more likely to be applicable in air than in water since the convection heat transfer coefficient and thus the Biot number is much smaller in air.

11-10C The lumped system analysis is more likely to be applicable for a golden apple than for an actual apple since the thermal conductivity is much larger and thus the Biot number is much smaller for gold.

11-11C The lumped system analysis is more likely to be applicable to slender bodies than the well-rounded bodies since the characteristic length (ratio of volume to surface area) and thus the Biot number is much smaller for slender bodies.

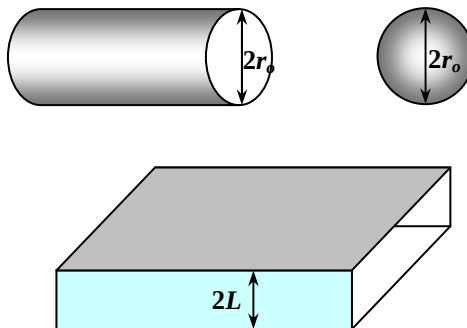
11-12 Relations are to be obtained for the characteristic lengths of a large plane wall of thickness $2L$, a very long cylinder of radius r_o and a sphere of radius r_o .

Analysis Relations for the characteristic lengths of a large plane wall of thickness $2L$, a very long cylinder of radius r_o and a sphere of radius r_o are

$$L_{c,wall} = \frac{V}{A_{surface}} = \frac{2LA}{2A} = L$$

$$L_{c,cylinder} = \frac{V}{A_{surface}} = \frac{\pi r_o^2 h}{2\pi r_o h} = \frac{r_o}{2}$$

$$L_{c,sphere} = \frac{V}{A_{surface}} = \frac{4\pi r_o^3 / 3}{4\pi r_o^2} = \frac{r_o}{3}$$



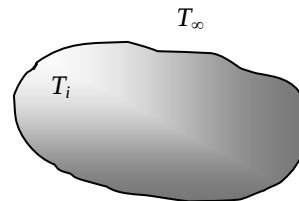
11-13 A relation for the time period for a lumped system to reach the average temperature $(T_i + T_\infty) / 2$ is to be obtained.

Analysis The relation for time period for a lumped system to reach the average temperature $(T_i + T_\infty) / 2$ can be determined as

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{\frac{T_i + T_\infty}{2} - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{T_i - T_\infty}{2(T_i - T_\infty)} = e^{-bt} \longrightarrow \frac{1}{2} = e^{-bt}$$

$$-bt = -\ln 2 \longrightarrow t = \frac{\ln 2}{b} = \frac{0.693}{b}$$



11-14 The temperature of a gas stream is to be measured by a thermocouple. The time it takes to register 99 percent of the initial ΔT is to be determined.

Assumptions **1** The junction is spherical in shape with a diameter of $D = 0.0012$ m. **2** The thermal properties of the junction are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** Radiation effects are negligible. **5** The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

Properties The properties of the junction are given to be $k = 35$ W/m. $^{\circ}$ C, $\rho = 8500$ kg/m 3 , and $c_p = 320$ J/kg. $^{\circ}$ C.

Analysis The characteristic length of the junction and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{0.0012 \text{ m}}{6} = 0.0002 \text{ m}$$

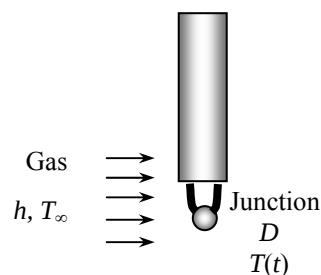
$$Bi = \frac{hL_c}{k} = \frac{(90 \text{ W/m}^2 \cdot ^{\circ}\text{C})(0.0002 \text{ m})}{(35 \text{ W/m} \cdot ^{\circ}\text{C})} = 0.00051 < 0.1$$

Since $Bi < 0.1$, the lumped system analysis is applicable. Then the time period for the thermocouple to read 99% of the initial temperature difference is determined from

$$\frac{T(t) - T_{\infty}}{T_i - T_{\infty}} = 0.01$$

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{90 \text{ W/m}^2 \cdot ^{\circ}\text{C}}{(8500 \text{ kg/m}^3)(320 \text{ J/kg} \cdot ^{\circ}\text{C})(0.0002 \text{ m})} = 0.1654 \text{ s}^{-1}$$

$$\frac{T(t) - T_{\infty}}{T_i - T_{\infty}} = e^{-bt} \longrightarrow 0.01 = e^{-(0.1654 \text{ s}^{-1})t} \longrightarrow t = \mathbf{27.8 \text{ s}}$$



11-15E A number of brass balls are to be quenched in a water bath at a specified rate. The temperature of the balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature constant are to be determined.

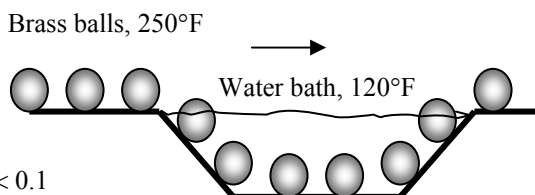
Assumptions 1 The balls are spherical in shape with a radius of $r_o = 1$ in. 2 The thermal properties of the balls are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

Properties The thermal conductivity, density, and specific heat of the brass balls are given to be $k = 64.1$ Btu/h.ft.°F, $\rho = 532$ lbm/ft³, and $c_p = 0.092$ Btu/lbm.°F.

Analysis (a) The characteristic length and the Biot number for the brass balls are

$$L_c = \frac{V}{A_s} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{2 / 12 \text{ ft}}{6} = 0.02778 \text{ ft}$$

$$Bi = \frac{hL_c}{k} = \frac{(42 \text{ Btu/h.ft}^2 \cdot \text{°F})(0.02778 \text{ ft})}{(64.1 \text{ Btu/h.ft.°F})} = 0.01820 < 0.1$$



The lumped system analysis is applicable since $Bi < 0.1$. Then the temperature of the balls after quenching becomes

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{42 \text{ Btu/h.ft}^2 \cdot \text{°F}}{(532 \text{ lbm/ft}^3)(0.092 \text{ Btu/lbm.°F})(0.02778 \text{ ft})} = 30.9 \text{ h}^{-1} = 0.00858 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{T(t) - 120}{250 - 120} = e^{-(0.00858 \text{ s}^{-1})(120 \text{ s})} \longrightarrow T(t) = \mathbf{166^\circ\text{F}}$$

(b) The total amount of heat transfer from a ball during a 2-minute period is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (532 \text{ lbm/ft}^3) \frac{\pi (2 / 12 \text{ ft})^3}{6} = 1.290 \text{ lbm}$$

$$Q = mc_p [T_i - T(t)] = (1.29 \text{ lbm})(0.092 \text{ Btu/lbm.°F})(250 - 166)^\circ\text{F} = 9.97 \text{ Btu}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{ball} = (120 \text{ balls/min}) \times (9.97 \text{ Btu}) = \mathbf{1196 \text{ Btu/min}}$$

Therefore, heat must be removed from the water at a rate of 1196 Btu/min in order to keep its temperature constant at 120°F.

11-16E A number of aluminum balls are to be quenched in a water bath at a specified rate. The temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature constant are to be determined.

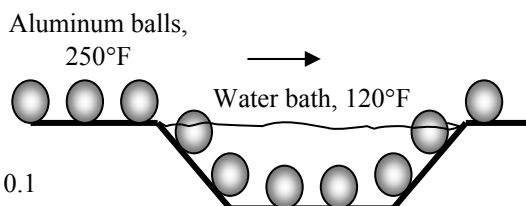
Assumptions 1 The balls are spherical in shape with a radius of $r_o = 1$ in. 2 The thermal properties of the balls are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

Properties The thermal conductivity, density, and specific heat of the aluminum balls are $k = 137$ Btu/h.ft.°F, $\rho = 168$ lbm/ft³, and $c_p = 0.216$ Btu/lbm.°F (Table A-24E).

Analysis (a) The characteristic length and the Biot number for the aluminum balls are

$$L_c = \frac{V}{A} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{2 / 12 \text{ ft}}{6} = 0.02778 \text{ ft}$$

$$Bi = \frac{hL_c}{k} = \frac{(42 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(0.02778 \text{ ft})}{(137 \text{ Btu/h.ft.}^\circ\text{F})} = 0.00852 < 0.1$$



The lumped system analysis is applicable since $Bi < 0.1$. Then the temperature of the balls after quenching becomes

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{42 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}}{(168 \text{ lbm/ft}^3)(0.216 \text{ Btu/lbm.}^\circ\text{F})(0.02778 \text{ ft})} = 41.66 \text{ h}^{-1} = 0.01157 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{T(t) - 120}{250 - 120} = e^{-(0.01157 \text{ s}^{-1})(120 \text{ s})} \longrightarrow T(t) = 152^\circ\text{F}$$

(b) The total amount of heat transfer from a ball during a 2-minute period is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (168 \text{ lbm/ft}^3) \frac{\pi (2 / 12 \text{ ft})^3}{6} = 0.4072 \text{ lbm}$$

$$Q = mc_p [T_i - T(t)] = (0.4072 \text{ lbm})(0.216 \text{ Btu/lbm.}^\circ\text{F})(250 - 152)^\circ\text{F} = 8.62 \text{ Btu}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{ball} = (120 \text{ balls/min}) \times (8.62 \text{ Btu}) = 1034 \text{ Btu/min}$$

Therefore, heat must be removed from the water at a rate of 1034 Btu/min in order to keep its temperature constant at 120°F.

11-17 Milk in a thin-walled glass container is to be warmed up by placing it into a large pan filled with hot water. The warming time of the milk is to be determined.

Assumptions 1 The glass container is cylindrical in shape with a radius of $r_0 = 3$ cm. 2 The thermal properties of the milk are taken to be the same as those of water. 3 Thermal properties of the milk are constant at room temperature. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Biot number in this case is large (much larger than 0.1). However, the lumped system analysis is still applicable since the milk is stirred constantly, so that its temperature remains uniform at all times.

Properties The thermal conductivity, density, and specific heat of the milk at 20°C are $k = 0.598$ W/m $\cdot^\circ\text{C}$, $\rho = 998$ kg/m 3 , and $c_p = 4.182$ kJ/kg $\cdot^\circ\text{C}$ (Table A-15).

Analysis The characteristic length and Biot number for the glass of milk are

$$L_c = \frac{V}{A_s} = \frac{\pi r_o^2 L}{2\pi r_o L + 2\pi r_o^2} = \frac{\pi(0.03 \text{ m})^2 (0.07 \text{ m})}{2\pi(0.03 \text{ m})(0.07 \text{ m}) + 2\pi(0.03 \text{ m})^2} = 0.01050 \text{ m}$$

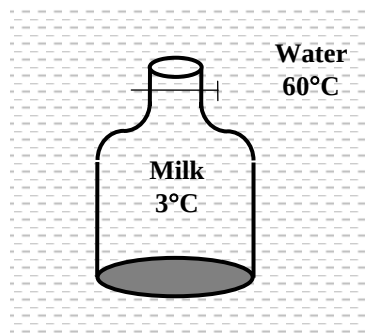
$$Bi = \frac{hL_c}{k} = \frac{(120 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0105 \text{ m})}{(0.598 \text{ W/m} \cdot ^\circ\text{C})} = 2.107 > 0.1$$

For the reason explained above we can use the lumped system analysis to determine how long it will take for the milk to warm up to 38°C :

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{120 \text{ W/m}^2 \cdot ^\circ\text{C}}{(998 \text{ kg/m}^3)(4182 \text{ J/kg} \cdot ^\circ\text{C})(0.0105 \text{ m})} = 0.002738 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{38 - 60}{3 - 60} = e^{-(0.002738 \text{ s}^{-1})t} \longrightarrow t = 348 \text{ s} = 5.8 \text{ min}$$

Therefore, it will take about 6 minutes to warm the milk from 3 to 38°C .



11-18 A thin-walled glass containing milk is placed into a large pan filled with hot water to warm up the milk. The warming time of the milk is to be determined.

Assumptions 1 The glass container is cylindrical in shape with a radius of $r_0 = 3$ cm. 2 The thermal properties of the milk are taken to be the same as those of water. 3 Thermal properties of the milk are constant at room temperature. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Biot number in this case is large (much larger than 0.1). However, the lumped system analysis is still applicable since the milk is stirred constantly, so that its temperature remains uniform at all times.

Properties The thermal conductivity, density, and specific heat of the milk at 20°C are $k = 0.598$ W/m $\cdot^\circ\text{C}$, $\rho = 998$ kg/m 3 , and $c_p = 4.182$ kJ/kg $\cdot^\circ\text{C}$ (Table A-15).

Analysis The characteristic length and Biot number for the glass of milk are

$$L_c = \frac{V}{A_s} = \frac{\pi r_o^2 L}{2\pi r_o L + 2\pi r_o^2} = \frac{\pi(0.03\text{ m})^2(0.07\text{ m})}{2\pi(0.03\text{ m})(0.07\text{ m}) + 2\pi(0.03\text{ m})^2} = 0.01050\text{ m}$$

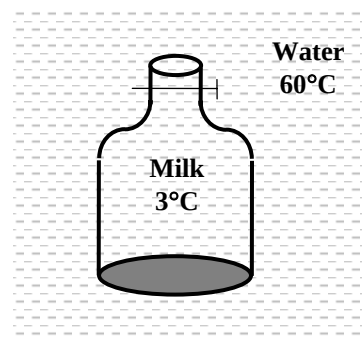
$$Bi = \frac{hL_c}{k} = \frac{(240\text{ W/m}^2\cdot^\circ\text{C})(0.01050\text{ m})}{(0.598\text{ W/m}\cdot^\circ\text{C})} = 4.21 > 0.1$$

For the reason explained above we can use the lumped system analysis to determine how long it will take for the milk to warm up to 38°C :

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{240\text{ W/m}^2\cdot^\circ\text{C}}{(998\text{ kg/m}^3)(4182\text{ J/kg}\cdot^\circ\text{C})(0.01050\text{ m})} = 0.005477\text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{38 - 60}{3 - 60} = e^{-(0.005477\text{ s}^{-1})t} \longrightarrow t = 174\text{ s} = 2.9\text{ min}$$

Therefore, it will take about 3 minutes to warm the milk from 3 to 38°C .



11-19 A long copper rod is cooled to a specified temperature. The cooling time is to be determined.

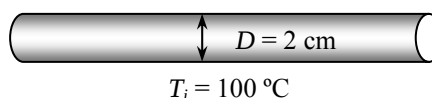
Assumptions 1 The thermal properties of the geometry are constant. 2 The heat transfer coefficient is constant and uniform over the entire surface.

Properties The properties of copper are $k = 401$ W/m $\cdot^\circ\text{C}$, $\rho = 8933$ kg/m 3 , and $c_p = 0.385$ kJ/kg $\cdot^\circ\text{C}$ (Table A-24).

Analysis For cylinder, the characteristic length and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{(\pi D^2 / 4)L}{\pi DL} = \frac{D}{4} = \frac{0.02\text{ m}}{4} = 0.005\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(200\text{ W/m}^2\cdot^\circ\text{C})(0.005\text{ m})}{(401\text{ W/m}\cdot^\circ\text{C})} = 0.0025 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then the cooling time is determined from

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{200\text{ W/m}^2\cdot^\circ\text{C}}{(8933\text{ kg/m}^3)(385\text{ J/kg}\cdot^\circ\text{C})(0.005\text{ m})} = 0.01163\text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{25 - 20}{100 - 20} = e^{-(0.01163\text{ s}^{-1})t} \longrightarrow t = 238\text{ s} = 4.0\text{ min}$$

11-20 The heating times of a sphere, a cube, and a rectangular prism with similar dimensions are to be determined.

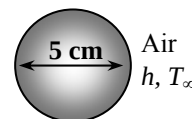
Assumptions 1 The thermal properties of the geometries are constant. **2** The heat transfer coefficient is constant and uniform over the entire surface.

Properties The properties of silver are given to be $k = 429 \text{ W/m}\cdot^\circ\text{C}$, $\rho = 10,500 \text{ kg/m}^3$, and $c_p = 0.235 \text{ kJ/kg}\cdot^\circ\text{C}$.

Analysis For sphere, the characteristic length and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{0.05 \text{ m}}{6} = 0.008333 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12 \text{ W/m}^2\cdot^\circ\text{C})(0.008333 \text{ m})}{(429 \text{ W/m}\cdot^\circ\text{C})} = 0.00023 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then the time period for the sphere temperature to reach to 25°C is determined from

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{12 \text{ W/m}^2\cdot^\circ\text{C}}{(10,500 \text{ kg/m}^3)(235 \text{ J/kg}\cdot^\circ\text{C})(0.008333 \text{ m})} = 0.0005836 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{25 - 33}{0 - 33} = e^{-(0.0005836 \text{ s}^{-1})t} \longrightarrow t = 2428 \text{ s} = \mathbf{40.5 \text{ min}}$$

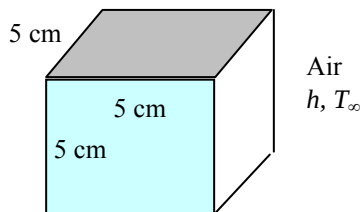
Cube:

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{L^3}{6L^2} = \frac{L}{6} = \frac{0.05 \text{ m}}{6} = 0.008333 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12 \text{ W/m}^2\cdot^\circ\text{C})(0.008333 \text{ m})}{(429 \text{ W/m}\cdot^\circ\text{C})} = 0.00023 < 0.1$$

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{12 \text{ W/m}^2\cdot^\circ\text{C}}{(10,500 \text{ kg/m}^3)(235 \text{ J/kg}\cdot^\circ\text{C})(0.008333 \text{ m})} = 0.0005836 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{25 - 33}{0 - 33} = e^{-(0.0005836 \text{ s}^{-1})t} \longrightarrow t = 2428 \text{ s} = \mathbf{40.5 \text{ min}}$$



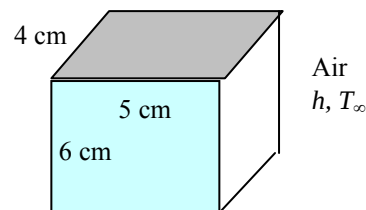
Rectangular prism:

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{(0.04 \text{ m})(0.05 \text{ m})(0.06 \text{ m})}{2(0.04 \text{ m})(0.05 \text{ m}) + 2(0.04 \text{ m})(0.06 \text{ m}) + 2(0.05 \text{ m})(0.06 \text{ m})} = 0.008108 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12 \text{ W/m}^2\cdot^\circ\text{C})(0.008108 \text{ m})}{(429 \text{ W/m}\cdot^\circ\text{C})} = 0.00023 < 0.1$$

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{12 \text{ W/m}^2\cdot^\circ\text{C}}{(10,500 \text{ kg/m}^3)(235 \text{ J/kg}\cdot^\circ\text{C})(0.008108 \text{ m})} = 0.0005998 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{25 - 33}{0 - 33} = e^{-(0.0005998 \text{ s}^{-1})t} \longrightarrow t = 2363 \text{ s} = \mathbf{39.4 \text{ min}}$$



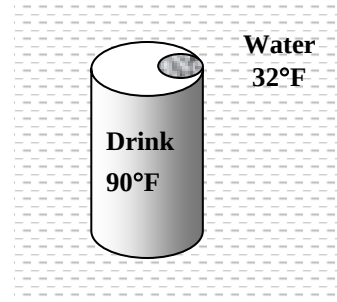
The heating times are same for the sphere and cube while it is smaller in rectangular prism.

11-21E A person shakes a can of drink in a iced water to cool it. The cooling time of the drink is to be determined.

Assumptions 1 The can containing the drink is cylindrical in shape with a radius of $r_o = 1.25$ in. 2 The thermal properties of the drink are taken to be the same as those of water. 3 Thermal properties of the drink are constant at room temperature. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Biot number in this case is large (much larger than 0.1). However, the lumped system analysis is still applicable since the drink is stirred constantly, so that its temperature remains uniform at all times.

Properties The density and specific heat of water at room temperature are $\rho = 62.22 \text{ lbm/ft}^3$, and $c_p = 0.999 \text{ Btu/lbm} \cdot ^\circ\text{F}$ (Table A-15E).

Analysis Application of lumped system analysis in this case gives



$$L_c = \frac{V}{A_s} = \frac{\pi r_o^2 L}{2\pi r_o L + 2\pi r_o^2} = \frac{\pi (1.25/12 \text{ ft})^2 (5/12 \text{ ft})}{2\pi (1.25/12 \text{ ft})(5/12 \text{ ft}) + 2\pi (1.25/12 \text{ ft})^2} = 0.04167 \text{ ft}$$

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{30 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}}{(62.22 \text{ lbm/ft}^3)(0.999 \text{ Btu/lbm} \cdot ^\circ\text{F})(0.04167 \text{ ft})} = 11.583 \text{ h}^{-1} = 0.00322 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{40 - 32}{90 - 32} = e^{-(0.00322 \text{ s}^{-1})t} \longrightarrow t = \mathbf{615 \text{ s}}$$

Therefore, it will take 10 minutes and 15 seconds to cool the canned drink to 40°F.

11-22 An iron whose base plate is made of an aluminum alloy is turned on. The time for the plate temperature to reach 140°C and whether it is realistic to assume the plate temperature to be uniform at all times are to be determined.

Assumptions 1 85 percent of the heat generated in the resistance wires is transferred to the plate. 2 The thermal properties of the plate are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface.

Properties The density, specific heat, and thermal diffusivity of the aluminum alloy plate are given to be $\rho = 2770 \text{ kg/m}^3$, $c_p = 875 \text{ J/kg} \cdot ^\circ\text{C}$, and $\alpha = 7.3 \times 10^{-5} \text{ m}^2/\text{s}$. The thermal conductivity of the plate can be determined from $k = \alpha \rho c_p = 177 \text{ W/m} \cdot ^\circ\text{C}$ (or it can be read from Table A-24).

Analysis The mass of the iron's base plate is

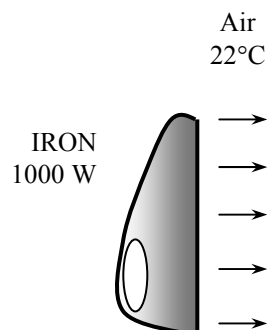
$$m = \rho V = \rho LA = (2770 \text{ kg/m}^3)(0.005 \text{ m})(0.03 \text{ m}^2) = 0.4155 \text{ kg}$$

Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\text{in}} = 0.85 \times 1000 \text{ W} = 850 \text{ W}$$

The temperature of the plate, and thus the rate of heat transfer from the plate, changes during the process. Using the average plate temperature, the average rate of heat loss from the plate is determined from

$$\dot{Q}_{\text{loss}} = hA(T_{\text{plate, ave}} - T_\infty) = (12 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03 \text{ m}^2) \left(\frac{140 + 22}{2} - 22 \right) ^\circ\text{C} = 21.2 \text{ W}$$



Energy balance on the plate can be expressed as

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{plate}} \rightarrow \dot{Q}_{\text{in}} \Delta t - \dot{Q}_{\text{out}} \Delta t = \Delta E_{\text{plate}} = mc_p \Delta T_{\text{plate}}$$

Solving for Δt and substituting,

$$\Delta t = \frac{mc_p \Delta T_{\text{plate}}}{\dot{Q}_{\text{in}} - \dot{Q}_{\text{out}}} = \frac{(0.4155 \text{ kg})(875 \text{ J/kg} \cdot ^\circ\text{C})(140 - 22)^\circ\text{C}}{(850 - 21.2) \text{ J/s}} = 51.8 \text{ s}$$

which is the time required for the plate temperature to reach 140°C. To determine whether it is realistic to assume the plate temperature to be uniform at all times, we need to calculate the Biot number,

$$L_c = \frac{V}{A_s} = \frac{LA}{A} = L = 0.005 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12 \text{ W/m}^2 \cdot ^\circ\text{C})(0.005 \text{ m})}{(177.0 \text{ W/m} \cdot ^\circ\text{C})} = 0.00034 < 0.1$$

It is realistic to assume uniform temperature for the plate since $Bi < 0.1$.

Discussion This problem can also be solved by obtaining the differential equation from an energy balance on the plate for a differential time interval, and solving the differential equation. It gives

$$T(t) = T_\infty + \frac{\dot{Q}_{\text{in}}}{hA} \left(1 - \exp\left(-\frac{hA}{mc_p} t\right) \right)$$

Substituting the known quantities and solving for t again gives 51.8 s.

11-23 EES Prob. 11-22 is reconsidered. The effects of the heat transfer coefficient and the final plate temperature on the time it will take for the plate to reach this temperature are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$E_{\text{dot}}=1000$ [W]
 $L=0.005$ [m]
 $A=0.03$ [m²]
 $T_{\text{infinity}}=22$ [C]
 $T_i=T_{\text{infinity}}$
 $h=12$ [W/m²-C]
 $f_{\text{heat}}=0.85$
 $T_f=140$ [C]

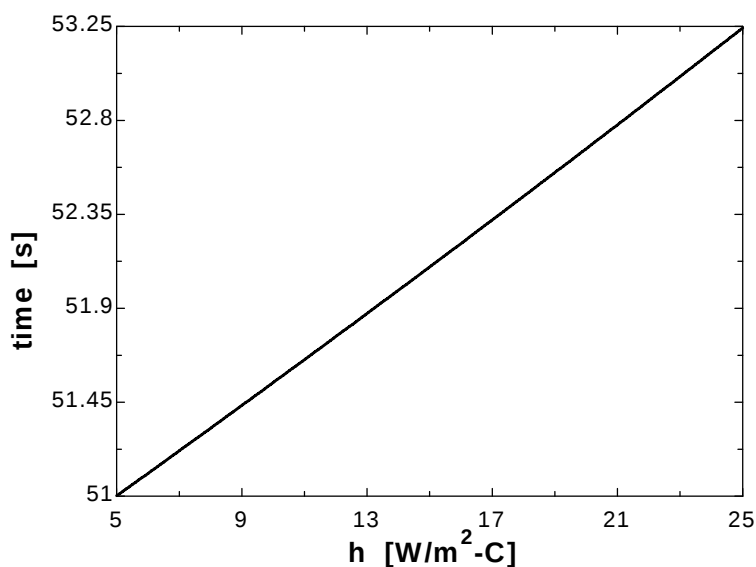
"PROPERTIES"

$\rho=2770$ [kg/m³]
 $c_p=875$ [J/kg-C]
 $\alpha=7.3E-5$ [m²/s]

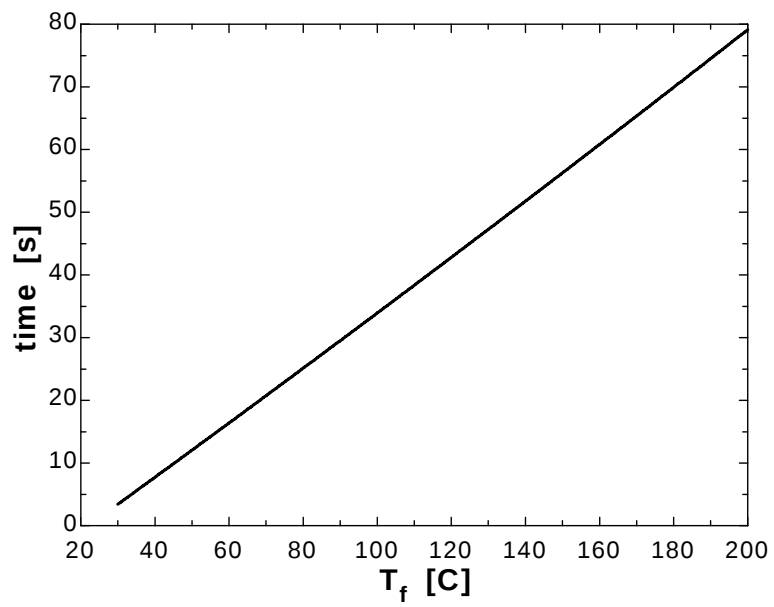
"ANALYSIS"

$V=L \cdot A$
 $m=\rho \cdot V$
 $Q_{\text{dot_in}}=f_{\text{heat}} \cdot E_{\text{dot}}$
 $Q_{\text{dot_out}}=h \cdot A \cdot (T_{\text{ave}}-T_{\text{infinity}})$
 $T_{\text{ave}}=1/2 \cdot (T_i+T_f)$
 $(Q_{\text{dot_in}}-Q_{\text{dot_out}}) \cdot \text{time}=m \cdot c_p \cdot (T_f-T_i)$ "energy balance on the plate"

h [W/m ² .C]	time [s]
5	51
7	51.22
9	51.43
11	51.65
13	51.88
15	52.1
17	52.32
19	52.55
21	52.78
23	53.01
25	53.24



T_f [C]	time [s]
30	3.428
40	7.728
50	12.05
60	16.39
70	20.74
80	25.12
90	29.51
100	33.92
110	38.35
120	42.8
130	47.28
140	51.76
150	56.27
160	60.8
170	65.35
180	69.92
190	74.51
200	79.12



11-24 Ball bearings leaving the oven at a uniform temperature of 900°C are exposed to air for a while before they are dropped into the water for quenching. The time they can stand in the air before their temperature falls below 850°C is to be determined.

Assumptions **1** The bearings are spherical in shape with a radius of $r_o = 0.6$ cm. **2** The thermal properties of the bearings are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Biot number is $\text{Bi} < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

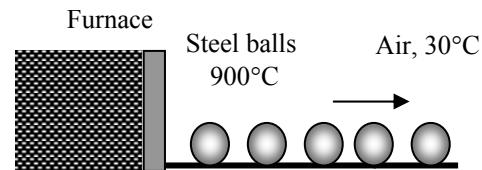
Properties The thermal conductivity, density, and specific heat of the bearings are given to be $k = 15.1$ W/m $\cdot^{\circ}\text{C}$, $\rho = 8085$ kg/m 3 , and $c_p = 0.480$ kJ/kg $\cdot^{\circ}\text{F}$.

Analysis The characteristic length of the steel ball bearings and Biot number are

$$L_c = \frac{V}{A_s} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{0.012 \text{ m}}{6} = 0.002 \text{ m}$$

$$\text{Bi} = \frac{hL_c}{k} = \frac{(125 \text{ W/m}^2 \cdot ^{\circ}\text{C})(0.002 \text{ m})}{(15.1 \text{ W/m} \cdot ^{\circ}\text{C})} = 0.0166 < 0.1$$

Therefore, the lumped system analysis is applicable. Then the allowable time is determined to be



$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{125 \text{ W/m}^2 \cdot ^{\circ}\text{C}}{(8085 \text{ kg/m}^3)(0.480 \text{ kJ/kg} \cdot ^{\circ}\text{F})(0.002 \text{ m})} = 0.01610 \text{ s}^{-1}$$

$$\frac{T(t) - T_{\infty}}{T_i - T_{\infty}} = e^{-bt} \longrightarrow \frac{850 - 30}{900 - 30} = e^{-(0.0161 \text{ s}^{-1})t} \longrightarrow t = \mathbf{3.68 \text{ s}}$$

The result indicates that the ball bearing can stay in the air about 4 s before being dropped into the water.

11-25 A number of carbon steel balls are to be annealed by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The time of annealing and the total rate of heat transfer from the balls to the ambient air are to be determined.

Assumptions 1 The balls are spherical in shape with a radius of $r_o = 4$ mm. 2 The thermal properties of the balls are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

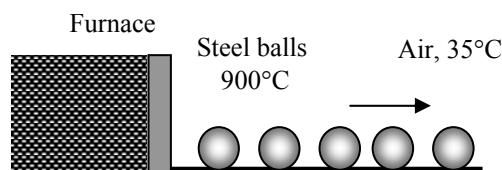
Properties The thermal conductivity, density, and specific heat of the balls are given to be $k = 54$ W/m·°C, $\rho = 7833$ kg/m³, and $c_p = 0.465$ kJ/kg·°C.

Analysis The characteristic length of the balls and the Biot number are

$$L_c = \frac{V}{A_s} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{0.008 \text{ m}}{6} = 0.0013 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(75 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0013 \text{ m})}{(54 \text{ W/m} \cdot ^\circ\text{C})} = 0.0018 < 0.1$$

Therefore, the lumped system analysis is applicable. Then the time for the annealing process is determined to be



$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{75 \text{ W/m}^2 \cdot ^\circ\text{C}}{(7833 \text{ kg/m}^3)(465 \text{ J/kg} \cdot ^\circ\text{C})(0.0013 \text{ m})} = 0.01584 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{100 - 35}{900 - 35} = e^{-(0.01584 \text{ s}^{-1})t} \longrightarrow t = 163 \text{ s} = 2.7 \text{ min}$$

The amount of heat transfer from a single ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (7833 \text{ kg/m}^3) \frac{\pi (0.008 \text{ m})^3}{6} = 0.0021 \text{ kg}$$

$$Q = mc_p [T_f - T_i] = (0.0021 \text{ kg})(465 \text{ J/kg} \cdot ^\circ\text{C})(900 - 100)^\circ\text{C} = 781 \text{ J} = 0.781 \text{ kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

$$\dot{Q} = \dot{n}_{\text{ball}} Q = (2500 \text{ balls/h}) \times (0.781 \text{ kJ/ball}) = 1,953 \text{ kJ/h} = 543 \text{ W}$$

11-26 EES Prob. 11-25 is reconsidered. The effect of the initial temperature of the balls on the annealing time and the total rate of heat transfer is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$D=0.008$ [m]; $T_i=900$ [C]
 $T_f=100$ [C]; $T_{\text{infinity}}=35$ [C]
 $h=75$ [W/m²-C]; $n_{\text{dot_ball}}=2500$ [1/h]

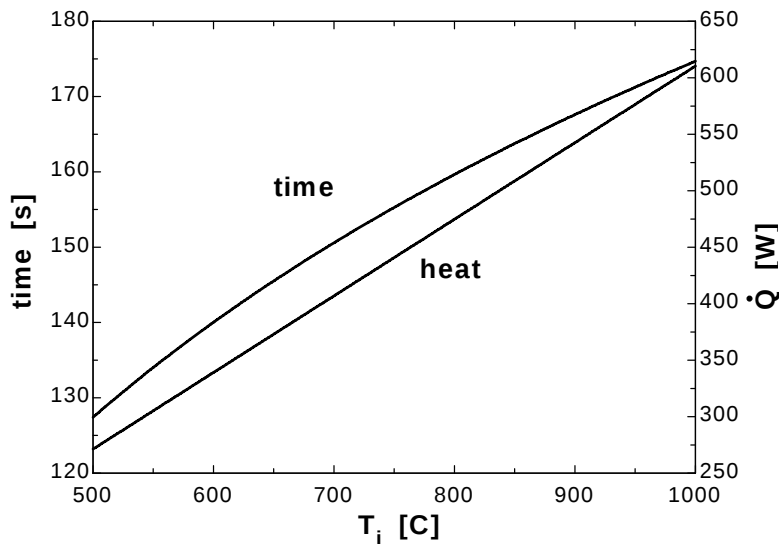
"PROPERTIES"

$\rho=7833$ [kg/m³]; $k=54$ [W/m-C]
 $c_p=465$ [J/kg-C]; $\alpha=1.474\text{E-}6$ [m²/s]

"ANALYSIS"

$A=\pi \cdot D^2$
 $V=\pi \cdot D^3/6$
 $L_c=V/A$
 $Bi=(h \cdot L_c)/k$ "if $Bi < 0.1$, the lumped sytem analysis is applicable"
 $b=(h \cdot A)/(\rho \cdot c_p \cdot V)$
 $(T_f - T_{\text{infinity}})/(T_i - T_{\text{infinity}}) = \exp(-b \cdot \text{time})$
 $m=\rho \cdot V$
 $Q=m \cdot c_p \cdot (T_i - T_f)$
 $Q_{\text{dot}}=n_{\text{dot_ball}} \cdot Q \cdot \text{Convert(J/h, W)}$

T_i [C]	time [s]	Q [W]
500	127.4	271.2
550	134	305.1
600	140	339
650	145.5	372.9
700	150.6	406.9
750	155.3	440.8
800	159.6	474.7
850	163.7	508.6
900	167.6	542.5
950	171.2	576.4
1000	174.7	610.3



11-27 An electronic device is on for 5 minutes, and off for several hours. The temperature of the device at the end of the 5-min operating period is to be determined for the cases of operation with and without a heat sink.

Assumptions 1 The device and the heat sink are isothermal. 2 The thermal properties of the device and of the sink are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface.

Properties The specific heat of the device is given to be $c_p = 850 \text{ J/kg} \cdot ^\circ\text{C}$. The specific heat of the aluminum sink is $903 \text{ J/kg} \cdot ^\circ\text{C}$ (Table A-24), but can be taken to be $850 \text{ J/kg} \cdot ^\circ\text{C}$ for simplicity in analysis.

Analysis (a) Approximate solution

This problem can be solved approximately by using an average temperature for the device when evaluating the heat loss. An energy balance on the device can be expressed as

$$E_{\text{in}} - E_{\text{out}} + E_{\text{generation}} = \Delta E_{\text{device}} \longrightarrow -\dot{Q}_{\text{out}} \Delta t + \dot{E}_{\text{generation}} \Delta t = mc_p \Delta T_{\text{device}}$$

$$\text{or,} \quad \dot{E}_{\text{generation}} \Delta t - hA_s \left(\frac{T + T_\infty}{2} - T_\infty \right) \Delta t = mc_p (T - T_\infty)$$

Substituting the given values,

$$(20 \text{ J/s})(5 \times 60 \text{ s}) - (12 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0004 \text{ m}^2) \left(\frac{T - 25}{2} \right) ^\circ\text{C}(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg} \cdot ^\circ\text{C})(T - 25)^\circ\text{C}$$

which gives $T = 363.6^\circ\text{C}$

If the device were attached to an aluminum heat sink, the temperature of the device would be

$$\begin{aligned} (20 \text{ J/s})(5 \times 60 \text{ s}) - (12 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0084 \text{ m}^2) \left(\frac{T - 25}{2} \right) ^\circ\text{C}(5 \times 60 \text{ s}) \\ = (0.20 + 0.02) \text{ kg} \times (850 \text{ J/kg} \cdot ^\circ\text{C})(T - 25)^\circ\text{C} \end{aligned}$$

which gives $T = 54.7^\circ\text{C}$

Note that the temperature of the electronic device drops considerably as a result of attaching it to a heat sink.

(b) Exact solution

This problem can be solved exactly by obtaining the differential equation from an energy balance on the device for a differential time interval dt . We will get

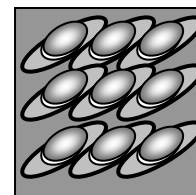
$$\frac{d(T - T_\infty)}{dt} + \frac{hA_s}{mc_p} (T - T_\infty) = \frac{\dot{E}_{\text{generation}}}{mc_p}$$

It can be solved to give

$$T(t) = T_\infty + \frac{\dot{E}_{\text{generation}}}{hA_s} \left(1 - \exp\left(-\frac{hA_s}{mc_p} t\right) \right)$$

Substituting the known quantities and solving for t gives 363.4°C for the first case and 54.6°C for the second case, which are practically identical to the results obtained from the approximate analysis.

Electronic
device
20 W



Transient Heat Conduction in Large Plane Walls, Long Cylinders, and Spheres with Spatial Effects

11-28C A cylinder whose diameter is small relative to its length can be treated as an infinitely long cylinder. When the diameter and length of the cylinder are comparable, it is not proper to treat the cylinder as being infinitely long. It is also not proper to use this model when finding the temperatures near the bottom or top surfaces of a cylinder since heat transfer at those locations can be two-dimensional.

11-29C Yes. A plane wall whose one side is insulated is equivalent to a plane wall that is twice as thick and is exposed to convection from both sides. The midplane in the latter case will behave like an insulated surface because of thermal symmetry.

11-30C The solution for determination of the one-dimensional transient temperature distribution involves many variables that make the graphical representation of the results impractical. In order to reduce the number of parameters, some variables are grouped into dimensionless quantities.

11-31C The Fourier number is a measure of heat conducted through a body relative to the heat stored. Thus a large value of Fourier number indicates faster propagation of heat through body. Since Fourier number is proportional to time, doubling the time will also double the Fourier number.

11-32C This case can be handled by setting the heat transfer coefficient h to infinity ∞ since the temperature of the surrounding medium in this case becomes equivalent to the surface temperature.

11-33C The maximum possible amount of heat transfer will occur when the temperature of the body reaches the temperature of the medium, and can be determined from $Q_{\max} = mc_p(T_{\infty} - T_i)$.

11-34C When the Biot number is less than 0.1, the temperature of the sphere will be nearly uniform at all times. Therefore, it is more convenient to use the lumped system analysis in this case.

11-35 A student calculates the total heat transfer from a spherical copper ball. It is to be determined whether his/her result is reasonable.

Assumptions The thermal properties of the copper ball are constant at room temperature.

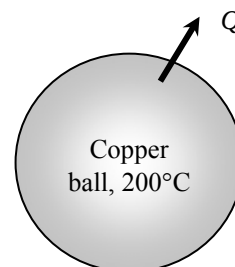
Properties The density and specific heat of the copper ball are $\rho = 8933 \text{ kg/m}^3$, and $c_p = 0.385 \text{ kJ/kg} \cdot ^\circ\text{C}$ (Table A-24).

Analysis The mass of the copper ball and the maximum amount of heat transfer from the copper ball are

$$m = \rho V = \rho \left(\frac{\pi D^3}{6} \right) = (8933 \text{ kg/m}^3) \left[\frac{\pi (0.18 \text{ m})^3}{6} \right] = 27.28 \text{ kg}$$

$$Q_{\max} = mc_p [T_i - T_{\infty}] = (27.28 \text{ kg})(0.385 \text{ kJ/kg} \cdot ^\circ\text{C})(200 - 25)^\circ\text{C} = 1838 \text{ kJ}$$

Discussion The student's result of 3150 kJ is **not reasonable** since it is greater than the maximum possible amount of heat transfer.



11-36 Tomatoes are placed into cold water to cool them. The heat transfer coefficient and the amount of heat transfer are to be determined.

Assumptions 1 The tomatoes are spherical in shape. 2 Heat conduction in the tomatoes is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the tomatoes are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

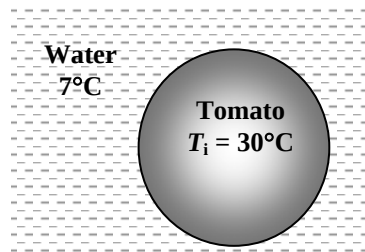
Properties The properties of the tomatoes are given to be $k = 0.59 \text{ W/m} \cdot ^\circ\text{C}$, $\alpha = 0.141 \times 10^{-6} \text{ m}^2/\text{s}$, $\rho = 999 \text{ kg/m}^3$ and $c_p = 3.99 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.141 \times 10^{-6} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}{(0.04 \text{ m})^2} = 0.635$$

which is greater than 0.2. Therefore one-term solution is applicable. The ratio of the dimensionless temperatures at the surface and center of the tomatoes are

$$\frac{\theta_{s,\text{sph}}}{\theta_{0,\text{sph}}} = \frac{\frac{T_s - T_\infty}{T_i - T_\infty}}{\frac{T_0 - T_\infty}{T_i - T_\infty}} = \frac{T_s - T_\infty}{T_0 - T_\infty} = \frac{A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1)}{\lambda_1}}{A_1 e^{-\lambda_1^2 \tau}} = \frac{\sin(\lambda_1)}{\lambda_1}$$



Substituting,

$$\frac{7.1 - 7}{10 - 7} = \frac{\sin(\lambda_1)}{\lambda_1} \longrightarrow \lambda_1 = 3.0401$$

From Table 11-2, the corresponding Biot number and the heat transfer coefficient are

$$\text{Bi} = 31.1$$

$$\text{Bi} = \frac{hr_o}{k} \longrightarrow h = \frac{k\text{Bi}}{r_o} = \frac{(0.59 \text{ W/m} \cdot ^\circ\text{C})(31.1)}{(0.04 \text{ m})} = 459 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The maximum amount of heat transfer is

$$m = 8\rho V = 8\rho \pi D^3 / 6 = 8(999 \text{ kg/m}^3)[\pi(0.08 \text{ m})^3 / 6] = 2.143 \text{ kg}$$

$$Q_{\max} = mc_p [T_i - T_\infty] = (2.143 \text{ kg})(3.99 \text{ kJ/kg} \cdot ^\circ\text{C})(30 - 7)^\circ\text{C} = 196.6 \text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\left(\frac{Q}{Q_{\max}} \right)_{\text{cyl}} = 1 - 3 \left(\frac{T_0 - T_\infty}{T_i - T_\infty} \right) \frac{\sin \lambda_1 - \lambda_1 \cos \lambda_1}{\lambda_1^3} = 1 - 3 \left(\frac{10 - 7}{30 - 7} \right) \frac{\sin(3.0401) - (3.0401) \cos(3.0401)}{(3.0401)^3} = 0.9565$$

$$Q = 0.9565 Q_{\max}$$

$$Q = 0.9565(196.6 \text{ kJ}) = 188 \text{ kJ}$$

11-37 An egg is dropped into boiling water. The cooking time of the egg is to be determined.

Assumptions **1** The egg is spherical in shape with a radius of $r_o = 2.75$ cm. **2** Heat conduction in the egg is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the egg are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and diffusivity of the eggs are given to be $k = 0.6$ W/m.°C and $\alpha = 0.14 \times 10^{-6}$ m²/s.

Analysis The Biot number for this process is

$$Bi = \frac{hr_o}{k} = \frac{(1400 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0275 \text{ m})}{(0.6 \text{ W/m} \cdot ^\circ\text{C})} = 64.2$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

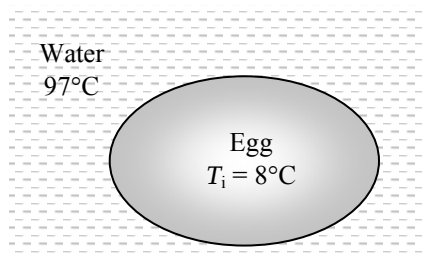
$$\lambda_1 = 3.0877 \quad \text{and} \quad A_1 = 1.9969$$

Then the Fourier number becomes

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{70 - 97}{8 - 97} = (1.9969)e^{-(3.0877)^2 \tau} \longrightarrow \tau = 0.198 \approx 0.2$$

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the time required for the temperature of the center of the egg to reach 70°C is determined to be

$$t = \frac{\tau_o^2}{\alpha} = \frac{(0.198)(0.0275 \text{ m})^2}{(0.14 \times 10^{-6} \text{ m}^2/\text{s})} = 1070 \text{ s} = \mathbf{17.8 \text{ min}}$$



11-38 EES Prob. 11-37 is reconsidered. The effect of the final center temperature of the egg on the time it will take for the center to reach this temperature is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$D=0.055$ [m]
 $T_i=8$ [C]
 $T_o=70$ [C]
 $T_{\text{infinity}}=97$ [C]
 $h=1400$ [W/m²-C]

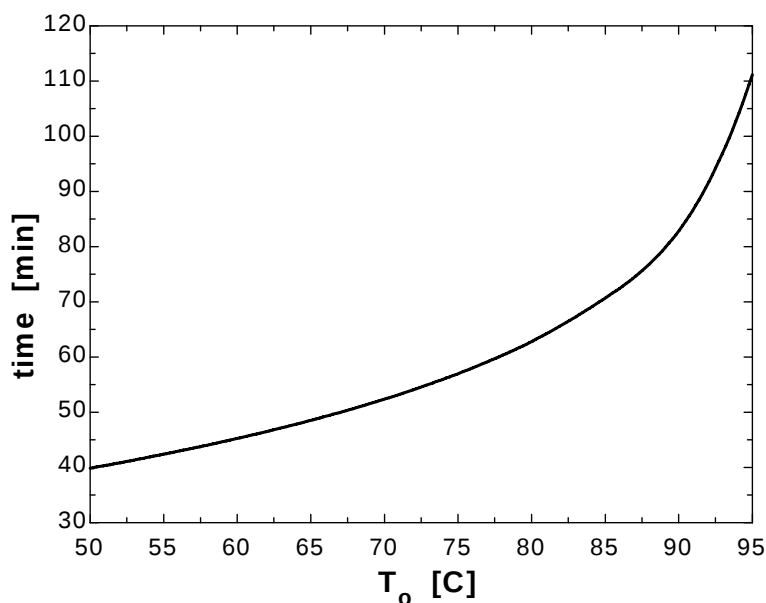
"PROPERTIES"

$k=0.6$ [W/m-C]
 $\alpha=0.14\text{E-}6$ [m²/s]

"ANALYSIS"

$Bi=(h*r_o)/k$
 $r_o=D/2$
 "From Table 11-2 corresponding to this Bi number, we read"
 $\lambda_1=1.9969$
 $A_1=3.0863$
 $(T_o-T_{\text{infinity}})/(T_i-T_{\text{infinity}})=A_1*\exp(-\lambda_1^2*\tau)$
 $\text{time}=(\tau*r_o^2)/\alpha*\text{Convert(s, min)}$

T_o [C]	time [min]
50	39.86
55	42.4
60	45.26
65	48.54
70	52.38
75	57
80	62.82
85	70.68
90	82.85
95	111.1



11-39 Large brass plates are heated in an oven. The surface temperature of the plates leaving the oven is to be determined.

Assumptions 1 Heat conduction in the plate is one-dimensional since the plate is large relative to its thickness and there is thermal symmetry about the center plane. 3 The thermal properties of the plate are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of brass at room temperature are given to be $k = 110 \text{ W/m} \cdot ^\circ\text{C}$, $\alpha = 33.9 \times 10^{-6} \text{ m}^2/\text{s}$

Analysis The Biot number for this process is

$$Bi = \frac{hL}{k} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.015 \text{ m})}{(110 \text{ W/m} \cdot ^\circ\text{C})} = 0.0109$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.1035 \quad \text{and} \quad A_1 = 1.0018$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(33.9 \times 10^{-6} \text{ m}^2/\text{s})(10 \text{ min} \times 60 \text{ s/min})}{(0.015 \text{ m})^2} = 90.4 > 0.2$$

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the temperature at the surface of the plates becomes

$$\theta(L, t)_{\text{wall}} = \frac{T(x, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L / L) = (1.0018) e^{-(0.1035)^2 (90.4)} \cos(0.1035) = 0.378$$

$$\frac{T(L, t) - 700}{25 - 700} = 0.378 \longrightarrow T(L, t) = \mathbf{445^\circ\text{C}}$$

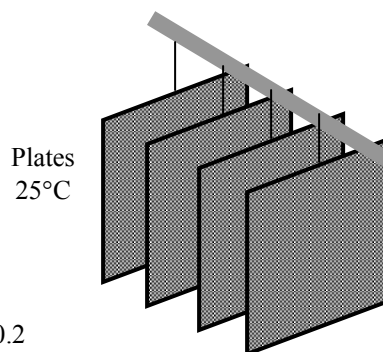
Discussion This problem can be solved easily using the lumped system analysis since $Bi < 0.1$, and thus the lumped system analysis is applicable. It gives

$$\alpha = \frac{k}{\rho c_p} \rightarrow \rho c_p = \frac{k}{\alpha} = \frac{110 \text{ W/m} \cdot ^\circ\text{C}}{33.9 \times 10^{-6} \text{ m}^2/\text{s}} = 3.245 \times 10^6 \text{ W} \cdot \text{s/m}^3 \cdot ^\circ\text{C}$$

$$b = \frac{hA}{\rho V c_p} = \frac{hA}{\rho(LA)c_p} = \frac{h}{\rho L c_p} = \frac{h}{L(k/\alpha)} = \frac{80 \text{ W/m}^2 \cdot ^\circ\text{C}}{(0.015 \text{ m})(3.245 \times 10^6 \text{ W} \cdot \text{s/m}^3 \cdot ^\circ\text{C})} = 0.001644 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \rightarrow T(t) = T_\infty + (T_i - T_\infty) e^{-bt} = 700^\circ\text{C} + (25 - 700^\circ\text{C}) e^{-(0.001644 \text{ s}^{-1})(600 \text{ s})} = \mathbf{448^\circ\text{C}}$$

which is almost identical to the result obtained above.



11-40 EES Prob. 11-39 is reconsidered. The effects of the temperature of the oven and the heating time on the final surface temperature of the plates are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$L = (0.03/2)$ [m]
 $T_i = 25$ [C]
 $T_{\infty} = 700$ [C]
 $\text{time} = 10$ [min]
 $h = 80$ [W/m²-C]

"PROPERTIES"

$k = 110$ [W/m-C]
 $\alpha = 33.9 \times 10^{-6}$ [m²/s]

"ANALYSIS"

$Bi = (h \cdot L) / k$

"From Table 11-2, corresponding to this Bi number, we read"

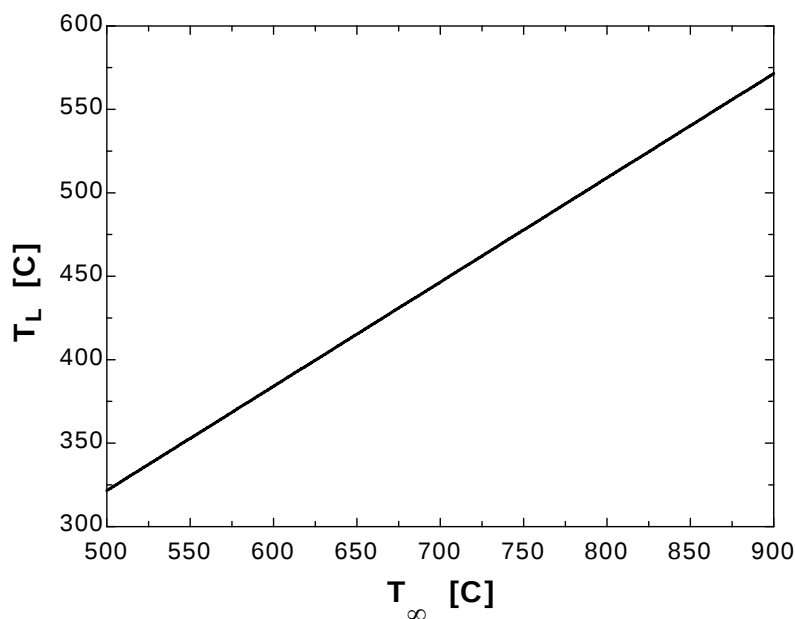
$\lambda_1 = 0.1039$

$A_1 = 1.0018$

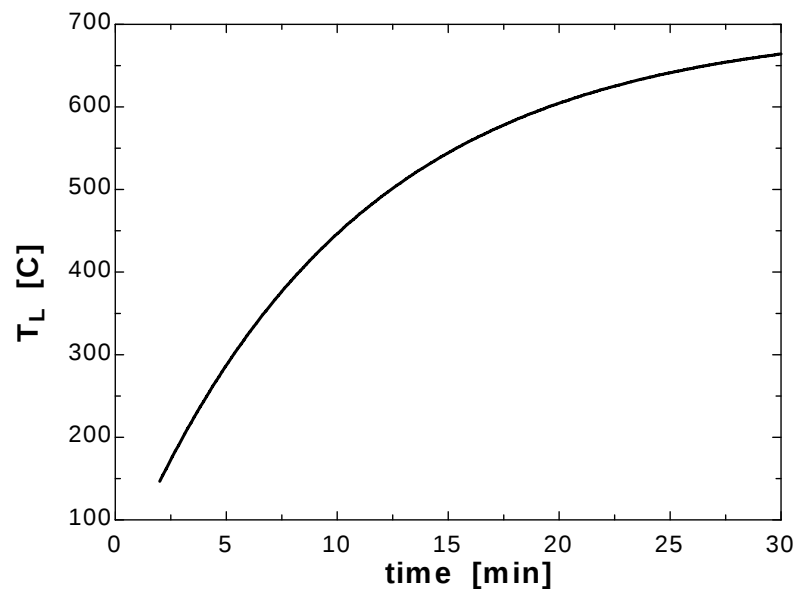
$\tau = (\alpha \cdot \text{time} \cdot \text{Convert}(\text{min}, \text{s})) / L^2$

$(T_L - T_{\infty}) / (T_i - T_{\infty}) = A_1 \cdot \exp(-\lambda_1^2 \cdot \tau) \cdot \text{Cos}(\lambda_1 \cdot L / L)$

T_{∞} [C]	T_L [C]
500	321.6
525	337.2
550	352.9
575	368.5
600	384.1
625	399.7
650	415.3
675	430.9
700	446.5
725	462.1
750	477.8
775	493.4
800	509
825	524.6
850	540.2
875	555.8
900	571.4



time [min]	T_L [C]
2	146.7
4	244.8
6	325.5
8	391.9
10	446.5
12	491.5
14	528.5
16	558.9
18	583.9
20	604.5
22	621.4
24	635.4
26	646.8
28	656.2
30	664



11-41 A long cylindrical shaft at 400°C is allowed to cool slowly. The center temperature and the heat transfer per unit length of the cylinder are to be determined.

Assumptions 1 Heat conduction in the shaft is one-dimensional since it is long and it has thermal symmetry about the center line. 2 The thermal properties of the shaft are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of stainless steel 304 at room temperature are given to be $k = 14.9 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 7900 \text{ kg/m}^3$, $c_p = 477 \text{ J/kg} \cdot ^\circ\text{C}$, $\alpha = 3.95 \times 10^{-6} \text{ m}^2/\text{s}$

Analysis First the Biot number is calculated to be

$$Bi = \frac{hr_o}{k} = \frac{(60 \text{ W/m}^2 \cdot ^\circ\text{C})(0.175 \text{ m})}{(14.9 \text{ W/m} \cdot ^\circ\text{C})} = 0.705$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.0904 \quad \text{and} \quad A_1 = 1.1548$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(3.95 \times 10^{-6} \text{ m}^2/\text{s})(20 \times 60 \text{ s})}{(0.175 \text{ m})^2} = 0.1548$$

which is very close to the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the temperature at the center of the shaft becomes

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.1548)e^{-(1.0904)^2 (0.1548)} = 0.9607$$

$$\frac{T_0 - 150}{400 - 150} = 0.9607 \longrightarrow T_0 = \mathbf{390^\circ\text{C}}$$

The maximum heat can be transferred from the cylinder per meter of its length is

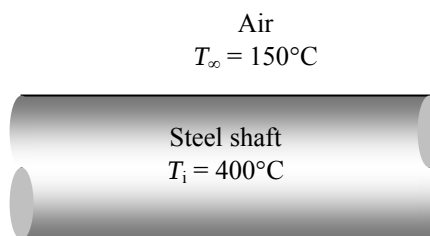
$$m = \rho V = \rho \pi r_o^2 L = (7900 \text{ kg/m}^3)[\pi(0.175 \text{ m})^2 (1 \text{ m})] = 760.1 \text{ kg}$$

$$Q_{\max} = mc_p [T_\infty - T_i] = (760.1 \text{ kg})(0.477 \text{ kJ/kg} \cdot ^\circ\text{C})(400 - 150)^\circ\text{C} = 90,640 \text{ kJ}$$

Once the constant $J_1 = 0.4679$ is determined from Table 11-3 corresponding to the constant $\lambda_1 = 1.0904$, the actual heat transfer becomes

$$\left(\frac{Q}{Q_{\max}} \right)_{\text{cyl}} = 1 - 2 \left(\frac{T_o - T_\infty}{T_i - T_\infty} \right) \frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2 \left(\frac{390 - 150}{400 - 150} \right) \frac{0.4679}{1.0904} = 0.1761$$

$$Q = 0.1761(90,640 \text{ kJ}) = \mathbf{15,960 \text{ kJ}}$$



11-42 EES Prob. 11-41 is reconsidered. The effect of the cooling time on the final center temperature of the shaft and the amount of heat transfer is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$r_o = (0.35/2)$ [m]
 $T_i = 400$ [C]
 $T_{\text{infinity}} = 150$ [C]
 $h = 60$ [W/m²-C]
 $\text{time} = 20$ [min]

"PROPERTIES"

$k = 14.9$ [W/m-C]
 $\rho = 7900$ [kg/m³]
 $c_p = 477$ [J/kg-C]
 $\alpha = 3.95\text{E-}6$ [m²/s]

"ANALYSIS"

$Bi = (h \cdot r_o) / k$

"From Table 11-2 corresponding to this Bi number, we read"

$\lambda_1 = 1.0935$

$A_1 = 1.1558$

$J_1 = 0.4709$ "From Table 11-3, corresponding to λ_1 "

$\tau = (\alpha \cdot \text{time} \cdot \text{Convert}(\text{min}, \text{s})) / r_o^2$

$(T_o - T_{\text{infinity}}) / (T_i - T_{\text{infinity}}) = A_1 \cdot \exp(-\lambda_1^2 \cdot \tau)$

$L = 1$ "[m], 1 m length of the cylinder is considered"

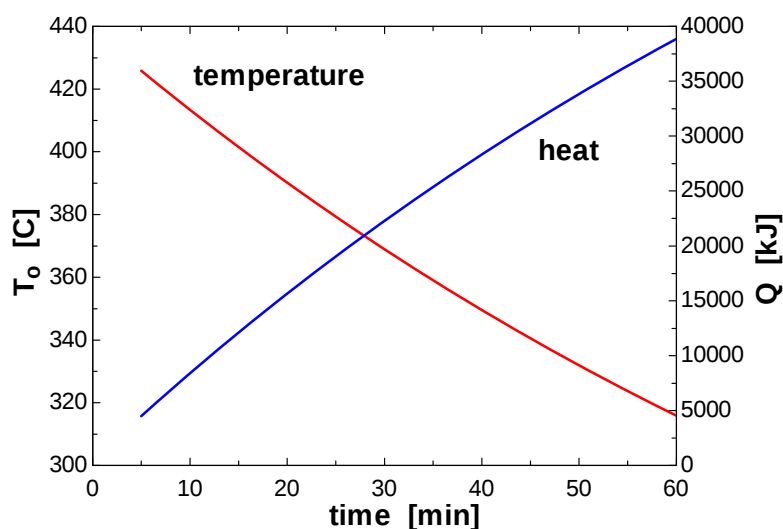
$V = \pi \cdot r_o^2 \cdot L$

$m = \rho \cdot V$

$Q_{\text{max}} = m \cdot c_p \cdot (T_i - T_{\text{infinity}}) \cdot \text{Convert}(\text{J}, \text{kJ})$

$Q / Q_{\text{max}} = 1 - 2 \cdot (T_o - T_{\text{infinity}}) / (T_i - T_{\text{infinity}}) \cdot J_1 / \lambda_1$

time [min]	T_o [C]	Q [kJ]
5	425.9	4491
10	413.4	8386
15	401.5	12105
20	390.1	15656
25	379.3	19046
30	368.9	22283
35	359	25374
40	349.6	28325
45	340.5	31142
50	331.9	33832
55	323.7	36401
60	315.8	38853



11-43E Long cylindrical steel rods are heat-treated in an oven. Their centerline temperature when they leave the oven is to be determined.

Assumptions 1 Heat conduction in the rods is one-dimensional since the rods are long and they have thermal symmetry about the center line. 2 The thermal properties of the rod are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

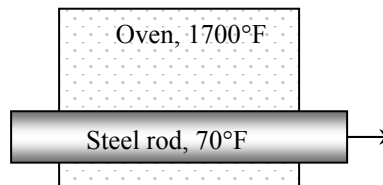
Properties The properties of AISI stainless steel rods are given to be $k = 7.74 \text{ Btu/h.ft.}^\circ\text{F}$, $\alpha = 0.135 \text{ ft}^2/\text{h}$.

Analysis The time the steel rods stays in the oven can be determined from

$$t = \frac{\text{length}}{\text{velocity}} = \frac{21 \text{ ft}}{7 \text{ ft/min}} = 3 \text{ min} = 180 \text{ s}$$

The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(20 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(2/12 \text{ ft})}{(7.74 \text{ Btu/h.ft.}^\circ\text{F})} = 0.4307$$



The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.8790 \quad \text{and} \quad A_1 = 1.0996$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.135 \text{ ft}^2/\text{h})(3/60 \text{ h})}{(2/12 \text{ ft})^2} = 0.243$$

Then the temperature at the center of the rods becomes

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.0996)e^{-(0.8790)^2 (0.243)} = 0.911$$

$$\frac{T_0 - 1700}{70 - 1700} = 0.911 \longrightarrow T_0 = 215^\circ\text{F}$$

11-44 Steaks are cooled by passing them through a refrigeration room. The time of cooling is to be determined.

Assumptions **1** Heat conduction in the steaks is one-dimensional since the steaks are large relative to their thickness and there is thermal symmetry about the center plane. **3** The thermal properties of the steaks are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of steaks are given to be $k = 0.45 \text{ W/m}\cdot^\circ\text{C}$ and $\alpha = 0.91 \times 10^{-7} \text{ m}^2/\text{s}$

Analysis The Biot number is

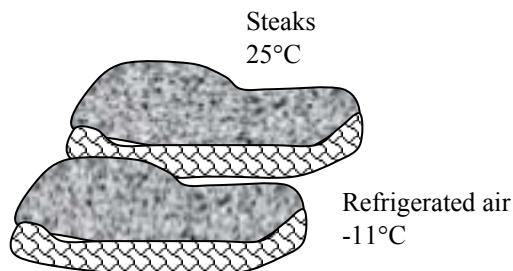
$$Bi = \frac{hL}{k} = \frac{(9 \text{ W/m}^2\cdot^\circ\text{C})(0.01 \text{ m})}{(0.45 \text{ W/m}\cdot^\circ\text{C})} = 0.200$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.4328 \quad \text{and} \quad A_1 = 1.0311$$

The Fourier number is

$$\begin{aligned} \frac{T(L, t) - T_\infty}{T_i - T_\infty} &= A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L / L) \\ \frac{2 - (-11)}{25 - (-11)} &= (1.0311) e^{-(0.4328)^2 \tau} \cos(0.4328) \longrightarrow \tau = 5.085 > 0.2 \end{aligned}$$



Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the length of time for the steaks to be kept in the refrigerator is determined to be

$$t = \frac{\tau L^2}{\alpha} = \frac{(5.085)(0.01 \text{ m})^2}{0.91 \times 10^{-7} \text{ m}^2/\text{s}} = 5590 \text{ s} = \mathbf{93.1 \text{ min}}$$

11-45 A long cylindrical wood log is exposed to hot gases in a fireplace. The time for the ignition of the wood is to be determined.

Assumptions 1 Heat conduction in the wood is one-dimensional since it is long and it has thermal symmetry about the center line. 2 The thermal properties of the wood are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of wood are given to be $k = 0.17 \text{ W/m}\cdot^\circ\text{C}$, $\alpha = 1.28 \times 10^{-7} \text{ m}^2/\text{s}$

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(13.6 \text{ W/m}^2 \cdot ^\circ\text{C})(0.05 \text{ m})}{(0.17 \text{ W/m}\cdot^\circ\text{C})} = 4.00$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

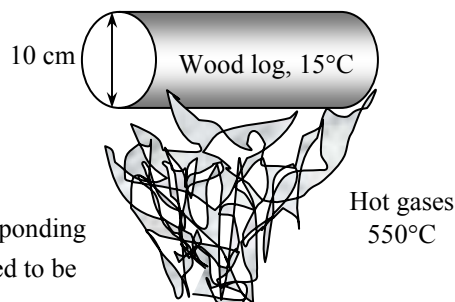
$$\lambda_1 = 1.9081 \quad \text{and} \quad A_1 = 1.4698$$

Once the constant J_0 is determined from Table 11-3 corresponding to the constant $\lambda_1 = 1.9081$, the Fourier number is determined to be

$$\begin{aligned} \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} &= A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r_o / r_o) \\ \frac{420 - 550}{15 - 550} &= (1.4698) e^{-(1.9081)^2 \tau} (0.2771) \longrightarrow \tau = 0.142 \end{aligned}$$

which is not above the value of 0.2 but it is close. We use one-term approximate solution (or the transient temperature charts) knowing that the result may be somewhat in error. Then the length of time before the log ignites is

$$t = \frac{\tau_o^2}{\alpha} = \frac{(0.142)(0.05 \text{ m})^2}{(1.28 \times 10^{-7} \text{ m}^2/\text{s})} = 2770 \text{ s} = \mathbf{46.2 \text{ min}}$$



11-46 A rib is roasted in an oven. The heat transfer coefficient at the surface of the rib, the temperature of the outer surface of the rib and the amount of heat transfer when it is rare done are to be determined. The time it will take to roast this rib to medium level is also to be determined.

Assumptions 1 The rib is a homogeneous spherical object. 2 Heat conduction in the rib is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the rib are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the rib are given to be $k = 0.45 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 1200 \text{ kg/m}^3$, $c_p = 4.1 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 0.91 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis (a) The radius of the roast is determined to be

$$m = \rho V \longrightarrow V = \frac{m}{\rho} = \frac{3.2 \text{ kg}}{1200 \text{ kg/m}^3} = 0.002667 \text{ m}^3$$

$$V = \frac{4}{3} \pi r_o^3 \longrightarrow r_o = \sqrt[3]{\frac{3V}{4\pi}} = \sqrt[3]{\frac{3(0.002667 \text{ m}^3)}{4\pi}} = 0.08603 \text{ m}$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.91 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 + 45 \times 60) \text{ s}}{(0.08603 \text{ m})^2} = 0.1217$$

which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the one-term solution can be written in the form

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{60 - 163}{4.5 - 163} = 0.65 = A_1 e^{-\lambda_1^2 (0.1217)}$$

It is determined from Table 11-2 by trial and error that this equation is satisfied when $Bi = 30$, which corresponds to $\lambda_1 = 3.0372$ and $A_1 = 1.9898$. Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.45 \text{ W/m} \cdot ^\circ\text{C})(30)}{(0.08603 \text{ m})} = \mathbf{156.9 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

This value seems to be larger than expected for problems of this kind. This is probably due to the Fourier number being less than 0.2.

(b) The temperature at the surface of the rib is

$$\theta(r_o, t)_{sph} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.9898) e^{-(3.0372)^2 (0.1217)} \frac{\sin(3.0372 \text{ rad})}{3.0372}$$

$$\frac{T(r_o, t) - 163}{4.5 - 163} = 0.0222 \longrightarrow T(r_o, t) = \mathbf{159.5^\circ\text{C}}$$

(c) The maximum possible heat transfer is

$$Q_{\max} = mc_p (T_\infty - T_i) = (3.2 \text{ kg})(4.1 \text{ kJ/kg} \cdot ^\circ\text{C})(163 - 4.5)^\circ\text{C} = 2080 \text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{0,sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3} = 1 - 3(0.65) \frac{\sin(3.0372) - (3.0372) \cos(3.0372)}{(3.0372)^3} = 0.783$$

$$Q = 0.783 Q_{\max} = (0.783)(2080 \text{ kJ}) = \mathbf{1629 \text{ kJ}}$$

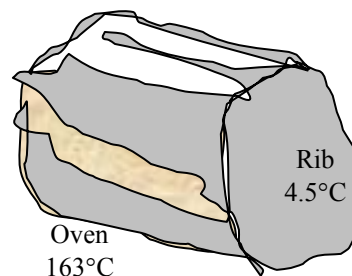
(d) The cooking time for medium-done rib is determined to be

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{71 - 163}{4.5 - 163} = (1.9898) e^{-(3.0372)^2 \tau} \longrightarrow \tau = 0.1336$$

$$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.1336)(0.08603 \text{ m})^2}{(0.91 \times 10^{-7} \text{ m}^2/\text{s})} = 10,866 \text{ s} = 181 \text{ min} \cong \mathbf{3 \text{ hr}}$$

This result is close to the listed value of 3 hours and 20 minutes. The difference between the two results is due to the Fourier number being less than 0.2 and thus the error in the one-term approximation.

Discussion The temperature of the outer parts of the rib is greater than that of the inner parts of the rib after it is taken out of the oven. Therefore, there will be a heat transfer from outer parts of the rib to the inner parts as a result of this temperature difference. The recommendation is logical.



11-47 A rib is roasted in an oven. The heat transfer coefficient at the surface of the rib, the temperature of the outer surface of the rib and the amount of heat transfer when it is well-done are to be determined. The time it will take to roast this rib to medium level is also to be determined.

Assumptions 1 The rib is a homogeneous spherical object. 2 Heat conduction in the rib is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the rib are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the rib are given to be $k = 0.45 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 1200 \text{ kg/m}^3$, $c_p = 4.1 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 0.91 \times 10^{-7} \text{ m}^2/\text{s}$

Analysis (a) The radius of the rib is determined to be

$$m = \rho V \longrightarrow V = \frac{m}{\rho} = \frac{3.2 \text{ kg}}{1200 \text{ kg/m}^3} = 0.00267 \text{ m}^3$$

$$V = \frac{4}{3} \pi r_o^3 \longrightarrow r_o = \sqrt[3]{\frac{3V}{4\pi}} = \sqrt[3]{\frac{3(0.00267 \text{ m}^3)}{4\pi}} = 0.08603 \text{ m}$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.91 \times 10^{-7} \text{ m}^2/\text{s})(4 \times 3600 + 15 \times 60) \text{ s}}{(0.08603 \text{ m})^2} = 0.1881$$

which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the one-term solution formulation can be written in the form

$$\theta_{0,sph} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{77 - 163}{4.5 - 163} = 0.543 = A_1 e^{-\lambda_1^2 (0.1881)}$$

It is determined from Table 11-2 by trial and error that this equation is satisfied when $Bi = 4.3$, which corresponds to $\lambda_1 = 2.4900$ and $A_1 = 1.7402$. Then the heat transfer coefficient can be determined from.

$$Bi = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.45 \text{ W/m} \cdot ^\circ\text{C})(4.3)}{(0.08603 \text{ m})} = 22.5 \text{ W/m}^2 \cdot ^\circ\text{C}$$

(b) The temperature at the surface of the rib is

$$\theta(r_o, t)_{sph} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.7402) e^{-(2.49)^2 (0.1881)} \frac{\sin(2.49)}{2.49}$$

$$\frac{T(r_o, t) - 163}{4.5 - 163} = 0.132 \longrightarrow T(r_o, t) = 142.1^\circ\text{C}$$

(c) The maximum possible heat transfer is

$$Q_{\max} = mc_p (T_\infty - T_i) = (3.2 \text{ kg})(4.1 \text{ kJ/kg} \cdot ^\circ\text{C})(163 - 4.5)^\circ\text{C} = 2080 \text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{0,sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3} = 1 - 3(0.543) \frac{\sin(2.49) - (2.49) \cos(2.49)}{(2.49)^3} = 0.727$$

$$Q = 0.727 Q_{\max} = (0.727)(2080 \text{ kJ}) = 1512 \text{ kJ}$$

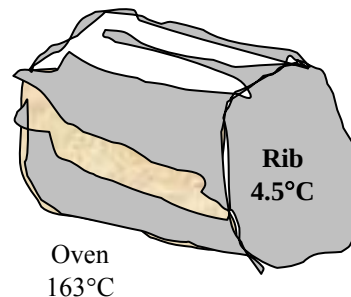
(d) The cooking time for medium-done rib is determined to be

$$\theta_{0,sph} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{77 - 163}{4.5 - 163} = (1.7402) e^{-(2.49)^2 \tau} \longrightarrow \tau = 0.177$$

$$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.177)(0.08603 \text{ m})^2}{(0.91 \times 10^{-7} \text{ m}^2/\text{s})} = 14,400 \text{ s} = 240 \text{ min} = 4 \text{ hr}$$

This result is close to the listed value of 4 hours and 15 minutes. The difference between the two results is probably due to the Fourier number being less than 0.2 and thus the error in the one-term approximation.

Discussion The temperature of the outer parts of the rib is greater than that of the inner parts of the rib after it is taken out of the oven. Therefore, there will be a heat transfer from outer parts of the rib to the inner parts as a result of this temperature difference. The recommendation is logical.



11-48 An egg is dropped into boiling water. The cooking time of the egg is to be determined.

Assumptions **1** The egg is spherical in shape with a radius of $r_0 = 2.75$ cm. **2** Heat conduction in the egg is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the egg are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and diffusivity of the eggs can be approximated by those of water at room temperature to be $k = 0.607$ W/m. $^{\circ}$ C, $\alpha = k / \rho c_p = 0.146 \times 10^{-6}$ m 2 /s (Table A-15).

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(800 \text{ W/m}^2 \cdot ^{\circ}\text{C})(0.0275 \text{ m})}{(0.607 \text{ W/m} \cdot ^{\circ}\text{C})} = 36.2$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

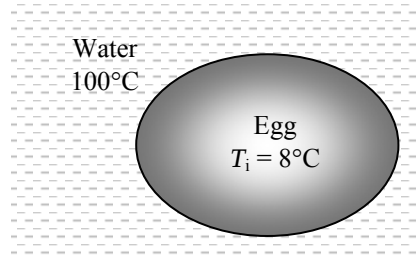
$$\lambda_1 = 3.0533 \quad \text{and} \quad A_1 = 1.9925$$

Then the Fourier number and the time period become

$$\theta_{0,sph} = \frac{T_0 - T_{\infty}}{T_i - T_{\infty}} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{60 - 100}{8 - 100} = (1.9925) e^{-(3.0533)^2 \tau} \longrightarrow \tau = 0.1633$$

which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the length of time for the egg to be kept in boiling water is determined to be

$$t = \frac{\tau_o^2}{\alpha} = \frac{(0.1633)(0.0275 \text{ m})^2}{0.146 \times 10^{-6} \text{ m}^2/\text{s}} = 846 \text{ s} = \mathbf{14.1 \text{ min}}$$



11-49 An egg is cooked in boiling water. The cooking time of the egg is to be determined for a location at 1610-m elevation.

Assumptions **1** The egg is spherical in shape with a radius of $r_o = 2.75$ cm. **2** Heat conduction in the egg is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the egg and heat transfer coefficient are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and diffusivity of the eggs can be approximated by those of water at room temperature to be $k = 0.607$ W/m. $^{\circ}$ C, $\alpha = k / \rho c_p = 0.146 \times 10^{-6}$ m 2 /s (Table A-15).

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(800 \text{ W/m}^2 \cdot ^{\circ}\text{C})(0.0275 \text{ m})}{(0.607 \text{ W/m} \cdot ^{\circ}\text{C})} = 36.2$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

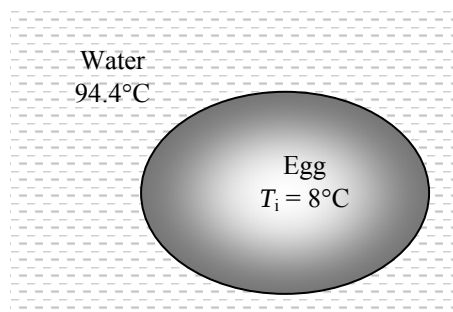
$$\lambda_1 = 3.0533 \quad \text{and} \quad A_1 = 1.9925$$

Then the Fourier number and the time period become

$$\theta_{0,sph} = \frac{T_0 - T_{\infty}}{T_i - T_{\infty}} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{60 - 94.4}{8 - 94.4} = (1.9925) e^{-(3.0533)^2 \tau} \longrightarrow \tau = 0.1727$$

which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the length of time for the egg to be kept in boiling water is determined to be

$$t = \frac{\tau_o^2}{\alpha} = \frac{(0.1727)(0.0275 \text{ m})^2}{(0.146 \times 10^{-6} \text{ m}^2/\text{s})} = 895 \text{ s} = \mathbf{14.9 \text{ min}}$$



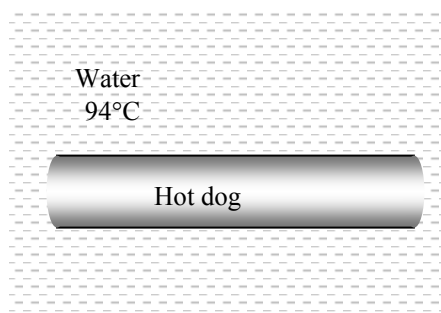
11-50 A hot dog is dropped into boiling water, and temperature measurements are taken at certain time intervals. The thermal diffusivity and thermal conductivity of the hot dog and the convection heat transfer coefficient are to be determined.

Assumptions 1 Heat conduction in the hot dog is one-dimensional since it is long and it has thermal symmetry about the centerline. 2 The thermal properties of the hot dog are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of hot dog available are given to be $\rho = 980 \text{ kg/m}^3$ and $c_p = 3900 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis (a) From Fig. 11-16b we have

$$\left. \begin{aligned} \frac{T - T_\infty}{T_0 - T_\infty} &= \frac{88 - 94}{59 - 94} = 0.17 \\ \frac{r}{r_o} &= \frac{r_o}{r_o} = 1 \end{aligned} \right\} \frac{1}{Bi} = \frac{k}{hr_o} = 0.15$$



The Fourier number is determined from Fig. 11-16a to be

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{k}{hr_o} = 0.15 \\ \frac{T_0 - T_\infty}{T_i - T_\infty} &= \frac{59 - 94}{20 - 94} = 0.47 \end{aligned} \right\} \tau = \frac{\alpha t}{r_o^2} = 0.20$$

The thermal diffusivity of the hot dog is determined to be

$$\frac{\alpha t}{r_o^2} = 0.20 \longrightarrow \alpha = \frac{0.2 r_o^2}{t} = \frac{(0.2)(0.011 \text{ m})^2}{120 \text{ s}} = \mathbf{2.017 \times 10^{-7} \text{ m}^2/\text{s}}$$

(b) The thermal conductivity of the hot dog is determined from

$$k = \alpha \rho c_p = (2.017 \times 10^{-7} \text{ m}^2/\text{s})(980 \text{ kg/m}^3)(3900 \text{ J/kg} \cdot ^\circ\text{C}) = \mathbf{0.771 \text{ W/m} \cdot ^\circ\text{C}}$$

(c) From part (a) we have $\frac{1}{Bi} = \frac{k}{hr_o} = 0.15$. Then,

$$\frac{k}{h} = 0.15 r_o = (0.15)(0.011 \text{ m}) = 0.00165 \text{ m}$$

Therefore, the heat transfer coefficient is

$$\frac{k}{h} = 0.00165 \longrightarrow h = \frac{0.771 \text{ W/m} \cdot ^\circ\text{C}}{0.00165 \text{ m}} = \mathbf{467 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

11-51 Using the data and the answers given in Prob. 11-50, the center and the surface temperatures of the hot dog 4 min after the start of the cooking and the amount of heat transferred to the hot dog are to be determined.

Assumptions 1 Heat conduction in the hot dog is one-dimensional since it is long and it has thermal symmetry about the center line. 2 The thermal properties of the hot dog are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of hot dog and the convection heat transfer coefficient are given or obtained in P11-50 to be $k = 0.771 \text{ W/m}\cdot^\circ\text{C}$, $\rho = 980 \text{ kg/m}^3$, $c_p = 3900 \text{ J/kg}\cdot^\circ\text{C}$, $\alpha = 2.017 \times 10^{-7} \text{ m}^2/\text{s}$, and $h = 467 \text{ W/m}^2\cdot^\circ\text{C}$.

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(467 \text{ W/m}^2\cdot^\circ\text{C})(0.011 \text{ m})}{(0.771 \text{ W/m}\cdot^\circ\text{C})} = 6.66$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 2.0785 \quad \text{and} \quad A_1 = 1.5357$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(2.017 \times 10^{-7} \text{ m}^2/\text{s})(4 \text{ min} \times 60 \text{ s/min})}{(0.011 \text{ m})^2} = 0.4001 > 0.2$$

Then the temperature at the center of the hot dog is determined to be

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5357)e^{-(2.0785)^2(0.4001)} = 0.2727$$

$$\frac{T_0 - 94}{20 - 94} = 0.2727 \longrightarrow T_0 = 73.8^\circ\text{C}$$

From Table 11-3 we read $J_0 = 0.1789$ corresponding to the constant $\lambda_1 = 2.0785$. Then the temperature at the surface of the hot dog becomes

$$\frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r_o / r_o) = (1.5357)e^{-(2.0785)^2(0.4001)}(0.1789) = 0.04878$$

$$\frac{T(r_o, t) - 94}{20 - 94} = 0.04878 \longrightarrow T(r_o, t) = 90.4^\circ\text{C}$$

The maximum possible amount of heat transfer is

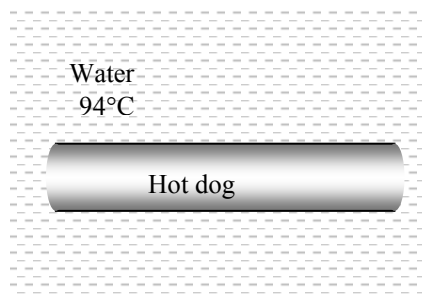
$$m = \rho V = \rho \pi r_o^2 L = (980 \text{ kg/m}^3) [\pi (0.011 \text{ m})^2 (0.125 \text{ m})] = 0.04657 \text{ kg}$$

$$Q_{\max} = mc_p (T_i - T_\infty) = (0.04657 \text{ kg})(3900 \text{ J/kg}\cdot^\circ\text{C})(94 - 20)^\circ\text{C} = 13,440 \text{ J}$$

From Table 11-3 we read $J_1 = 0.5701$ corresponding to the constant $\lambda_1 = 2.0785$. Then the actual heat transfer becomes

$$\left(\frac{Q}{Q_{\max}} \right)_{\text{cyl}} = 1 - 2\theta_{0,\text{cyl}} \frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2(0.2727) \frac{0.5701}{2.0785} = 0.8504$$

$$Q = 0.8504(13,440 \text{ kJ}) = 11,430 \text{ kJ}$$



11-52E Whole chickens are to be cooled in the racks of a large refrigerator. Heat transfer coefficient that will enable to meet temperature constraints of the chickens while keeping the refrigeration time to a minimum is to be determined.

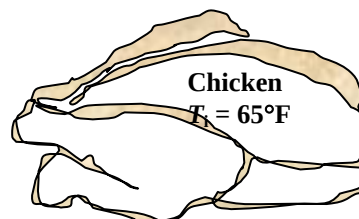
Assumptions 1 The chicken is a homogeneous spherical object. 2 Heat conduction in the chicken is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the chicken are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the chicken are given to be $k = 0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$, $\rho = 74.9 \text{ lbm/ft}^3$, $c_p = 0.98 \text{ Btu/lbm}\cdot^\circ\text{F}$, and $\alpha = 0.0035 \text{ ft}^2/\text{h}$.

Analysis The radius of the chicken is determined to be

$$m = \rho V \longrightarrow V = \frac{m}{\rho} = \frac{5 \text{ lbm}}{74.9 \text{ lbm/ft}^3} = 0.06676 \text{ ft}^3$$

$$V = \frac{4}{3} \pi r_o^3 \longrightarrow r_o = \sqrt[3]{\frac{3V}{4\pi}} = \sqrt[3]{\frac{3(0.06676 \text{ ft}^3)}{4\pi}} = 0.2517 \text{ ft}$$



From Fig. 11-17b we have

$$\left. \begin{aligned} \frac{T - T_\infty}{T_0 - T_\infty} &= \frac{35 - 5}{45 - 5} = 0.75 \\ \frac{x}{r_o} &= \frac{r_o}{r_o} = 1 \end{aligned} \right\} \frac{1}{Bi} = \frac{k}{hr_o} = 2$$

Then the heat transfer coefficients becomes

$$h = \frac{k}{2r_o} = \frac{0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{2(0.2517 \text{ ft})} = \mathbf{0.516 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}}$$

11-53 A person puts apples into the freezer to cool them quickly. The center and surface temperatures of the apples, and the amount of heat transfer from each apple in 1 h are to be determined.

Assumptions 1 The apples are spherical in shape with a diameter of 9 cm. 2 Heat conduction in the apples is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the apples are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the apples are given to be $k = 0.418 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 840 \text{ kg/m}^3$, $c_p = 3.81 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 1.3 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(8 \text{ W/m}^2 \cdot ^\circ\text{C})(0.045 \text{ m})}{(0.418 \text{ W/m} \cdot ^\circ\text{C})} = 0.861$$

The constants λ_1 and A_1 corresponding to this

Biot number are, from Table 11-2,

$$\lambda_1 = 1.476 \quad \text{and} \quad A_1 = 1.2390$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(1.3 \times 10^{-7} \text{ m}^2/\text{s})(1 \text{ h} \times 3600 \text{ s/h})}{(0.045 \text{ m})^2} = 0.231 > 0.2$$

Then the temperature at the center of the apples becomes

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - (-15)}{20 - (-15)} = (1.239)e^{-(1.476)^2(0.231)} = 0.749 \longrightarrow T_0 = 11.2^\circ\text{C}$$

The temperature at the surface of the apples is

$$\theta(r_o, t)_{sph} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.239)e^{-(1.476)^2(0.231)} \frac{\sin(1.476 \text{ rad})}{1.476} = 0.505$$

$$\frac{T(r_o, t) - (-15)}{20 - (-15)} = 0.505 \longrightarrow T(r_o, t) = 2.7^\circ\text{C}$$

The maximum possible heat transfer is

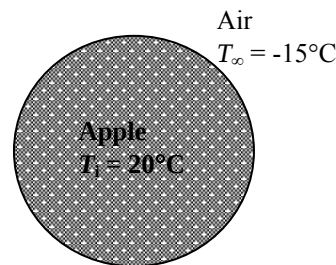
$$m = \rho V = \rho \frac{4}{3} \pi r_o^3 = (840 \text{ kg/m}^3) \left[\frac{4}{3} \pi (0.045 \text{ m})^3 \right] = 0.3206 \text{ kg}$$

$$Q_{\max} = mc_p (T_i - T_\infty) = (0.3206 \text{ kg})(3.81 \text{ kJ/kg} \cdot ^\circ\text{C})[20 - (-15)]^\circ\text{C} = 42.75 \text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{0,sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3} = 1 - 3(0.749) \frac{\sin(1.476 \text{ rad}) - (1.476) \cos(1.476 \text{ rad})}{(1.476)^3} = 0.402$$

$$Q = 0.402 Q_{\max} = (0.402)(42.75 \text{ kJ}) = 17.2 \text{ kJ}$$



11-54 EES Prob. 11-53 is reconsidered. The effect of the initial temperature of the apples on the final center and surface temperatures and the amount of heat transfer is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$T_{\infty} = -15$ [C]
 $T_i = 20$ [C]
 $h = 8$ [W/m²-C]
 $r_o = 0.09/2$ [m]
 $\text{time} = 1 \times 3600$ [s]

"PROPERTIES"

$k = 0.513$ [W/m-C]
 $\rho = 840$ [kg/m³]
 $c_p = 3.6$ [kJ/kg-C]
 $\alpha = 1.3 \times 10^{-7}$ [m²/s]

"ANALYSIS"

$Bi = (h \cdot r_o) / k$

"From Table 11-2 corresponding to this Bi number, we read"

$\lambda_1 = 1.3525$

$A_1 = 1.1978$

$\tau = (\alpha \cdot \text{time}) / r_o^2$

$(T_o - T_{\infty}) / (T_i - T_{\infty}) = A_1 \cdot \exp(-\lambda_1^2 \cdot \tau)$

$(T_r - T_{\infty}) / (T_i - T_{\infty}) = A_1 \cdot \exp(-\lambda_1^2 \cdot \tau) \cdot \sin(\lambda_1 \cdot r_o / r_o) / (\lambda_1 \cdot r_o / r_o)$

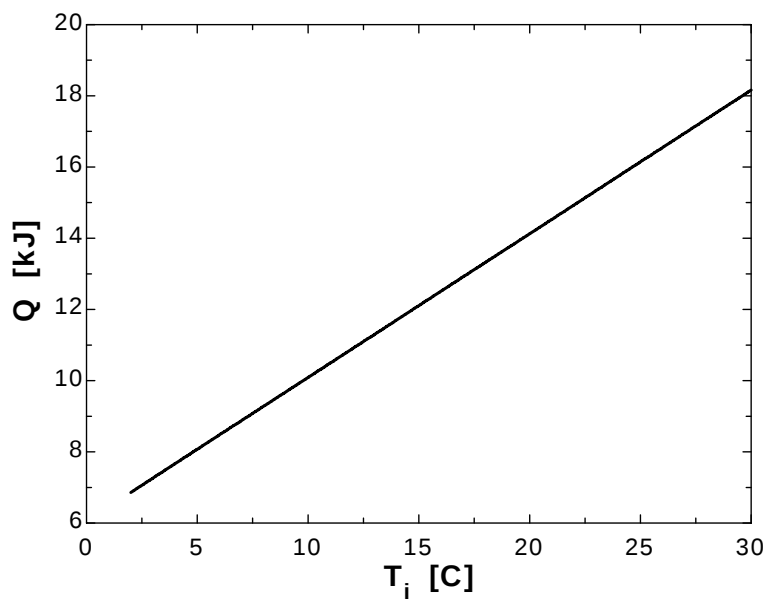
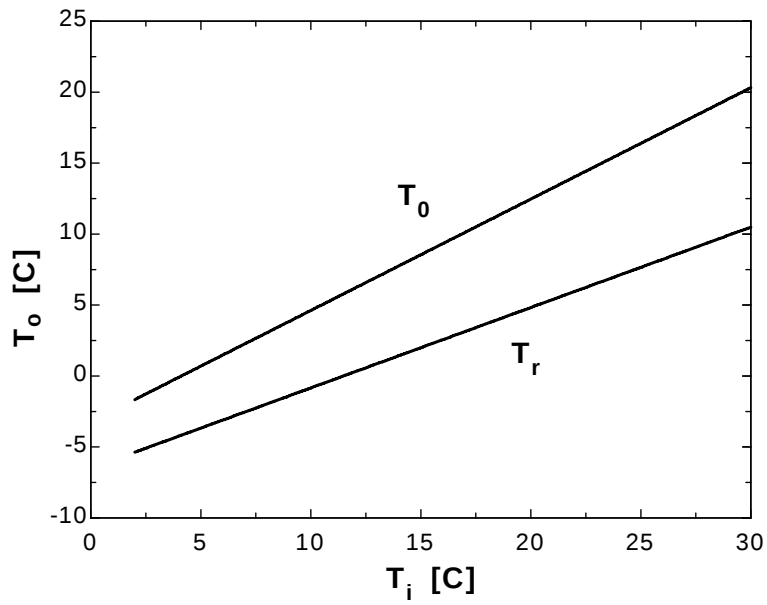
$V = 4/3 \cdot \pi \cdot r_o^3$

$m = \rho \cdot V$

$Q_{\max} = m \cdot c_p \cdot (T_i - T_{\infty})$

$Q/Q_{\max} = 1 - 3 \cdot (T_o - T_{\infty}) / (T_i - T_{\infty}) \cdot (\sin(\lambda_1) - \lambda_1 \cdot \cos(\lambda_1)) / \lambda_1^3$

T_i [C]	T_o [C]	T_r [C]	Q [kJ]
2	-1.658	-5.369	6.861
4	-0.08803	-4.236	7.668
6	1.482	-3.103	8.476
8	3.051	-1.97	9.283
10	4.621	-0.8371	10.09
12	6.191	0.296	10.9
14	7.76	1.429	11.7
16	9.33	2.562	12.51
18	10.9	3.695	13.32
20	12.47	4.828	14.13
22	14.04	5.961	14.93
24	15.61	7.094	15.74
26	17.18	8.227	16.55
28	18.75	9.36	17.35
30	20.32	10.49	18.16



11-55 An orange is exposed to very cold ambient air. It is to be determined whether the orange will freeze in 4 h in subfreezing temperatures.

Assumptions **1** The orange is spherical in shape with a diameter of 8 cm. **2** Heat conduction in the orange is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the orange are constant, and are those of water. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the orange are approximated by those of water at the average temperature of about 5°C, $k = 0.571 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = k / \rho c_p = 0.571 / (999.9 \times 4205) = 0.136 \times 10^{-6} \text{ m}^2/\text{s}$ (Table A-15).

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(15 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04 \text{ m})}{(0.571 \text{ W/m} \cdot ^\circ\text{C})} = 1.051 \approx 1.0$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.5708 \quad \text{and} \quad A_1 = 1.2732$$

The Fourier number is

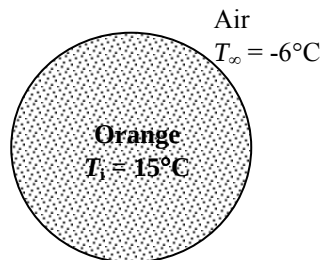
$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.136 \times 10^{-6} \text{ m}^2/\text{s})(4 \text{ h} \times 3600 \text{ s/h})}{(0.04 \text{ m})^2} = 1.224 > 0.2$$

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the temperature at the surface of the oranges becomes

$$\theta(r_o, t)_{sph} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.2732) e^{-(1.5708)^2 (1.224)} \frac{\sin(1.5708 \text{ rad})}{1.5708} = 0.0396$$

$$\frac{T(r_o, t) - (-6)}{15 - (-6)} = 0.0396 \longrightarrow T(r_o, t) = -5.2^\circ\text{C}$$

which is less than 0°C. Therefore, the oranges will freeze.



11-56 A hot baked potato is taken out of the oven and wrapped so that no heat is lost from it. The time the potato is baked in the oven and the final equilibrium temperature of the potato after it is wrapped are to be determined.

Assumptions 1 The potato is spherical in shape with a diameter of 9 cm. 2 Heat conduction in the potato is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the potato are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the potato are given to be $k = 0.6$ W/m·°C, $\rho = 1100$ kg/m³, $c_p = 3.9$ kJ/kg·°C, and $\alpha = 1.4 \times 10^{-7}$ m²/s.

Analysis (a) The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.045 \text{ m})}{(0.6 \text{ W/m} \cdot ^\circ\text{C})} = 3$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 2.2889 \quad \text{and} \quad A_1 = 1.6227$$

Then the Fourier number and the time period become

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{70 - 170}{25 - 170} = 0.69 = (1.6227)e^{-(2.2889)^2 \tau} \longrightarrow \tau = 0.163$$

which is not greater than 0.2 but it is close. We may use one-term approximation knowing that the result may be somewhat in error. Then the baking time of the potatoes is determined to be

$$t = \frac{\tau_o^2}{\alpha} = \frac{(0.163)(0.045 \text{ m})^2}{1.4 \times 10^{-7} \text{ m}^2/\text{s}} = 2358 \text{ s} = \mathbf{39.3 \text{ min}}$$

(b) The maximum amount of heat transfer is

$$m = \rho V = \rho \frac{4}{3} \pi r_o^3 = (1100 \text{ kg/m}^3) \left[\frac{4}{3} \pi (0.045 \text{ m})^3 \right] = 0.420 \text{ kg}$$

$$Q_{\max} = mc_p (T_\infty - T_i) = (0.420 \text{ kg})(3.900 \text{ kJ/kg} \cdot ^\circ\text{C})(170 - 25)^\circ\text{C} = 237 \text{ kJ}$$

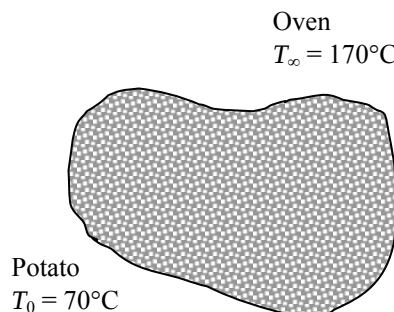
Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{0,sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3} = 1 - 3(0.69) \frac{\sin(2.2889) - (2.2889) \cos(2.2889)}{(2.2889)^3} = 0.610$$

$$Q = 0.610 Q_{\max} = (0.610)(237 \text{ kJ}) = \mathbf{145 \text{ kJ}}$$

The final equilibrium temperature of the potato after it is wrapped is

$$Q = mc_p (T_{eqv} - T_i) \longrightarrow T_{eqv} = T_i + \frac{Q}{mc_p} = 25^\circ\text{C} + \frac{145 \text{ kJ}}{(0.420 \text{ kg})(3.9 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{114^\circ\text{C}}$$



11-57 The center temperature of potatoes is to be lowered to 6°C during cooling. The cooling time and if any part of the potatoes will suffer chilling injury during this cooling process are to be determined.

Assumptions 1 The potatoes are spherical in shape with a radius of $r_o = 3$ cm. 2 Heat conduction in the potato is one-dimensional in the radial direction because of the symmetry about the midpoint. 3 The thermal properties of the potato are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of potatoes are given to be $k = 0.50$ W/m·°C and $\alpha = 0.13 \times 10^{-6}$ m²/s.

Analysis First we find the Biot number:

$$Bi = \frac{hr_o}{k} = \frac{(19 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03 \text{ m})}{0.5 \text{ W/m} \cdot ^\circ\text{C}} = 1.14$$

From Table 11-2 we read, for a sphere, $\lambda_1 = 1.635$ and $A_1 = 1.302$. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{6 - 2}{25 - 2} = 1.302 e^{-(1.635)^2 \tau} \rightarrow \tau = 0.753$$

which is greater than 0.2 and thus the one-term solution is applicable. Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \longrightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.753)(0.03 \text{ m})^2}{0.13 \times 10^{-6} \text{ m}^2/\text{s}} = 5213 \text{ s} = \mathbf{1.45 \text{ h}}$$

The lowest temperature during cooling will occur on the surface ($r/r_o = 1$), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_o)}{\lambda_1 r / r_o} \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = \frac{T_o - T_\infty}{T_i - T_\infty} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

Substituting,
$$\frac{T(r_o) - 2}{25 - 2} = \left(\frac{6 - 2}{25 - 2} \right) \frac{\sin(1.635 \text{ rad})}{1.635} \longrightarrow T(r_o) = 4.44^\circ\text{C}$$

which is above the temperature range of 3 to 4 °C for chilling injury for potatoes. Therefore, **no part** of the potatoes will experience chilling injury during this cooling process.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{k}{hr_o} = \frac{0.50 \text{ W/m} \cdot ^\circ\text{C}}{(19 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03 \text{ m})} = 0.877 \\ \frac{T_0 - T_\infty}{T_i - T_\infty} &= \frac{6 - 2}{25 - 2} = 0.174 \end{aligned} \right\} \tau = \frac{\alpha t}{r_o^2} = 0.75 \quad (\text{Fig. 11-17a})$$

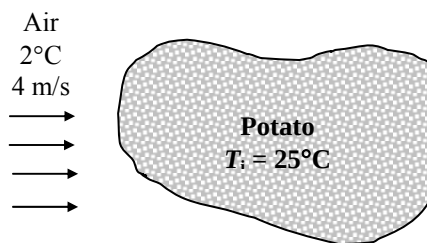
Therefore,
$$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.75)(0.03)^2}{0.13 \times 10^{-6} \text{ m}^2/\text{s}} = 5192 \text{ s} = \mathbf{1.44 \text{ h}}$$

The surface temperature is determined from

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{k}{hr_o} = 0.877 \\ \frac{r}{r_o} &= 1 \end{aligned} \right\} \frac{T(r) - T_\infty}{T_o - T_\infty} = 0.6 \quad (\text{Fig. 11-17b})$$

which gives $T_{\text{surface}} = T_\infty + 0.6(T_o - T_\infty) = 2 + 0.6(6 - 2) = 4.4^\circ\text{C}$

The slight difference between the two results is due to the reading error of the charts.



11-58E The center temperature of oranges is to be lowered to 40°F during cooling. The cooling time and if any part of the oranges will freeze during this cooling process are to be determined.

Assumptions 1 The oranges are spherical in shape with a radius of $r_o = 1.25 \text{ in} = 0.1042 \text{ ft}$. 2 Heat conduction in the orange is one-dimensional in the radial direction because of the symmetry about the midpoint. 3 The thermal properties of the orange are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of oranges are given to be $k = 0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ and $\alpha = 1.4 \times 10^{-6} \text{ ft}^2/\text{s}$.

Analysis First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(4.6 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(1.25/12 \text{ ft})}{0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}} = 1.843$$

From Table 11-2 we read, for a sphere, $\lambda_1 = 1.9569$ and $A_1 = 1.447$. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{40 - 25}{78 - 25} = 1.447 e^{-(1.9569)^2 \tau} \rightarrow \tau = 0.426$$

which is greater than 0.2 and thus the one-term solution is applicable.

Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.426)(1.25/12 \text{ ft})^2}{1.4 \times 10^{-6} \text{ ft}^2/\text{s}} = 3302 \text{ s} = \mathbf{55.0 \text{ min}}$$

The lowest temperature during cooling will occur on the surface ($r/r_o = 1$), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_o)}{\lambda_1 r / r_o} \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = \frac{T_o - T_\infty}{T_i - T_\infty} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

$$\text{Substituting, } \frac{T(r_o) - 25}{78 - 25} = \left(\frac{40 - 25}{78 - 25} \right) \frac{\sin(1.9569 \text{ rad})}{1.9569} \longrightarrow T(r_o) = 32.1^\circ\text{F}$$

which is above the freezing temperature of 31°F for oranges. Therefore, no part of the oranges will freeze during this cooling process.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left. \begin{aligned} \frac{1}{\text{Bi}} &= \frac{k}{hr_o} = \frac{0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{(4.6 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(1.25/12 \text{ ft})} = 0.543 \\ \frac{T_0 - T_\infty}{T_i - T_\infty} &= \frac{40 - 25}{78 - 25} = 0.283 \end{aligned} \right\} \tau = \frac{\alpha t}{r_o^2} = 0.43 \quad (\text{Fig. 11-17a})$$

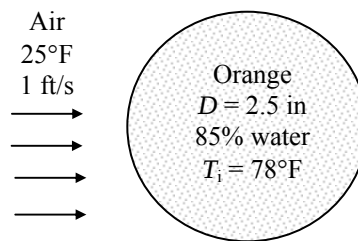
$$\text{Therefore, } t = \frac{\tau r_o^2}{\alpha} = \frac{(0.43)(1.25/12 \text{ ft})^2}{1.4 \times 10^{-6} \text{ ft}^2/\text{s}} = 3333 \text{ s} = 55.6 \text{ min}$$

The lowest temperature during cooling will occur on the surface ($r/r_o = 1$) of the oranges is determined to be

$$\left. \begin{aligned} \frac{1}{\text{Bi}} &= \frac{k}{hr_o} = 0.543 \\ \frac{r}{r_o} &= 1 \end{aligned} \right\} \frac{T(r) - T_\infty}{T_0 - T_\infty} = 0.45 \quad (\text{Fig. 11-17b})$$

$$\text{which gives } T_{\text{surface}} = T_\infty + 0.45(T_0 - T_\infty) = 25 + 0.45(40 - 25) = 31.8^\circ\text{F}$$

The slight difference between the two results is due to the reading error of the charts.



11-59 The center temperature of a beef carcass is to be lowered to 4°C during cooling. The cooling time and if any part of the carcass will suffer freezing injury during this cooling process are to be determined.

Assumptions 1 The beef carcass can be approximated as a cylinder with insulated top and base surfaces having a radius of $r_o = 12$ cm and a height of $H = 1.4$ m. 2 Heat conduction in the carcass is one-dimensional in the radial direction because of the symmetry about the centerline. 3 The thermal properties of the carcass are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of carcass are given to be $k = 0.47$ W/m·°C and $\alpha = 0.13 \times 10^{-6}$ m²/s.

Analysis First we find the Biot number:

$$Bi = \frac{hr_o}{k} = \frac{(22 \text{ W/m}^2 \cdot ^\circ\text{C})(0.12 \text{ m})}{0.47 \text{ W/m} \cdot ^\circ\text{C}} = 5.62$$

From Table 11-2 we read, for a cylinder, $\lambda_1 = 2.027$ and $A_1 = 1.517$. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{4 - (-10)}{37 - (-10)} = 1.517 e^{-(2.027)^2 \tau} \rightarrow \tau = 0.396$$

which is greater than 0.2 and thus the one-term solution is applicable.

Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\pi_o^2}{\alpha} = \frac{(0.396)(0.12 \text{ m})^2}{0.13 \times 10^{-6} \text{ m}^2/\text{s}} = 43,865 \text{ s} = \mathbf{12.2 \text{ h}}$$

The lowest temperature during cooling will occur on the surface ($r/r_o = 1$), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r / r_o) \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \theta_0 J_0(\lambda_1 r / r_o) = \frac{T_0 - T_\infty}{T_i - T_\infty} J_0(\lambda_1 r_o / r_o)$$

Substituting,
$$\frac{T(r_o) - (-10)}{37 - (-10)} = \left(\frac{4 - (-10)}{37 - (-10)} \right) J_0(\lambda_1) = 0.2979 \times 0.2084 = 0.0621 \longrightarrow T(r_o) = -7.1^\circ\text{C}$$

which is below the freezing temperature of -1.7°C . Therefore, the outer part of the beef carcass will freeze during this cooling process.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{k}{hr_o} = \frac{0.47 \text{ W/m} \cdot ^\circ\text{C}}{(22 \text{ W/m}^2 \cdot ^\circ\text{C})(0.12 \text{ m})} = 0.178 \\ \frac{T_0 - T_\infty}{T_i - T_\infty} &= \frac{4 - (-10)}{37 - (-10)} = 0.298 \end{aligned} \right\} \tau = \frac{\alpha t}{r_o^2} = 0.4 \quad (\text{Fig. 11-16a})$$

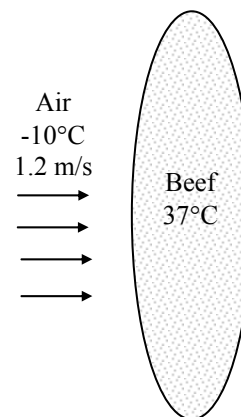
Therefore,
$$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.4)(0.12 \text{ m})^2}{0.13 \times 10^{-6} \text{ m}^2/\text{s}} = 44,308 \text{ s} \approx 12.3 \text{ h}$$

The surface temperature is determined from

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{k}{hr_o} = 0.178 \\ \frac{r}{r_o} &= 1 \end{aligned} \right\} \frac{T(r) - T_\infty}{T_0 - T_\infty} = 0.17 \quad (\text{Fig. 11-16b})$$

which gives $T_{\text{surface}} = T_\infty + 0.17(T_0 - T_\infty) = -10 + 0.17[4 - (-10)] = -7.6^\circ\text{C}$

The difference between the two results is due to the reading error of the charts.



11-60 The center temperature of meat slabs is to be lowered to -18°C during cooling. The cooling time and the surface temperature of the slabs at the end of the cooling process are to be determined.

Assumptions 1 The meat slabs can be approximated as very large plane walls of half-thickness $L = 11.5$ cm. 2 Heat conduction in the meat slabs is one-dimensional because of the symmetry about the centerplane. 3 The thermal properties of the meat slabs are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified). 6 The phase change effects are not considered, and thus the actual cooling time will be much longer than the value determined.

Properties The thermal conductivity and thermal diffusivity of meat slabs are given to be $k = 0.47 \text{ W/m}\cdot^{\circ}\text{C}$ and $\alpha = 0.13 \times 10^{-6} \text{ m}^2/\text{s}$. These properties will be used for both fresh and frozen meat.

Analysis First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(20 \text{ W/m}^2 \cdot ^{\circ}\text{C})(0.115 \text{ m})}{0.47 \text{ W/m}\cdot^{\circ}\text{C}} = 4.89$$

From Table 11-2 we read, for a plane wall, $\lambda_1 = 1.308$ and $A_1 = 1.239$. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_{\infty}}{T_i - T_{\infty}} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{-18 - (-30)}{7 - (-30)} = 1.239 e^{-(1.308)^2 \tau} \rightarrow \tau = 0.783$$

which is greater than 0.2 and thus the one-term solution is applicable. Then the cooling time becomes

$$\tau = \frac{\alpha t}{L^2} \rightarrow t = \frac{\tau L^2}{\alpha} = \frac{(0.783)(0.115 \text{ m})^2}{0.13 \times 10^{-6} \text{ m}^2/\text{s}} = 79,650 \text{ s} = \mathbf{22.1 \text{ h}}$$

The lowest temperature during cooling will occur on the surface ($x/L = 1$), and is determined to be

$$\frac{T(x) - T_{\infty}}{T_i - T_{\infty}} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 x / L) \rightarrow \frac{T(L) - T_{\infty}}{T_i - T_{\infty}} = \theta_0 \cos(\lambda_1 L / L) = \frac{T_o - T_{\infty}}{T_i - T_{\infty}} \cos(\lambda_1)$$

Substituting,

$$\frac{T(L) - (-30)}{7 - (-30)} = \left(\frac{-18 - (-30)}{7 - (-30)} \right) \cos(\lambda_1) = 0.3243 \times 0.2598 = 0.08425 \rightarrow T(L) = \mathbf{-26.9^{\circ}\text{C}}$$

which is close the temperature of the refrigerated air.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left. \begin{aligned} \frac{1}{\text{Bi}} &= \frac{k}{hL} = \frac{0.47 \text{ W/m}\cdot^{\circ}\text{C}}{(20 \text{ W/m}^2 \cdot ^{\circ}\text{C})(0.115 \text{ m})} = 0.204 \\ \frac{T_o - T_{\infty}}{T_i - T_{\infty}} &= \frac{-18 - (-30)}{7 - (-30)} = 0.324 \end{aligned} \right\} \tau = \frac{\alpha t}{L^2} = 0.75 \quad (\text{Fig. 11-15a})$$

$$\text{Therefore, } t = \frac{\tau L^2}{\alpha} = \frac{(0.75)(0.115 \text{ m})^2}{0.13 \times 10^{-6} \text{ m}^2/\text{s}} = 76,300 \text{ s} \approx 21.2 \text{ h}$$

The surface temperature is determined from

$$\left. \begin{aligned} \frac{1}{\text{Bi}} &= \frac{k}{hL} = 0.204 \\ \frac{x}{L} &= 1 \end{aligned} \right\} \frac{T(x) - T_{\infty}}{T_o - T_{\infty}} = 0.22 \quad (\text{Fig. 11-15b})$$

$$\text{which gives } T_{\text{surface}} = T_{\infty} + 0.22(T_o - T_{\infty}) = -30 + 0.22[-18 - (-30)] = \mathbf{-27.4^{\circ}\text{C}}$$

The slight difference between the two results is due to the reading error of the charts.

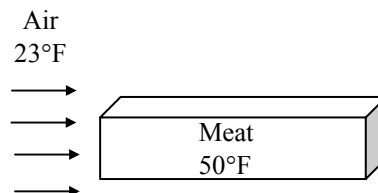
11-61E The center temperature of meat slabs is to be lowered to 36°F during 12-h of cooling. The average heat transfer coefficient during this cooling process is to be determined.

Assumptions **1** The meat slabs can be approximated as very large plane walls of half-thickness $L = 3\text{-in.}$ **2** Heat conduction in the meat slabs is one-dimensional because of symmetry about the centerplane. **3** The thermal properties of the meat slabs are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of meat slabs are given to be $k = 0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ and $\alpha = 1.4 \times 10^{-6} \text{ ft}^2/\text{s}$.

Analysis The average heat transfer coefficient during this cooling process is determined from the transient temperature charts for a flat plate as follows:

$$\left. \begin{aligned} \tau = \frac{\alpha t}{L^2} &= \frac{(1.4 \times 10^{-6} \text{ ft}^2/\text{s})(12 \times 3600 \text{ s})}{(3/12 \text{ ft})^2} = 0.968 \\ \frac{T_0 - T_\infty}{T_i - T_\infty} &= \frac{36 - 23}{50 - 23} = 0.481 \end{aligned} \right\} \frac{1}{Bi} = 0.7 \quad (\text{Fig. 11-15a})$$



Therefore,

$$h = \frac{kBi}{L} = \frac{(0.26 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1/0.7)}{(3/12) \text{ ft}} = \mathbf{1.5 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}}$$

Discussion We could avoid the uncertainty associated with the reading of the charts and obtain a more accurate result by using the one-term solution relation for an infinite plane wall, but it would require a trial and error approach since the Bi number is not known.

11-62 Chickens are to be chilled by holding them in agitated brine for 2.75 h. The center and surface temperatures of the chickens are to be determined, and if any part of the chickens will freeze during this cooling process is to be assessed.

Assumptions 1 The chickens are spherical in shape. 2 Heat conduction in the chickens is one-dimensional in the radial direction because of symmetry about the midpoint. 3 The thermal properties of the chickens are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified). 6 The phase change effects are not considered, and thus the actual the temperatures will be much higher than the values determined since a considerable part of the cooling process will occur during phase change (freezing of chicken).

Properties The thermal conductivity, thermal diffusivity, and density of chickens are given to be $k = 0.45 \text{ W/m}\cdot^\circ\text{C}$, $\alpha = 0.13 \times 10^{-6} \text{ m}^2/\text{s}$, and $\rho = 950 \text{ kg/m}^3$. These properties will be used for both fresh and frozen chicken.

Analysis We first find the volume and equivalent radius of the chickens:

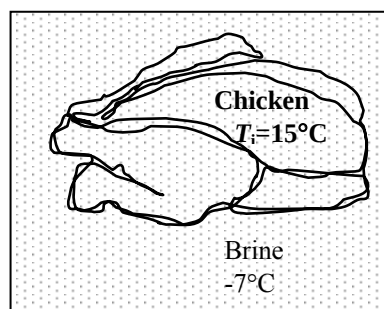
$$V = m / \rho = 1700 \text{ g} / (0.95 \text{ g/cm}^3) = 1789 \text{ cm}^3$$

$$r_o = \left(\frac{3}{4\pi} V \right)^{1/3} = \left(\frac{3}{4\pi} 1789 \text{ cm}^3 \right)^{1/3} = 7.53 \text{ cm} = 0.0753 \text{ m}$$

Then the Biot and Fourier numbers become

$$\text{Bi} = \frac{hr_o}{k} = \frac{(440 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0753 \text{ m})}{0.45 \text{ W/m}\cdot^\circ\text{C}} = 73.6$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.13 \times 10^{-6} \text{ m}^2/\text{s})(2.75 \times 3600 \text{ s})}{(0.0753 \text{ m})^2} = 0.2270$$



Note that $\tau = 0.2270 > 0.2$, and thus the one-term solution is applicable. From Table 11-2 we read, for a sphere, $\lambda_1 = 3.094$ and $A_1 = 1.998$. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{T_0 - (-7)}{15 - (-7)} = 1.998 e^{-(3.094)^2 (0.2270)} = 0.2274 \rightarrow T_0 = -2.0^\circ\text{C}$$

The lowest temperature during cooling will occur on the surface ($r/r_o = 1$), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_o)}{\lambda_1 r / r_o} \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = \frac{T_0 - T_\infty}{T_i - T_\infty} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

$$\text{Substituting, } \frac{T(r_o) - (-7)}{15 - (-7)} = 0.2274 \frac{\sin(3.094 \text{ rad})}{3.094} \rightarrow T(r_o) = -6.9^\circ\text{C}$$

Most parts of chicken will freeze during this process since the freezing point of chicken is -2.8°C .

Discussion We could also solve this problem using transient temperature charts, but the data in this case falls at a point on the chart which is very difficult to read:

$$\left. \begin{aligned} \tau &= \frac{\alpha t}{r_o^2} = \frac{(0.13 \times 10^{-6} \text{ m}^2/\text{s})(2.75 \times 3600 \text{ s})}{(0.0753 \text{ m})^2} = 0.227 \\ \frac{1}{\text{Bi}} &= \frac{k}{hr_o} = \frac{0.45 \text{ W/m}\cdot^\circ\text{C}}{(440 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0753 \text{ m})} = 0.0136 \end{aligned} \right\} \frac{T_0 - T_\infty}{T_i - T_\infty} = 0.15 \dots 0.30 \text{ ?? (Fig. 11-17)}$$

Transient Heat Conduction in Semi-Infinite Solids

11-63C A semi-infinite medium is an idealized body which has a single exposed plane surface and extends to infinity in all directions. The earth and thick walls can be considered to be semi-infinite media.

11-64C A thick plane wall can be treated as a semi-infinite medium if all we are interested in is the variation of temperature in a region near one of the surfaces for a time period during which the temperature in the mid section of the wall does not experience any change.

11-65C The total amount of heat transfer from a semi-infinite solid up to a specified time t_0 can be determined by integration from

$$Q = \int_0^{t_0} Ah[T(0, t) - T_\infty] dt$$

where the surface temperature $T(0, t)$ is obtained from Eq. 11-47 by substituting $x = 0$.

11-66 The water pipes are buried in the ground to prevent freezing. The minimum burial depth at a particular location is to be determined.

Assumptions 1 The temperature in the soil is affected by the thermal conditions at one surface only, and thus the soil can be considered to be a semi-infinite medium with a specified surface temperature. **2** The thermal properties of the soil are constant.

Properties The thermal properties of the soil are given to be $k = 0.35 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.15 \times 10^{-6} \text{ m}^2/\text{s}$.

Analysis The length of time the snow pack stays on the ground is

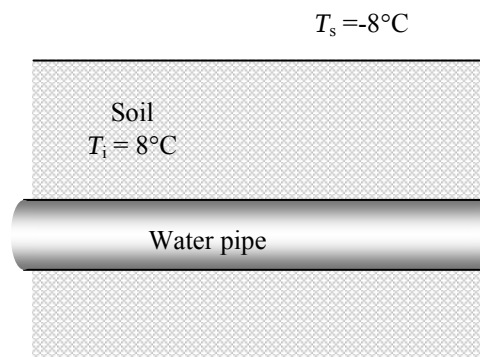
$$t = (60 \text{ days})(24 \text{ hr/day})(3600 \text{ s/hr}) = 5.184 \times 10^6 \text{ s}$$

The surface is kept at -8°C at all times. The depth at which freezing at 0°C occurs can be determined from the analytical solution,

$$\begin{aligned} \frac{T(x, t) - T_i}{T_s - T_i} &= \text{erfc}\left(\frac{x}{\sqrt{\alpha t}}\right) \\ \frac{0 - 8}{-8 - 8} &= \text{erfc}\left(\frac{x}{2\sqrt{(0.15 \times 10^{-6} \text{ m}^2/\text{s})(5.184 \times 10^6 \text{ s})}}\right) \\ 0.5 &= \text{erfc}\left(\frac{x}{1.7636}\right) \end{aligned}$$

Then from Table 11-4 we get $\frac{x}{1.7636} = 0.4796 \longrightarrow x = \mathbf{0.846 \text{ m}}$

Discussion The solution could also be determined using the chart, but it would be subject to reading error.



11-67 An area is subjected to cold air for a 10-h period. The soil temperatures at distances 0, 10, 20, and 50 cm from the earth's surface are to be determined.

Assumptions 1 The temperature in the soil is affected by the thermal conditions at one surface only, and thus the soil can be considered to be a semi-infinite medium with a specified surface temperature. **2** The thermal properties of the soil are constant.

Properties The thermal properties of the soil are given to be $k = 0.9 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.6 \times 10^{-5} \text{ m}^2/\text{s}$.

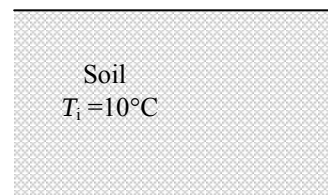
Analysis The one-dimensional transient temperature distribution in the ground can be determined from

$$\frac{T(x, t) - T_i}{T_\infty - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right) - \exp\left(\frac{hx}{k} + \frac{h^2 \alpha t}{k^2}\right) \left[\text{erfc}\left(\frac{x}{2\sqrt{\alpha t}} + \frac{h\sqrt{\alpha t}}{k}\right) \right] \quad \begin{array}{l} \longrightarrow \text{Winds} \\ \longrightarrow \\ \longrightarrow T_\infty = -10^\circ\text{C} \end{array}$$

where

$$\frac{h\sqrt{\alpha t}}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})\sqrt{(1.6 \times 10^{-5} \text{ m}^2/\text{s})(10 \times 3600 \text{ s})}}{0.9 \text{ W/m} \cdot ^\circ\text{C}} = 33.7$$

$$\frac{h^2 \alpha t}{k^2} = \left(\frac{h\sqrt{\alpha t}}{k}\right)^2 = 33.7^2 = 1138$$



Then we conclude that the last term in the temperature distribution relation above must be zero regardless of x despite the exponential term tending to infinity since (1) $\text{erfc}(\eta) \rightarrow 0$ for $\eta > 4$ (see Table 11-4) and (2) the term has to remain less than 1 to have physically meaningful solutions. That is,

$$\exp\left(\frac{hx}{k} + \frac{h^2 \alpha t}{k^2}\right) \left[\text{erfc}\left(\frac{x}{2\sqrt{\alpha t}} + \frac{h\sqrt{\alpha t}}{k}\right) \right] = \exp\left(\frac{hx}{k} + 1138\right) \left[\text{erfc}\left(\frac{x}{2\sqrt{\alpha t}} + 33.7\right) \right] \cong 0$$

Therefore, the temperature distribution relation simplifies to

$$\frac{T(x, t) - T_i}{T_\infty - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right) \rightarrow T(x, t) = T_i + (T_\infty - T_i) \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become

$$x = 0: \quad T(0, 10 \text{ h}) = T_i + (T_\infty - T_i) \text{erfc}\left(\frac{0}{2\sqrt{\alpha t}}\right) = T_i + (T_\infty - T_i) \text{erfc}(0) = T_i + (T_\infty - T_i) \times 1 = T_\infty = -10^\circ\text{C}$$

$$\begin{aligned} x = 0.1 \text{ m}: \quad T(0.1 \text{ m}, 10 \text{ h}) &= 10 + (-10 - 10) \text{erfc}\left(\frac{0.1 \text{ m}}{2\sqrt{(1.6 \times 10^{-5} \text{ m}^2/\text{s})(10 \text{ h} \times 3600 \text{ s/h})}}\right) \\ &= 10 - 20 \text{erfc}(0.066) = 10 - 20 \times 0.9257 = -8.5^\circ\text{C} \end{aligned}$$

$$\begin{aligned} x = 0.2 \text{ m}: \quad T(0.2 \text{ m}, 10 \text{ h}) &= 10 + (-10 - 10) \text{erfc}\left(\frac{0.2 \text{ m}}{2\sqrt{(1.6 \times 10^{-5} \text{ m}^2/\text{s})(10 \text{ h} \times 3600 \text{ s/h})}}\right) \\ &= 10 - 20 \text{erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^\circ\text{C} \end{aligned}$$

$$\begin{aligned} x = 0.5 \text{ m}: \quad T(0.5 \text{ m}, 10 \text{ h}) &= 10 + (-10 - 10) \text{erfc}\left(\frac{0.5 \text{ m}}{2\sqrt{(1.6 \times 10^{-5} \text{ m}^2/\text{s})(10 \text{ h} \times 3600 \text{ s/h})}}\right) \\ &= 10 - 20 \text{erfc}(0.329) = 10 - 20 \times 0.6418 = -2.8^\circ\text{C} \end{aligned}$$

11-68 EES Prob. 11-67 is reconsidered. The soil temperature as a function of the distance from the earth's surface is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$T_i = 10$ [C]
 $T_{\infty} = -10$ [C]
 $h = 40$ [W/m²-C]
 $\text{time} = 10 \times 3600$ [s]
 $x = 0.1$ [m]

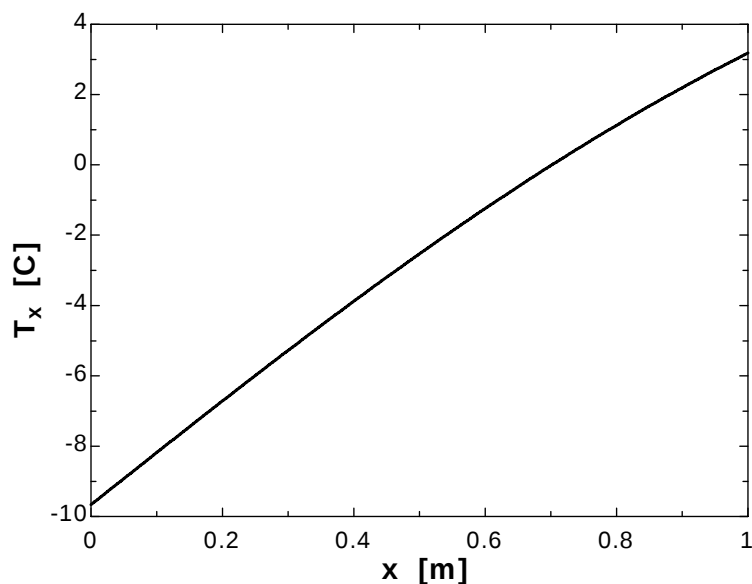
"PROPERTIES"

$k = 0.9$ [W/m-C]
 $\alpha = 1.6 \times 10^{-5}$ [m²/s]

"ANALYSIS"

$$\frac{(T_x - T_i)}{(T_{\infty} - T_i)} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha \cdot \text{time}}}\right) - \frac{\exp\left(\frac{h \cdot x}{k} + \frac{h^2 \cdot \alpha \cdot \text{time}}{k^2}\right) \cdot \text{erfc}\left(\frac{x}{2\sqrt{\alpha \cdot \text{time}}} + \frac{h \cdot \sqrt{\alpha \cdot \text{time}}}{k}\right)}{k}$$

x [m]	T _x [C]
0	-9.666
0.05	-8.923
0.1	-8.183
0.15	-7.447
0.2	-6.716
0.25	-5.993
0.3	-5.277
0.35	-4.572
0.4	-3.878
0.45	-3.197
0.5	-2.529
0.55	-1.877
0.6	-1.24
0.65	-0.6207
0.7	-0.01894
0.75	0.5643
0.8	1.128
0.85	1.672
0.9	2.196
0.95	2.7
1	3.183



11-69 An aluminum block is subjected to heat flux. The surface temperature of the block is to be determined.

Assumptions **1** All heat flux is absorbed by the block. **2** Heat loss from the block is disregarded (and thus the result obtained is the maximum temperature). **3** The block is sufficiently thick to be treated as a semi-infinite solid, and the properties of the block are constant.

Properties Thermal conductivity and diffusivity of aluminum at room temperature are $k = 237 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 97.1 \times 10^{-6} \text{ m}^2/\text{s}$.

Analysis This is a transient conduction problem in a semi-infinite medium subjected to constant surface heat flux, and the surface temperature can be determined to be

$$T_s = T_i + \frac{\dot{q}_s}{k} \sqrt{\frac{4\alpha t}{\pi}} = 20^\circ\text{C} + \frac{4000 \text{ W/m}^2}{237 \text{ W/m} \cdot ^\circ\text{C}} \sqrt{\frac{4(9.71 \times 10^{-5} \text{ m}^2/\text{s})(30 \times 60 \text{ s})}{\pi}} = 28.0^\circ\text{C}$$

Then the temperature rise of the surface becomes

$$\Delta T_s = 28 - 20 = \mathbf{8.0^\circ\text{C}}$$

11-70 The contact surface temperatures when a bare footed person steps on aluminum and wood blocks are to be determined.

Assumptions **1** Both bodies can be treated as the semi-infinite solids. **2** Heat loss from the solids is disregarded. **3** The properties of the solids are constant.

Properties The $\sqrt{k\rho c_p}$ value is $24 \text{ kJ/m}^2 \cdot ^\circ\text{C}$ for aluminum, $0.38 \text{ kJ/m}^2 \cdot ^\circ\text{C}$ for wood, and $1.1 \text{ kJ/m}^2 \cdot ^\circ\text{C}$ for the human flesh.

Analysis The surface temperature is determined from Eq. 11-49 to be

$$T_s = \frac{\sqrt{(k\rho c_p)_{\text{human}}} T_{\text{human}} + \sqrt{(k\rho c_p)_{\text{Al}}} T_{\text{Al}}}{\sqrt{(k\rho c_p)_{\text{human}}} + \sqrt{(k\rho c_p)_{\text{Al}}}} = \frac{(1.1 \text{ kJ/m}^2 \cdot ^\circ\text{C})(32^\circ\text{C}) + (24 \text{ kJ/m}^2 \cdot ^\circ\text{C})(20^\circ\text{C})}{(1.1 \text{ kJ/m}^2 \cdot ^\circ\text{C}) + (24 \text{ kJ/m}^2 \cdot ^\circ\text{C})} = \mathbf{20.5^\circ\text{C}}$$

In the case of wood block, we obtain

$$\begin{aligned} T_s &= \frac{\sqrt{(k\rho c_p)_{\text{human}}} T_{\text{human}} + \sqrt{(k\rho c_p)_{\text{wood}}} T_{\text{wood}}}{\sqrt{(k\rho c_p)_{\text{human}}} + \sqrt{(k\rho c_p)_{\text{wood}}}} \\ &= \frac{(1.1 \text{ kJ/m}^2 \cdot ^\circ\text{C})(32^\circ\text{C}) + (0.38 \text{ kJ/m}^2 \cdot ^\circ\text{C})(20^\circ\text{C})}{(1.1 \text{ kJ/m}^2 \cdot ^\circ\text{C}) + (0.38 \text{ kJ/m}^2 \cdot ^\circ\text{C})} \\ &= \mathbf{28.9^\circ\text{C}} \end{aligned}$$

11-71E The walls of a furnace made of concrete are exposed to hot gases at the inner surfaces. The time it will take for the temperature of the outer surface of the furnace to change is to be determined.

Assumptions 1 The temperature in the wall is affected by the thermal conditions at inner surfaces only and the convection heat transfer coefficient inside is given to be very large. Therefore, the wall can be considered to be a semi-infinite medium with a specified surface temperature of 1800°F. **2** The thermal properties of the concrete wall are constant.

Properties The thermal properties of the concrete are given to be $k = 0.64 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ and $\alpha = 0.023 \text{ ft}^2/\text{h}$.

Analysis The one-dimensional transient temperature distribution in the wall for that time period can be determined from

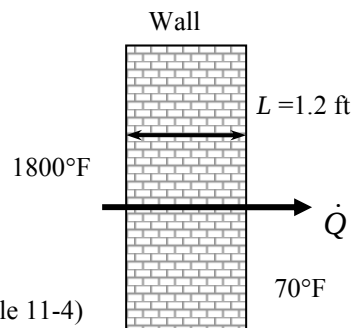
$$\frac{T(x, t) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

But,

$$\frac{T(x, t) - T_i}{T_s - T_i} = \frac{70.1 - 70}{1800 - 70} = 0.00006 \rightarrow 0.00006 = \text{erfc}(2.85) \quad (\text{Table 11-4})$$

Therefore,

$$\frac{x}{2\sqrt{\alpha t}} = 2.85 \rightarrow t = \frac{x^2}{4 \times (2.85)^2 \alpha} = \frac{(1.2 \text{ ft})^2}{4 \times (2.85)^2 (0.023 \text{ ft}^2/\text{h})} = 1.93 \text{ h} = \mathbf{116 \text{ min}}$$



11-72 A thick wood slab is exposed to hot gases for a period of 5 minutes. It is to be determined whether the wood will ignite.

Assumptions 1 The wood slab is treated as a semi-infinite medium subjected to convection at the exposed surface. **2** The thermal properties of the wood slab are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface.

Properties The thermal properties of the wood are $k = 0.17 \text{ W/m}\cdot^\circ\text{C}$ and $\alpha = 1.28 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis The one-dimensional transient temperature distribution in the wood can be determined from

$$\frac{T(x, t) - T_i}{T_\infty - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right) - \exp\left(\frac{hx}{k} + \frac{h^2 \alpha t}{k^2}\right) \left[\text{erfc}\left(\frac{x}{2\sqrt{\alpha t}} + \frac{h\sqrt{\alpha t}}{k}\right) \right]$$

where

$$\frac{h\sqrt{\alpha t}}{k} = \frac{(35 \text{ W/m}^2 \cdot ^\circ\text{C}) \sqrt{(1.28 \times 10^{-7} \text{ m}^2/\text{s})(5 \times 60 \text{ s})}}{0.17 \text{ W/m}\cdot^\circ\text{C}} = 1.276$$

$$\frac{h^2 \alpha t}{k^2} = \left(\frac{h\sqrt{\alpha t}}{k}\right)^2 = 1.276^2 = 1.628$$

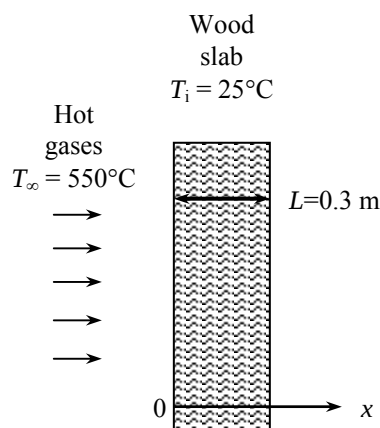
Noting that $x = 0$ at the surface and using Table 11-4 for erfc values,

$$\begin{aligned} \frac{T(x, t) - 25}{550 - 25} &= \text{erfc}(0) - \exp(0 + 1.628) \text{erfc}(0 + 1.276) \\ &= 1 - (5.0937)(0.0712) \\ &= 0.637 \end{aligned}$$

Solving for $T(x, t)$ gives

$$T(x, t) = \mathbf{360^\circ\text{C}}$$

which is less than the ignition temperature of 450°C. Therefore, the wood will not ignite.



11-73 The outer surfaces of a large cast iron container filled with ice are exposed to hot water. The time before the ice starts melting and the rate of heat transfer to the ice are to be determined.

Assumptions 1 The temperature in the container walls is affected by the thermal conditions at outer surfaces only and the convection heat transfer coefficient outside is given to be very large. Therefore, the wall can be considered to be a semi-infinite medium with a specified surface temperature. **2** The thermal properties of the wall are constant.

Properties The thermal properties of the cast iron are given to be $k = 52 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.70 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis The one-dimensional transient temperature distribution in the wall for that time period can be determined from

$$\frac{T(x, t) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

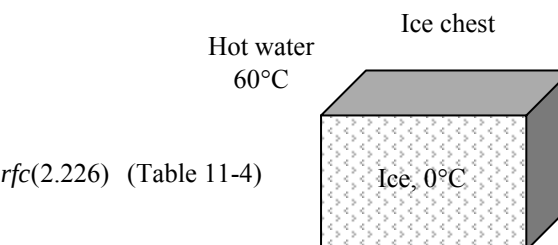
But,

$$\frac{T(x, t) - T_i}{T_s - T_i} = \frac{0.1 - 0}{60 - 0} = 0.00167 \rightarrow 0.00167 = \text{erfc}(2.226) \quad (\text{Table 11-4})$$

Therefore,

$$\frac{x}{2\sqrt{\alpha t}} = 2.226 \rightarrow t = \frac{x^2}{4 \times (2.226)^2 \alpha} = \frac{(0.05 \text{ m})^2}{4(2.226)^2 (1.7 \times 10^{-5} \text{ m}^2/\text{s})} = 7.4 \text{ s}$$

The rate of heat transfer to the ice when steady operation conditions are reached can be determined by applying the thermal resistance network concept as



$$R_{conv,i} = \frac{1}{h_i A} = \frac{1}{(250 \text{ W/m}^2 \cdot ^\circ\text{C})(1.2 \times 2 \text{ m}^2)} = 0.00167^\circ\text{C/W}$$

$$R_{wall} = \frac{L}{kA} = \frac{0.05 \text{ m}}{(52 \text{ W/m} \cdot ^\circ\text{C})(1.2 \times 2 \text{ m}^2)} = 0.00040^\circ\text{C/W}$$

$$R_{conv,o} = \frac{1}{h_o A} = \frac{1}{(\infty)(1.2 \times 2 \text{ m}^2)} \cong 0^\circ\text{C/W}$$

$$R_{total} = R_{conv,i} + R_{wall} + R_{conv,o} = 0.00167 + 0.00040 + 0 = 0.00207^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_2 - T_1}{R_{total}} = \frac{(60 - 0)^\circ\text{C}}{0.00207^\circ\text{C/W}} = 28,990 \text{ W}$$

Transient Heat Conduction in Multidimensional Systems

11-74C The product solution enables us to determine the dimensionless temperature of two- or three-dimensional heat transfer problems as the product of dimensionless temperatures of one-dimensional heat transfer problems. The dimensionless temperature for a two-dimensional problem is determined by determining the dimensionless temperatures in both directions, and taking their product.

11-75C The dimensionless temperature for a three-dimensional heat transfer is determined by determining the dimensionless temperatures of one-dimensional geometries whose intersection is the three dimensional geometry, and taking their product.

11-76C This short cylinder is physically formed by the intersection of a long cylinder and a plane wall. The dimensionless temperatures at the center of plane wall and at the center of the cylinder are determined first. Their product yields the dimensionless temperature at the center of the short cylinder.

11-77C The heat transfer in this short cylinder is one-dimensional since there is no heat transfer in the axial direction. The temperature will vary in the radial direction only.

11-78 A short cylinder is allowed to cool in atmospheric air. The temperatures at the centers of the cylinder and the top surface as well as the total heat transfer from the cylinder for 15 min of cooling are to be determined.

Assumptions 1 Heat conduction in the short cylinder is two-dimensional, and thus the temperature varies in both the axial x - and the radial r - directions. 2 The thermal properties of the cylinder are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of brass are given to be $\rho = 8530 \text{ kg/m}^3$, $c_p = 0.389 \text{ kJ/kg} \cdot ^\circ\text{C}$, $k = 110 \text{ W/m} \cdot ^\circ\text{C}$, and $\alpha = 3.39 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis This short cylinder can physically be formed by the intersection of a long cylinder of radius $D/2 = 4 \text{ cm}$ and a plane wall of thickness $2L = 15 \text{ cm}$. We measure x from the midplane.

(a) The Biot number is calculated for the plane wall to be

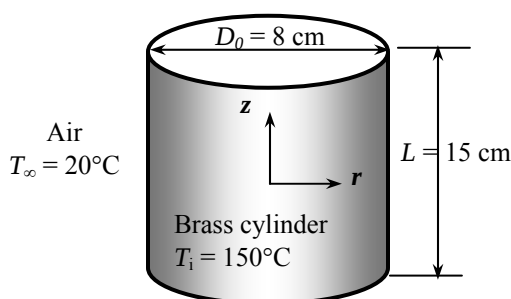
$$Bi = \frac{hL}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.075 \text{ m})}{(110 \text{ W/m} \cdot ^\circ\text{C})} = 0.02727$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.1620 \quad \text{and} \quad A_1 = 1.0045$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(3.39 \times 10^{-5} \text{ m}^2/\text{s})(15 \text{ min} \times 60 \text{ s/min})}{(0.075 \text{ m})^2} = 5.424 > 0.2$$



Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the dimensionless temperature at the center of the plane wall is determined from

$$\theta_{0,wall} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.0045) e^{-(0.1620)^2 (5.424)} = 0.871$$

We repeat the same calculations for the long cylinder,

$$Bi = \frac{hr_o}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04 \text{ m})}{(110 \text{ W/m} \cdot ^\circ\text{C})} = 0.01455$$

$$\lambda_1 = 0.1677 \quad \text{and} \quad A_1 = 1.0036$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(3.39 \times 10^{-5} \text{ m}^2/\text{s})(15 \times 60 \text{ s})}{(0.04 \text{ m})^2} = 19.069 > 0.2$$

$$\theta_{o,cyl} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.0036) e^{-(0.1677)^2 (19.069)} = 0.587$$

Then the center temperature of the short cylinder becomes

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{0,wall} \times \theta_{o,cyl} = 0.871 \times 0.587 = 0.511$$

$$\frac{T(0,0,t) - 20}{150 - 20} = 0.511 \longrightarrow T(0,0,t) = 86.4^\circ\text{C}$$

(b) The center of the top surface of the cylinder is still at the center of the long cylinder ($r = 0$), but at the outer surface of the plane wall ($x = L$). Therefore, we first need to determine the dimensionless temperature at the surface of the wall.

$$\theta(L, t)_{wall} = \frac{T(x, t) - T_{\infty}}{T_i - T_{\infty}} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L / L) = (1.0045) e^{-(0.1620)^2 (5.424)} \cos(0.1620) = 0.860$$

Then the center temperature of the top surface of the cylinder becomes

$$\left[\frac{T(L, 0, t) - T_{\infty}}{T_i - T_{\infty}} \right]_{short\ cylinder} = \theta(L, t)_{wall} \times \theta_{o, cyl} = 0.860 \times 0.587 = 0.505$$

$$\frac{T(L, 0, t) - 20}{150 - 20} = 0.505 \longrightarrow T(L, 0, t) = \mathbf{85.6^{\circ}C}$$

(c) We first need to determine the maximum heat can be transferred from the cylinder

$$m = \rho V = \rho \pi r_o^2 L = (8530 \text{ kg/m}^3) [\pi (0.04 \text{ m})^2 (0.15 \text{ m})] = 6.43 \text{ kg}$$

$$Q_{max} = mc_p (T_i - T_{\infty}) = (6.43 \text{ kg})(0.389 \text{ kJ/kg} \cdot ^{\circ}\text{C})(150 - 20)^{\circ}\text{C} = 325 \text{ kJ}$$

Then we determine the dimensionless heat transfer ratios for both geometries as

$$\left(\frac{Q}{Q_{max}} \right)_{wall} = 1 - \theta_{o, wall} \frac{\sin(\lambda_1)}{\lambda_1} = 1 - (0.871) \frac{\sin(0.1620)}{0.1620} = 0.133$$

$$\left(\frac{Q}{Q_{max}} \right)_{cyl} = 1 - 2\theta_{o, cyl} \frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2(0.587) \frac{0.0835}{0.1677} = 0.415$$

The heat transfer ratio for the short cylinder is

$$\left(\frac{Q}{Q_{max}} \right)_{short\ cylinder} = \left(\frac{Q}{Q_{max}} \right)_{plane\ wall} + \left(\frac{Q}{Q_{max}} \right)_{long\ cylinder} \left[1 - \left(\frac{Q}{Q_{max}} \right)_{plane\ wall} \right] = 0.133 + (0.415)(1 - 0.133) = 0.493$$

Then the total heat transfer from the short cylinder during the first 15 minutes of cooling becomes

$$Q = 0.493 Q_{max} = (0.493)(325 \text{ kJ}) = \mathbf{160 \text{ kJ}}$$

11-79 EES Prob. 11-78 is reconsidered. The effect of the cooling time on the center temperature of the cylinder, the center temperature of the top surface of the cylinder, and the total heat transfer is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D=0.08 [m]
 $r_o = D/2$
 height=0.15 [m]
 $L = \text{height}/2$
 $T_i = 150$ [C]
 $T_{\text{infinity}} = 20$ [C]
 $h = 40$ [W/m^2-C]
 time=15 [min]

"PROPERTIES"

$k = 110$ [W/m-C]
 $\rho = 8530$ [kg/m^3]
 $c_p = 0.389$ [kJ/kg-C]
 $\alpha = 3.39E-5$ [m^2/s]

"ANALYSIS"

"(a)"

"This short cylinder can physically be formed by the intersection of a long cylinder of radius r_o and a plane wall of thickness $2L$ "

"For plane wall"

$Bi_w = (h \cdot L) / k$

"From Table 11-2 corresponding to this Bi number, we read"

$\lambda_{1w} = 0.1620$ "w stands for wall"

$A_{1w} = 1.0045$

$\tau_w = (\alpha \cdot \text{time} \cdot \text{Convert}(\text{min}, \text{s})) / L^2$

$\theta_{ow} = A_{1w} \cdot \exp(-\lambda_{1w}^2 \cdot \tau_w)$ " $\theta_{ow} = (T_{ow} - T_{\text{infinity}}) / (T_i - T_{\text{infinity}})$ "

"For long cylinder"

$Bi_c = (h \cdot r_o) / k$ "c stands for cylinder"

"From Table 11-2 corresponding to this Bi number, we read"

$\lambda_{1c} = 0.1677$

$A_{1c} = 1.0036$

$\tau_c = (\alpha \cdot \text{time} \cdot \text{Convert}(\text{min}, \text{s})) / r_o^2$

$\theta_{oc} = A_{1c} \cdot \exp(-\lambda_{1c}^2 \cdot \tau_c)$ " $\theta_{oc} = (T_{oc} - T_{\text{infinity}}) / (T_i - T_{\text{infinity}})$ "

$(T_{oo} - T_{\text{infinity}}) / (T_i - T_{\text{infinity}}) = \theta_{ow} \cdot \theta_{oc}$ "center temperature of short cylinder"

"(b)"

$\theta_{Lw} = A_{1w} \cdot \exp(-\lambda_{1w}^2 \cdot \tau_w) \cdot \cos(\lambda_{1w} \cdot L / L)$ " $\theta_{Lw} = (T_{Lw} - T_{\text{infinity}}) / (T_i - T_{\text{infinity}})$ "

$(T_{Lo} - T_{\text{infinity}}) / (T_i - T_{\text{infinity}}) = \theta_{Lw} \cdot \theta_{oc}$ "center temperature of the top surface"

"(c)"

$V = \pi \cdot r_o^2 \cdot (2 \cdot L)$

$m = \rho \cdot V$

$Q_{\text{max}} = m \cdot c_p \cdot (T_i - T_{\text{infinity}})$

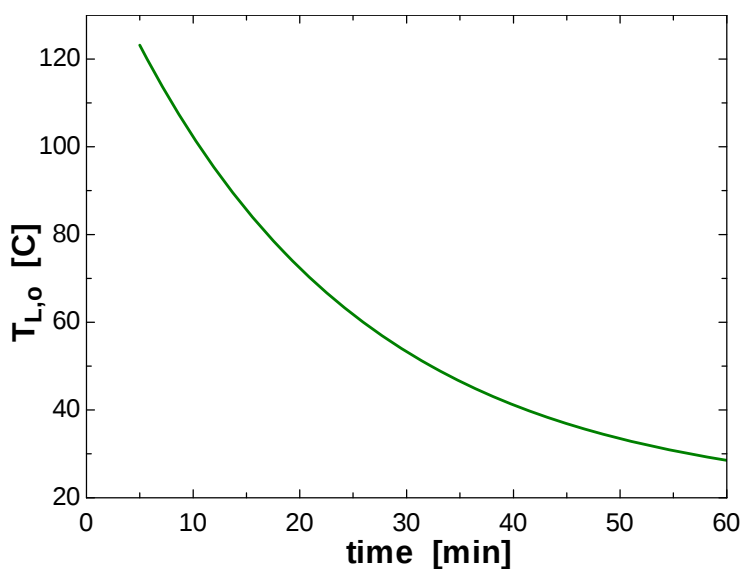
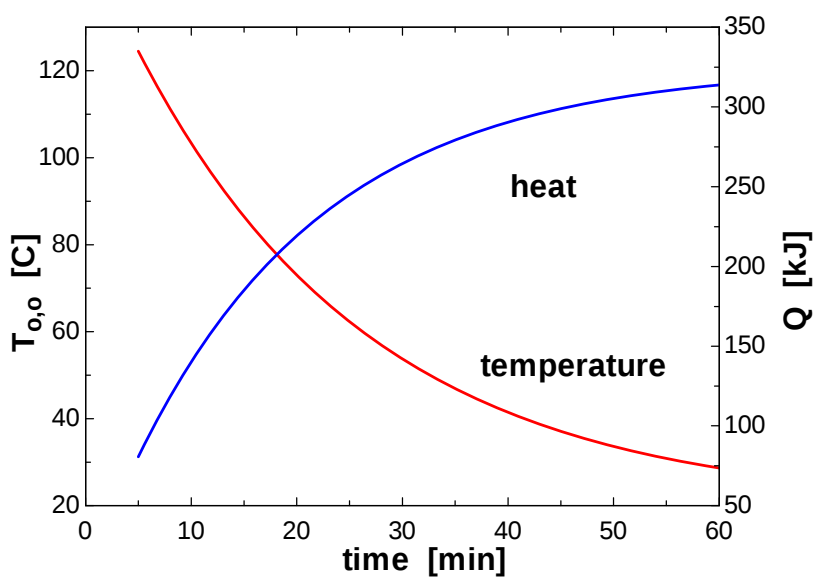
$Q_w = 1 - \theta_{ow} \cdot \sin(\lambda_{1w}) / \lambda_{1w}$ " $Q_w = (Q / Q_{\text{max}})_w$ "

$Q_c = 1 - 2 \cdot \theta_{oc} \cdot J_1 / \lambda_{1c}$ " $Q_c = (Q / Q_{\text{max}})_c$ "

$J_1 = 0.0835$ "From Table 11-3, at λ_{1c} "

$Q / Q_{\text{max}} = Q_w + Q_c \cdot (1 - Q_w)$ "total heat transfer"

time [min]	$T_{o,o}$ [C]	$T_{L,o}$ [C]	Q [kJ]
5	124.5	123.2	65.97
10	103.4	102.3	118.5
15	86.49	85.62	160.3
20	73.03	72.33	193.7
25	62.29	61.74	220.3
30	53.73	53.29	241.6
35	46.9	46.55	258.5
40	41.45	41.17	272
45	37.11	36.89	282.8
50	33.65	33.47	291.4
55	30.88	30.74	298.2
60	28.68	28.57	303.7



11-80 A semi-infinite aluminum cylinder is cooled by water. The temperature at the center of the cylinder 5 cm from the end surface is to be determined.

Assumptions 1 Heat conduction in the semi-infinite cylinder is two-dimensional, and thus the temperature varies in both the axial x - and the radial r - directions. 2 The thermal properties of the cylinder are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of aluminum are given to be $k = 237 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 9.71 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis This semi-infinite cylinder can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = 7.5 \text{ cm}$ and a semi-infinite medium. The dimensionless temperature 5 cm from the surface of a semi-infinite medium is first determined from

$$\begin{aligned} \frac{T(x, t) - T_i}{T_\infty - T_i} &= \text{erfc}\left(\frac{x}{\sqrt{\alpha t}}\right) - \exp\left(\frac{hx}{k} + \frac{h^2 \alpha t}{k^2}\right) \left[\text{erfc}\left(\frac{x}{2\sqrt{\alpha t}} + \frac{h\sqrt{\alpha t}}{k}\right) \right] \\ &= \text{erfc}\left(\frac{0.05}{2\sqrt{(9.71 \times 10^{-5})(8 \times 60)}}\right) - \exp\left(\frac{(140)(0.05)}{237} + \frac{(140)^2 (9.71 \times 10^{-5})(8 \times 60)}{(237)^2}\right) \\ &\quad \times \left[\text{erfc}\left(\frac{0.05}{2\sqrt{(9.71 \times 10^{-5})(8 \times 60)}} + \frac{(140)\sqrt{(9.71 \times 10^{-5})(8 \times 60)}}{237}\right) \right] \\ &= \text{erfc}(0.1158) - \exp(0.0458) \text{erfc}(0.2433) = 0.8699 - (1.0468)(0.7308) = 0.1049 \end{aligned}$$

$$\theta_{\text{semi-inf}} = \frac{T(x, t) - T_\infty}{T_i - T_\infty} = 1 - 0.1049 = 0.8951$$

The Biot number is calculated for the long cylinder to be

$$Bi = \frac{hr_o}{k} = \frac{(140 \text{ W/m}^2 \cdot ^\circ\text{C})(0.075 \text{ m})}{237 \text{ W/m} \cdot ^\circ\text{C}} = 0.0443$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.2948 \quad \text{and} \quad A_1 = 1.0110$$

The Fourier number is

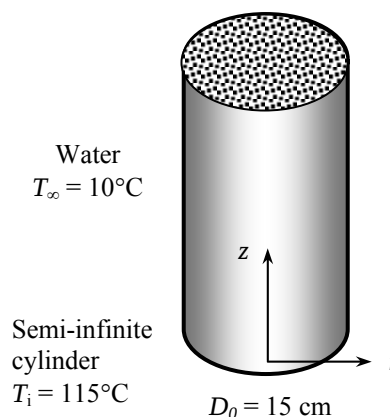
$$\tau = \frac{\alpha t}{r_o^2} = \frac{(9.71 \times 10^{-5} \text{ m}^2/\text{s})(8 \times 60 \text{ s})}{(0.075 \text{ m})^2} = 8.286 > 0.2$$

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the dimensionless temperature at the center of the plane wall is determined from

$$\theta_{o, \text{cyl}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.0110) e^{-(0.2948)^2 (8.286)} = 0.4921$$

The center temperature of the semi-infinite cylinder then becomes

$$\begin{aligned} \left[\frac{T(x, 0, t) - T_\infty}{T_i - T_\infty} \right]_{\text{semi-infinite cylinder}} &= \theta_{\text{semi-inf}}(x, t) \times \theta_{o, \text{cyl}} = 0.8951 \times 0.4921 = 0.4405 \\ \left[\frac{T(x, 0, t) - 10}{115 - 10} \right]_{\text{semi-infinite cylinder}} &= 0.4405 \longrightarrow T(x, 0, t) = \mathbf{56.3^\circ\text{C}} \end{aligned}$$



11-81E A hot dog is dropped into boiling water. The center temperature of the hot dog is to be determined by treating hot dog as a finite cylinder and also as an infinitely long cylinder.

Assumptions 1 When treating hot dog as a finite cylinder, heat conduction in the hot dog is two-dimensional, and thus the temperature varies in both the axial x - and the radial r - directions. When treating hot dog as an infinitely long cylinder, heat conduction is one-dimensional in the radial r - direction. **2** The thermal properties of the hot dog are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the hot dog are given to be $k = 0.44 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$, $\rho = 61.2 \text{ lbm/ft}^3$, $c_p = 0.93 \text{ Btu/lbm}\cdot^\circ\text{F}$, and $\alpha = 0.0077 \text{ ft}^2/\text{h}$.

Analysis (a) This hot dog can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = (0.4/12) \text{ ft}$ and a plane wall of thickness $2L = (5/12) \text{ ft}$. The distance x is measured from the midplane.

After 5 minutes

First the Biot number is calculated for the plane wall to be

$$Bi = \frac{hL}{k} = \frac{(120 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(2.5/12 \text{ ft})}{(0.44 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 56.8$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.5421 \quad \text{and} \quad A_1 = 1.2728$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(5/60 \text{ h})}{(2.5/12 \text{ ft})^2} = 0.015 < 0.2 \quad (\text{Be cautious!})$$

Then the dimensionless temperature at the center of the plane wall is determined from

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2728)e^{-(1.5421)^2(0.015)} = 1.228$$

We repeat the same calculations for the long cylinder,

$$Bi = \frac{hr_o}{k} = \frac{(120 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.4/12 \text{ ft})}{(0.44 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 9.1$$

$$\lambda_1 = 2.1589 \quad \text{and} \quad A_1 = 1.5618$$

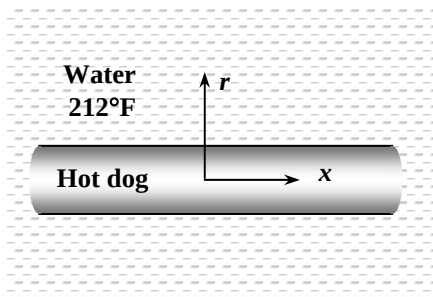
$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(5/60 \text{ h})}{(0.4/12 \text{ ft})^2} = 0.578 > 0.2$$

$$\theta_{o,\text{cyl}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5618)e^{-(2.1589)^2(0.578)} = 0.106$$

Then the center temperature of the short cylinder becomes

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{o,\text{wall}} \times \theta_{o,\text{cyl}} = 1.228 \times 0.106 = 0.130$$

$$\frac{T(0,0,t) - 212}{40 - 212} = 0.130 \longrightarrow T(0,0,t) = \mathbf{190^\circ\text{F}}$$



After 10 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(10/60 \text{ h})}{(2.5/12 \text{ ft})^2} = 0.03 < 0.2 \quad (\text{Be cautious!})$$

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2728)e^{-(1.5421)^2(0.03)} = 1.185$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(10/60 \text{ h})}{(0.4/12 \text{ ft})^2} = 1.156 > 0.2$$

$$\theta_{o,\text{cyl}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5618)e^{-(2.1589)^2(1.156)} = 0.0071$$

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{o,\text{wall}} \times \theta_{o,\text{cyl}} = 1.185 \times 0.0071 = 0.0084$$

$$\frac{T(0,0,t) - 212}{40 - 212} = 0.0084 \longrightarrow T(0,0,t) = \mathbf{211^\circ\text{F}}$$

After 15 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(15/60 \text{ h})}{(2.5/12 \text{ ft})^2} = 0.045 < 0.2 \quad (\text{Be cautious!})$$

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2728)e^{-(1.5421)^2(0.045)} = 1.143$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(15/60 \text{ h})}{(0.4/12 \text{ ft})^2} = 1.734 > 0.2$$

$$\theta_{o,\text{cyl}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5618)e^{-(2.1589)^2(1.734)} = 0.00048$$

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{o,\text{wall}} \times \theta_{o,\text{cyl}} = 1.143 \times 0.00048 = 0.00055$$

$$\frac{T(0,0,t) - 212}{40 - 212} = 0.00055 \longrightarrow T(0,0,t) = \mathbf{212^\circ\text{F}}$$

(b) Treating the hot dog as an infinitely long cylinder will not change the results obtained in the part (a) since dimensionless temperatures for the plane wall is 1 for all cases.

11-82E A hot dog is dropped into boiling water. The center temperature of the hot dog is to be determined by treating hot dog as a finite cylinder and an infinitely long cylinder.

Assumptions 1 When treating hot dog as a finite cylinder, heat conduction in the hot dog is two-dimensional, and thus the temperature varies in both the axial x - and the radial r - directions. When treating hot dog as an infinitely long cylinder, heat conduction is one-dimensional in the radial r - direction. **2** The thermal properties of the hot dog are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the hot dog are given to be $k = 0.44 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$, $\rho = 61.2 \text{ lbm/ft}^3$, $c_p = 0.93 \text{ Btu/lbm}\cdot^\circ\text{F}$, and $\alpha = 0.0077 \text{ ft}^2/\text{h}$.

Analysis (a) This hot dog can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = (0.4/12) \text{ ft}$ and a plane wall of thickness $2L = (5/12) \text{ ft}$. The distance x is measured from the midplane.

After 5 minutes

First the Biot number is calculated for the plane wall to be

$$Bi = \frac{hL}{k} = \frac{(120 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(2.5/12 \text{ ft})}{(0.44 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 56.8$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.5421 \quad \text{and} \quad A_1 = 1.2728$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(5/60 \text{ h})}{(2.5/12 \text{ ft})^2} = 0.015 < 0.2 \quad (\text{Be cautious!})$$

Then the dimensionless temperature at the center of the plane wall is determined from

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2728)e^{-(1.5421)^2(0.015)} = 1.228$$

We repeat the same calculations for the long cylinder,

$$Bi = \frac{hr_o}{k} = \frac{(120 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.4/12 \text{ ft})}{(0.44 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 9.1$$

$$\lambda_1 = 2.1589 \quad \text{and} \quad A_1 = 1.5618$$

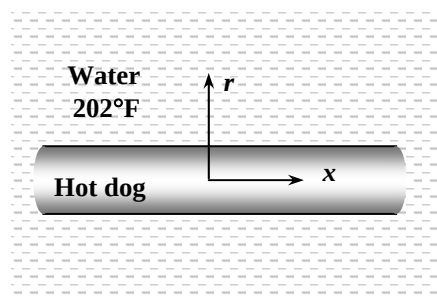
$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(5/60 \text{ h})}{(0.4/12 \text{ ft})^2} = 0.578 > 0.2$$

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5618)e^{-(2.1589)^2(0.578)} = 0.106$$

Then the center temperature of the short cylinder becomes

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{0,\text{wall}} \times \theta_{0,\text{cyl}} = 1.228 \times 0.106 = 0.130$$

$$\frac{T(0,0,t) - 202}{40 - 202} = 0.130 \longrightarrow T(0,0,t) = \mathbf{181^\circ\text{F}}$$



After 10 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(10/60 \text{ h})}{(2.5/12 \text{ ft})^2} = 0.03 < 0.2 \quad (\text{Be cautious!})$$

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2728)e^{-(1.5421)^2(0.03)} = 1.185$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(10/60 \text{ h})}{(0.4/12 \text{ ft})^2} = 1.156 > 0.2$$

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5618)e^{-(2.1589)^2(1.156)} = 0.007$$

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{0,\text{wall}} \times \theta_{0,\text{cyl}} = 1.185 \times 0.0071 = 0.0084$$

$$\frac{T(0,0,t) - 202}{40 - 202} = 0.0084 \longrightarrow T(0,0,t) = \mathbf{201^\circ\text{F}}$$

After 15 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(15/60 \text{ h})}{(2.5/12 \text{ ft})^2} = 0.045 < 0.2 \quad (\text{Be cautious!})$$

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2728)e^{-(1.5421)^2(0.045)} = 1.143$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.0077 \text{ ft}^2/\text{h})(15/60 \text{ h})}{(0.4/12 \text{ ft})^2} = 1.734 > 0.2$$

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.5618)e^{-(2.1589)^2(1.734)} = 0.00048$$

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta_{0,\text{wall}} \times \theta_{0,\text{cyl}} = 1.143 \times 0.00048 = 0.00055$$

$$\frac{T(0,0,t) - 202}{40 - 202} = 0.00055 \longrightarrow T(0,0,t) = \mathbf{202^\circ\text{F}}$$

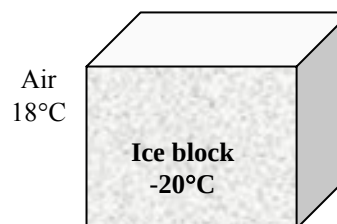
(b) Treating the hot dog as an infinitely long cylinder will not change the results obtained in the part (a) since dimensionless temperatures for the plane wall is 1 for all cases.

11-83 A rectangular ice block is placed on a table. The time the ice block starts melting is to be determined.

Assumptions 1 Heat conduction in the ice block is two-dimensional, and thus the temperature varies in both x - and y - directions. 2 The thermal properties of the ice block are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the ice are given to be $k = 2.22 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.124 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis This rectangular ice block can be treated as a short rectangular block that can physically be formed by the intersection of two infinite plane wall of thickness $2L = 4 \text{ cm}$ and an infinite plane wall of thickness $2L = 10 \text{ cm}$. We measure x from the bottom surface of the block since this surface represents the adiabatic center surface of the plane wall of thickness $2L = 10 \text{ cm}$. Since the melting starts at the corner of the top surface, we need to determine the time required to melt ice block which will happen when the temperature drops below 0°C at this location. The Biot numbers and the corresponding constants are first determined to be



$$Bi_{\text{wall},1} = \frac{hL_1}{k} = \frac{(12 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02 \text{ m})}{(2.22 \text{ W/m} \cdot ^\circ\text{C})} = 0.1081 \longrightarrow \lambda_1 = 0.3208 \text{ and } A_1 = 1.0173$$

$$Bi_{\text{wall},3} = \frac{hL_3}{k} = \frac{(12 \text{ W/m}^2 \cdot ^\circ\text{C})(0.05 \text{ m})}{(2.22 \text{ W/m} \cdot ^\circ\text{C})} = 0.2703 \longrightarrow \lambda_1 = 0.4951 \text{ and } A_1 = 1.0408$$

The ice will start melting at the corners because of the maximum exposed surface area there. Noting that $\tau = \alpha / L^2$ and assuming that $\tau > 0.2$ in all dimensions so that the one-term approximate solution for transient heat conduction is applicable, the product solution method can be written for this problem as

$$\begin{aligned} \theta(L_1, L_2, L_3, t)_{\text{block}} &= \theta(L_1, t)_{\text{wall},1} \theta(L_3, t)_{\text{wall},2} \\ \frac{0-18}{-20-18} &= \left[A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L_1 / L_1) \right]^2 \left[A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L_3 / L_3) \right] \\ 0.4737 &= \left\{ (1.0173) \exp \left[- (0.3208)^2 \frac{(0.124 \times 10^{-7}) t}{(0.02)^2} \right] \cos(0.3208) \right\}^2 \\ &\quad \times \left\{ (1.0408) \exp \left[- (0.4951)^2 \frac{(0.124 \times 10^{-7}) t}{(0.05)^2} \right] \cos(0.4951) \right\} \end{aligned}$$

$$\longrightarrow t = 77,500 \text{ s} = 1292 \text{ min} = \mathbf{21.5 \text{ hours}}$$

Therefore, the ice will start melting in about 21 hours.

Discussion Note that

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.124 \times 10^{-7} \text{ m}^2/\text{s})(77,500 \text{ s/h})}{(0.05 \text{ m})^2} = 0.384 > 0.2$$

and thus the assumption of $\tau > 0.2$ for the applicability of the one-term approximate solution is verified.

11-84 EES Prob. 11-83 is reconsidered. The effect of the initial temperature of the ice block on the time period before the ice block starts melting is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$2*L_1=0.04 \text{ [m]}$$

$$L_2=L_1$$

$$2*L_3=0.10 \text{ [m]}$$

$$T_i=-20 \text{ [C]}$$

$$T_{\text{infinity}}=18 \text{ [C]}$$

$$h=12 \text{ [W/m}^2\text{-C]}$$

$$T_{L1_L2_L3}=0 \text{ [C]}$$

"PROPERTIES"

$$k=2.22 \text{ [W/m-C]}$$

$$\alpha=0.124\text{E-7} \text{ [m}^2\text{/s]}$$

"ANALYSIS"

"This block can physically be formed by the intersection of two infinite plane wall of thickness $2L=4$ cm and an infinite plane wall of thickness $2L=10$ cm"

"For the two plane walls"

$$Bi_{w1}=(h*L_1)/k$$

"From Table 11-2 corresponding to this Bi number, we read"

$$\lambda_{1_w1}=0.3208 \text{ "w stands for wall"}$$

$$A_{1_w1}=1.0173$$

$$\text{time*Convert(min, s)}=\tau_{w1}*L_1^2/\alpha$$

"For the third plane wall"

$$Bi_{w3}=(h*L_3)/k$$

"From Table 11-2 corresponding to this Bi number, we read"

$$\lambda_{1_w3}=0.4951$$

$$A_{1_w3}=1.0408$$

$$\text{time*Convert(min, s)}=\tau_{w3}*L_3^2/\alpha$$

$$\theta_{L_w1}=A_{1_w1}*\exp(-\lambda_{1_w1}^2*\tau_{w1})*\cos(\lambda_{1_w1}*L_1/L_1)$$

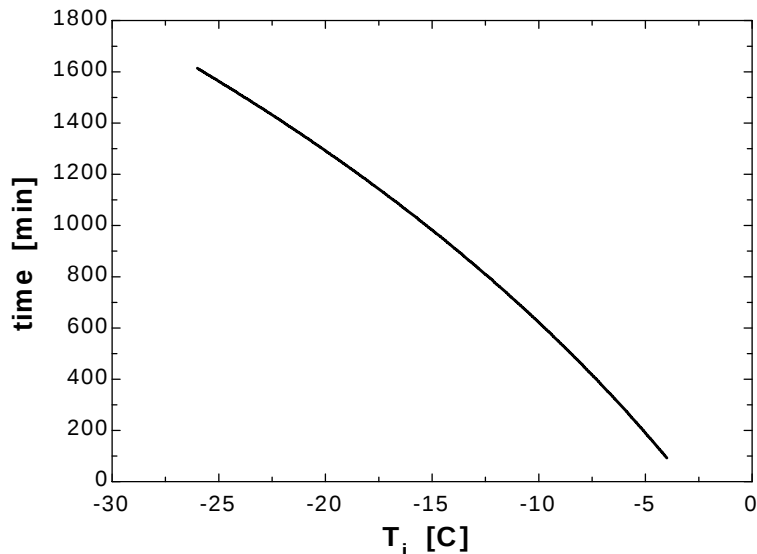
$$\theta_{L_w1}=(T_{L_w1}-T_{\text{infinity}})/(T_i-T_{\text{infinity}})$$

$$\theta_{L_w3}=A_{1_w3}*\exp(-\lambda_{1_w3}^2*\tau_{w3})*\cos(\lambda_{1_w3}*L_3/L_3)$$

$$\theta_{L_w3}=(T_{L_w3}-T_{\text{infinity}})/(T_i-T_{\text{infinity}})$$

$$(T_{L1_L2_L3}-T_{\text{infinity}})/(T_i-T_{\text{infinity}})=\theta_{L_w1}^2*\theta_{L_w3} \text{ "corner temperature"}$$

T_i [C]	time [min]
-26	1614
-24	1512
-22	1405
-20	1292
-18	1173
-16	1048
-14	914.9
-12	773.3
-10	621.9
-8	459.4
-6	283.7
-4	92.84

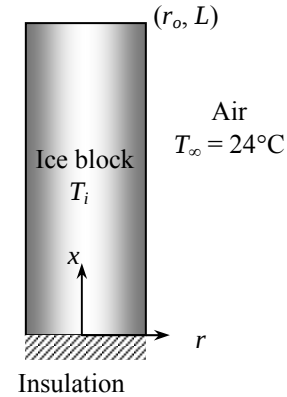


11-85 A cylindrical ice block is placed on a table. The initial temperature of the ice block to avoid melting for 2 h is to be determined.

Assumptions 1 Heat conduction in the ice block is two-dimensional, and thus the temperature varies in both x - and r - directions. 2 Heat transfer from the base of the ice block to the table is negligible. 3 The thermal properties of the ice block are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the ice are given to be $k = 2.22 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.124 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis This cylindrical ice block can be treated as a short cylinder that can physically be formed by the intersection of a long cylinder of diameter $D = 2 \text{ cm}$ and an infinite plane wall of thickness $2L = 4 \text{ cm}$. We measure x from the bottom surface of the block since this surface represents the adiabatic center surface of the plane wall of thickness $2L = 4 \text{ cm}$. The melting starts at the outer surfaces of the top surface when the temperature drops below 0°C at this location. The Biot numbers, the corresponding constants, and the Fourier numbers are



$$Bi_{\text{wall}} = \frac{hL}{k} = \frac{(13 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02 \text{ m})}{(2.22 \text{ W/m} \cdot ^\circ\text{C})} = 0.1171 \longrightarrow \lambda_1 = 0.3319 \quad \text{and} \quad A_1 = 1.0187$$

$$Bi_{\text{cyl}} = \frac{hr_o}{k} = \frac{(13 \text{ W/m}^2 \cdot ^\circ\text{C})(0.01 \text{ m})}{(2.22 \text{ W/m} \cdot ^\circ\text{C})} = 0.05856 \longrightarrow \lambda_1 = 0.3393 \quad \text{and} \quad A_1 = 1.0144$$

$$\tau_{\text{wall}} = \frac{\alpha t}{L^2} = \frac{(0.124 \times 10^{-7} \text{ m}^2/\text{s})(3 \text{ h} \times 3600 \text{ s/h})}{(0.02 \text{ m})^2} = 0.3348 > 0.2$$

$$\tau_{\text{cyl}} = \frac{\alpha t}{r_o^2} = \frac{(0.124 \times 10^{-7} \text{ m}^2/\text{s})(3 \text{ h} \times 3600 \text{ s/h})}{(0.01 \text{ m})^2} = 1.3392 > 0.2$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable. The product solution for this problem can be written as

$$\begin{aligned} \theta(L, r_o, t)_{\text{block}} &= \theta(L, t)_{\text{wall}} \theta(r_o, t)_{\text{cyl}} \\ \frac{0 - 24}{T_i - 24} &= \left[A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L / L) \right] \left[A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r_o / r_o) \right] \\ \frac{0 - 24}{T_i - 24} &= \left[(1.0187) e^{-(0.3319)^2 (0.3348)} \cos(0.3319) \right] \left[(1.0146) e^{-(0.3393)^2 (1.3392)} (0.9708) \right] \end{aligned}$$

which gives $T_i = -6.6^\circ\text{C}$

Therefore, the ice will not start melting for at least 3 hours if its initial temperature is -6.6°C or below.

11-86 A cubic block and a cylindrical block are exposed to hot gases on all of their surfaces. The center temperatures of each geometry in 10, 20, and 60 min are to be determined.

Assumptions 1 Heat conduction in the cubic block is three-dimensional, and thus the temperature varies in all x -, y , and z - directions. 2 Heat conduction in the cylindrical block is two-dimensional, and thus the temperature varies in both axial x - and radial r - directions. 3 The thermal properties of the granite are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the granite are given to be $k = 2.5 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.15 \times 10^{-6} \text{ m}^2/\text{s}$.

Analysis:

Cubic block: This cubic block can physically be formed by the intersection of three infinite plane walls of thickness $2L = 5 \text{ cm}$.

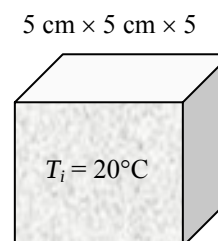
After 10 minutes: The Biot number, the corresponding constants, and the Fourier number are

$$Bi = \frac{hL}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.025 \text{ m})}{(2.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.400 \longrightarrow \lambda_1 = 0.5932 \text{ and } A_1 = 1.0580$$

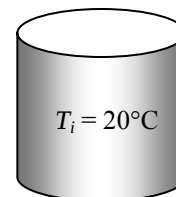
$$\tau = \frac{\alpha t}{L^2} = \frac{(1.15 \times 10^{-6} \text{ m}^2/\text{s})(10 \text{ min} \times 60 \text{ s/min})}{(0.025 \text{ m})^2} = 1.104 > 0.2$$

To determine the center temperature, the product solution can be written as

$$\begin{aligned} \theta(0,0,0,t)_{\text{block}} &= [\theta(0,t)_{\text{wall}}]^3 \\ \frac{T(0,0,0,t) - T_\infty}{T_i - T_\infty} &= \left(A_1 e^{-\lambda_1^2 \tau} \right)^3 \\ \frac{T(0,0,0,t) - 500}{20 - 500} &= \left\{ (1.0580) e^{-(0.5932)^2 (1.104)} \right\}^3 = 0.369 \\ T(0,0,0,t) &= \mathbf{323^\circ\text{C}} \end{aligned}$$



Hot gases
500°C



After 20 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(1.15 \times 10^{-6} \text{ m}^2/\text{s})(20 \text{ min} \times 60 \text{ s/min})}{(0.025 \text{ m})^2} = 2.208 > 0.2$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = \left\{ (1.0580) e^{-(0.5932)^2 (2.208)} \right\}^3 = 0.115 \longrightarrow T(0,0,0,t) = \mathbf{445^\circ\text{C}}$$

After 60 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(1.15 \times 10^{-6} \text{ m}^2/\text{s})(60 \text{ min} \times 60 \text{ s/min})}{(0.025 \text{ m})^2} = 6.624 > 0.2$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = \left\{ (1.0580) e^{-(0.5932)^2 (6.624)} \right\}^3 = 0.00109 \longrightarrow T(0,0,0,t) = \mathbf{500^\circ\text{C}}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

Cylinder: This cylindrical block can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = 2.5$ cm and a plane wall of thickness $2L = 5$ cm.

After 10 minutes: The Biot number and the corresponding constants for the long cylinder are

$$Bi = \frac{hr_o}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.025 \text{ m})}{(2.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.400 \longrightarrow \lambda_1 = 0.8516 \text{ and } A_1 = 1.0931$$

To determine the center temperature, the product solution can be written as

$$\begin{aligned} \theta(0,0,t)_{block} &= [\theta(0,t)_{wall}] [\theta(0,t)_{cyl}] \\ \frac{T(0,0,t) - T_\infty}{T_i - T_\infty} &= \left(A_1 e^{-\lambda_1^2 \tau} \right)_{wall} \left(A_1 e^{-\lambda_1^2 \tau} \right)_{cyl} \\ \frac{T(0,0,t) - 500}{20 - 500} &= \left\{ (1.0580) e^{-(0.5932)^2 (1.104)} \right\} \left\{ (1.0931) e^{-(0.8516)^2 (1.104)} \right\} = 0.352 \longrightarrow T(0,0,t) = 331^\circ\text{C} \end{aligned}$$

After 20 minutes

$$\frac{T(0,0,t) - 500}{20 - 500} = \left\{ (1.0580) e^{-(0.5932)^2 (2.208)} \right\} \left\{ (1.0931) e^{-(0.8516)^2 (2.208)} \right\} = 0.107 \longrightarrow T(0,0,t) = 449^\circ\text{C}$$

After 60 minutes

$$\frac{T(0,0,t) - 500}{20 - 500} = \left\{ (1.0580) e^{-(0.5932)^2 (6.624)} \right\} \left\{ (1.0931) e^{-(0.8516)^2 (6.624)} \right\} = 0.00092 \longrightarrow T(0,0,t) = 500^\circ\text{C}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

11-87 A cubic block and a cylindrical block are exposed to hot gases on all of their surfaces. The center temperatures of each geometry in 10, 20, and 60 min are to be determined.

Assumptions 1 Heat conduction in the cubic block is three-dimensional, and thus the temperature varies in all x -, y -, and z - directions. 2 Heat conduction in the cylindrical block is two-dimensional, and thus the temperature varies in both axial x - and radial r - directions. 3 The thermal properties of the granite are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the granite are $k = 2.5 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.15 \times 10^{-6} \text{ m}^2/\text{s}$.

Analysis:

Cubic block: This cubic block can physically be formed by the intersection of three infinite plane wall of thickness $2L = 5 \text{ cm}$. Two infinite plane walls are exposed to the hot gases with a heat transfer coefficient of $h = 40 \text{ W/m}^2 \cdot ^\circ\text{C}$ and one with $h = 80 \text{ W/m}^2 \cdot ^\circ\text{C}$.

After 10 minutes: The Biot number and the corresponding constants for $h = 40 \text{ W/m}^2 \cdot ^\circ\text{C}$ are

$$Bi = \frac{hL}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.025 \text{ m})}{(2.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.400 \longrightarrow \lambda_1 = 0.5932 \text{ and } A_1 = 1.0580$$

The Biot number and the corresponding constants for $h = 80 \text{ W/m}^2 \cdot ^\circ\text{C}$ are

$$Bi = \frac{hL}{k} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.025 \text{ m})}{(2.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.800$$

$$\longrightarrow \lambda_1 = 0.7910 \text{ and } A_1 = 1.1016$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(1.15 \times 10^{-6} \text{ m}^2/\text{s})(10 \text{ min} \times 60 \text{ s/min})}{(0.025 \text{ m})^2} = 1.104 > 0.2$$

To determine the center temperature, the product solution method can be written as

$$\begin{aligned} \theta(0,0,0,t)_{\text{block}} &= [\theta(0,t)_{\text{wall}}]^2 [\theta(0,t)_{\text{wall}}] \\ \frac{T(0,0,0,t) - T_\infty}{T_i - T_\infty} &= \left(A_1 e^{-\lambda_1^2 \tau} \right)^2 \left(A_1 e^{-\lambda_1^2 \tau} \right) \\ \frac{T(0,0,0,t) - 500}{20 - 500} &= \left\{ (1.0580) e^{-(0.5932)^2 (1.104)} \right\}^2 \left\{ (1.1016) e^{-(0.7910)^2 (1.104)} \right\} = 0.284 \end{aligned}$$

$$T(0,0,0,t) = 364^\circ\text{C}$$

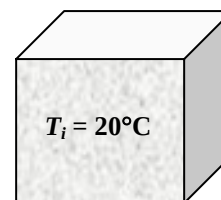
After 20 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(1.15 \times 10^{-6} \text{ m}^2/\text{s})(20 \text{ min} \times 60 \text{ s/min})}{(0.025 \text{ m})^2} = 2.208 > 0.2$$

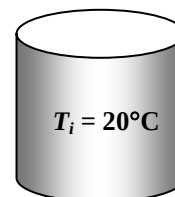
$$\frac{T(0,0,0,t) - 500}{20 - 500} = \left\{ (1.0580) e^{-(0.5932)^2 (2.208)} \right\}^2 \left\{ (1.1016) e^{-(0.7910)^2 (2.208)} \right\} = 0.0654$$

$$\longrightarrow T(0,0,0,t) = 469^\circ\text{C}$$

5 cm $\times$ 5 cm $\times$ 5



Hot gases
500°C



After 60 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(1.15 \times 10^{-6} \text{ m}^2/\text{s})(60 \text{ min} \times 60 \text{ s/min})}{(0.025 \text{ m})^2} = 6.624 > 0.2$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = \left\{ (1.0580)e^{-(0.5932)^2(6.624)} \right\}^2 \left\{ (1.1016)e^{-(0.7910)^2(6.624)} \right\} = 0.000186$$

$$\longrightarrow T(0,0,0,t) = \mathbf{500^\circ\text{C}}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

Cylinder: This cylindrical block can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = 2.5 \text{ cm}$ exposed to the hot gases with a heat transfer coefficient of $h = 40 \text{ W/m}^2 \cdot ^\circ\text{C}$ and a plane wall of thickness $2L = 5 \text{ cm}$ exposed to the hot gases with $h = 80 \text{ W/m}^2 \cdot ^\circ\text{C}$.

After 10 minutes: The Biot number and the corresponding constants for the long cylinder are

$$Bi = \frac{hr_o}{k} = \frac{(40 \text{ W/m}^2 \cdot ^\circ\text{C})(0.025 \text{ m})}{(2.5 \text{ W/m} \cdot ^\circ\text{C})} = 0.400 \longrightarrow \lambda_1 = 0.8516 \text{ and } A_1 = 1.0931$$

To determine the center temperature, the product solution method can be written as

$$\theta(0,0,t)_{\text{block}} = [\theta(0,t)_{\text{wall}}][\theta(0,t)_{\text{cyl}}]$$

$$\frac{T(0,0,t) - T_\infty}{T_i - T_\infty} = \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{wall}} \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{cyl}}$$

$$\frac{T(0,0,t) - 500}{20 - 500} = \left\{ (1.1016)e^{-(0.7910)^2(1.104)} \right\} \left\{ (1.0931)e^{-(0.8516)^2(1.104)} \right\} = 0.271$$

$$T(0,0,t) = \mathbf{370^\circ\text{C}}$$

After 20 minutes

$$\frac{T(0,0,t) - 500}{20 - 500} = \left\{ (1.1016)e^{-(0.7910)^2(2.208)} \right\} \left\{ (1.0931)e^{-(0.8516)^2(2.208)} \right\} = 0.06094 \longrightarrow T(0,0,t) = \mathbf{471^\circ\text{C}}$$

After 60 minutes

$$\frac{T(0,0,t) - 500}{20 - 500} = \left\{ (1.1016)e^{-(0.7910)^2(6.624)} \right\} \left\{ (1.0931)e^{-(0.8516)^2(6.624)} \right\} = 0.0001568 \longrightarrow T(0,0,t) = \mathbf{500^\circ\text{C}}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

11-88 A cylindrical aluminum block is heated in a furnace. The length of time the block should be kept in the furnace and the amount of heat transfer to the block are to be determined.

Assumptions 1 Heat conduction in the cylindrical block is two-dimensional, and thus the temperature varies in both axial x - and radial r - directions. 2 The thermal properties of the aluminum are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (it will be verified).

Properties The thermal properties of the aluminum block are given to be $k = 236 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 2702 \text{ kg/m}^3$, $c_p = 0.896 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 9.75 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis This cylindrical aluminum block can physically be formed by the intersection of an infinite plane wall of thickness $2L = 20 \text{ cm}$, and a long cylinder of radius $r_o = D/2 = 7.5 \text{ cm}$. The Biot numbers and the corresponding constants are first determined to be

$$Bi = \frac{hL}{k} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1 \text{ m})}{(236 \text{ W/m} \cdot ^\circ\text{C})} = 0.0339 \quad \longrightarrow \lambda_1 = 0.1811 \quad \text{and} \quad A_1 = 1.0056$$

$$Bi = \frac{hr_o}{k} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.075 \text{ m})}{236 \text{ W/m} \cdot ^\circ\text{C}} = 0.0254 \quad \longrightarrow \lambda_1 = 0.2217 \quad \text{and} \quad A_1 = 1.0063$$

Noting that $\tau = \alpha t / L^2$ and assuming $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable, the product solution for this problem can be written as

$$\begin{aligned} \theta(0,0,t)_{\text{block}} &= \theta(0,t)_{\text{wall}} \theta(0,t)_{\text{cyl}} = \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{wall}} \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{cyl}} \\ \frac{300 - 1200}{20 - 1200} &= \left\{ (1.0056) \exp \left[- (0.1811)^2 \frac{(9.75 \times 10^{-5}) t}{(0.1)^2} \right] \right\} \times \left\{ (1.0063) \exp \left[- (0.2217)^2 \frac{(9.75 \times 10^{-5}) t}{(0.075)^2} \right] \right\} \\ &= 0.7627 \end{aligned}$$

Solving for the time t gives

$$t = 241 \text{ s} = \mathbf{4.0 \text{ min.}}$$

We note that

$$\begin{aligned} \tau_{\text{wall}} &= \frac{\alpha t}{L^2} = \frac{(9.75 \times 10^{-5} \text{ m}^2/\text{s})(241 \text{ s})}{(0.1 \text{ m})^2} = 2.350 > 0.2 \\ \tau_{\text{cyl}} &= \frac{\alpha t}{r_o^2} = \frac{(9.75 \times 10^{-5} \text{ m}^2/\text{s})(241 \text{ s})}{(0.075 \text{ m})^2} = 4.177 > 0.2 \end{aligned}$$

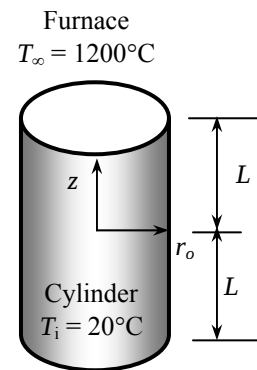
and thus the assumption of $\tau > 0.2$ for the applicability of the one-term approximate solution is verified. The dimensionless temperatures at the center are

$$\begin{aligned} \theta(0,t)_{\text{wall}} &= \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{wall}} = (1.0056) \exp \left[- (0.1811)^2 (2.350) \right] = 0.9310 \\ \theta(0,t)_{\text{cyl}} &= \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{cyl}} = (1.0063) \exp \left[- (0.2217)^2 (4.177) \right] = 0.8195 \end{aligned}$$

The maximum amount of heat transfer is

$$\begin{aligned} m &= \rho V = \rho \pi r_o^2 L = (2702 \text{ kg/m}^3) [\pi (0.075 \text{ m})^2 (0.2 \text{ m})] = 9.550 \text{ kg} \\ Q_{\text{max}} &= mc_p (T_i - T_\infty) = (9.550 \text{ kg})(0.896 \text{ kJ/kg} \cdot ^\circ\text{C})(20 - 1200)^\circ\text{C} = 10,100 \text{ kJ} \end{aligned}$$

Then we determine the dimensionless heat transfer ratios for both geometries as



$$\left(\frac{Q}{Q_{\max}}\right)_{\text{wall}} = 1 - \theta_{o,\text{wall}} \frac{\sin(\lambda_1)}{\lambda_1} = 1 - (0.9310) \frac{\sin(0.1811)}{0.1811} = 0.07408$$

$$\left(\frac{Q}{Q_{\max}}\right)_{\text{cyl}} = 1 - 2\theta_{o,\text{cyl}} \frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2(0.8195) \frac{0.1101}{0.2217} = 0.1860$$

The heat transfer ratio for the short cylinder is

$$\begin{aligned} \left(\frac{Q}{Q_{\max}}\right)_{\text{short cylinder}} &= \left(\frac{Q}{Q_{\max}}\right)_{\text{plane wall}} + \left(\frac{Q}{Q_{\max}}\right)_{\text{long cylinder}} \left[1 - \left(\frac{Q}{Q_{\max}}\right)_{\text{plane wall}} \right] \\ &= 0.07408 + (0.1860)(1 - 0.07408) = 0.2463 \end{aligned}$$

Then the total heat transfer from the short cylinder as it is cooled from 300°C at the center to 20°C becomes

$$Q = 0.2463 Q_{\max} = (0.2463)(10,100 \text{ kJ}) = \mathbf{2490 \text{ kJ}}$$

which is identical to the heat transfer to the cylinder as the cylinder at 20°C is heated to 300°C at the center.

11-89 A cylindrical aluminum block is heated in a furnace. The length of time the block should be kept in the furnace and the amount of heat transferred to the block are to be determined.

Assumptions 1 Heat conduction in the cylindrical block is two-dimensional, and thus the temperature varies in both axial x - and radial r - directions. 2 Heat transfer from the bottom surface of the block is negligible. 3 The thermal properties of the aluminum are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the aluminum block are given to be $k = 236 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 2702 \text{ kg/m}^3$, $c_p = 0.896 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 9.75 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis This cylindrical aluminum block can physically be formed by the intersection of an infinite plane wall of thickness $2L = 40 \text{ cm}$ and a long cylinder of radius $r_o = D/2 = 7.5 \text{ cm}$. Note that the height of the short cylinder represents the half thickness of the infinite plane wall where the bottom surface of the short cylinder is adiabatic. The Biot numbers and corresponding constants are first determined to be

$$Bi = \frac{hL}{k} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.2 \text{ m})}{(236 \text{ W/m} \cdot ^\circ\text{C})} = 0.0678 \rightarrow \lambda_1 = 0.2568 \text{ and } A_1 = 1.0110$$

$$Bi = \frac{hr_o}{k} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.075 \text{ m})}{(236 \text{ W/m} \cdot ^\circ\text{C})} = 0.0254 \rightarrow \lambda_1 = 0.2217 \text{ and } A_1 = 1.0063$$

Noting that $\tau = \alpha t / L^2$ and assuming $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable, the product solution for this problem can be written as

$$\begin{aligned} \theta(0,0,t)_{\text{block}} &= \theta(0,t)_{\text{wall}} \theta(0,t)_{\text{cyl}} = \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{wall}} \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{cyl}} \\ \frac{300-1200}{20-1200} &= \left\{ (1.0110) \exp \left[- (0.2568)^2 \frac{(9.75 \times 10^{-5})t}{(0.2)^2} \right] \right\} \left\{ (1.0063) \exp \left[- (0.2217)^2 \frac{(9.75 \times 10^{-5})t}{(0.075)^2} \right] \right\} \\ &= 0.7627 \end{aligned}$$

Solving for the time t gives

$$t = 285 \text{ s} = \mathbf{4.7 \text{ min.}}$$

We note that

$$\tau_{\text{wall}} = \frac{\alpha t}{L^2} = \frac{(9.75 \times 10^{-5} \text{ m}^2/\text{s})(285 \text{ s})}{(0.2 \text{ m})^2} = 0.6947 > 0.2$$

$$\tau_{\text{cyl}} = \frac{\alpha t}{r_o^2} = \frac{(9.75 \times 10^{-5} \text{ m}^2/\text{s})(285 \text{ s})}{(0.075 \text{ m})^2} = 4.940 > 0.2$$

and thus the assumption of $\tau > 0.2$ for the applicability of the one-term approximate solution is verified. The dimensionless temperatures at the center are

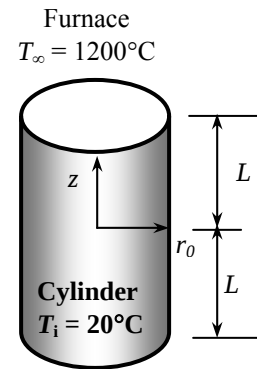
$$\theta(0,t)_{\text{wall}} = \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{wall}} = (1.0110) \exp \left[- (0.2568)^2 (0.6947) \right] = 0.9658$$

$$\theta(0,t)_{\text{cyl}} = \left(A_1 e^{-\lambda_1^2 \tau} \right)_{\text{cyl}} = (1.0063) \exp \left[- (0.2217)^2 (4.940) \right] = 0.7897$$

The maximum amount of heat transfer is

$$\begin{aligned} m &= \rho V = \rho \pi r_o^2 L = (2702 \text{ kg/m}^3) [\pi (0.075 \text{ m})^2 (0.2 \text{ m})] = 9.55 \text{ kg} \\ Q_{\text{max}} &= mc_p (T_i - T_\infty) = (9.55 \text{ kg})(0.896 \text{ kJ/kg} \cdot ^\circ\text{C})(20 - 1200)^\circ\text{C} = 10,100 \text{ kJ} \end{aligned}$$

Then we determine the dimensionless heat transfer ratios for both geometries as



$$\left(\frac{Q}{Q_{\max}}\right)_{\text{wall}} = 1 - \theta_{o,\text{wall}} \frac{\sin(\lambda_1)}{\lambda_1} = 1 - (0.9658) \frac{\sin(0.2568)}{0.2568} = 0.04477$$

$$\left(\frac{Q}{Q_{\max}}\right)_{\text{cyl}} = 1 - 2\theta_{o,\text{cyl}} \frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2(0.7897) \frac{0.1101}{0.2217} = 0.2156$$

The heat transfer ratio for the short cylinder is

$$\begin{aligned} \left(\frac{Q}{Q_{\max}}\right)_{\text{short cylinder}} &= \left(\frac{Q}{Q_{\max}}\right)_{\text{plane wall}} + \left(\frac{Q}{Q_{\max}}\right)_{\text{long cylinder}} \left[1 - \left(\frac{Q}{Q_{\max}}\right)_{\text{plane wall}} \right] \\ &= 0.04477 + (0.2156)(1 - 0.04477) = 0.2507 \end{aligned}$$

Then the total heat transfer from the short cylinder as it is cooled from 300°C at the center to 20°C becomes

$$Q = 0.2507 Q_{\max} = (0.2507)(10,100 \text{ kJ}) = \mathbf{2530 \text{ kJ}}$$

which is identical to the heat transfer to the cylinder as the cylinder at 20°C is heated to 300°C at the center.

11-90 EES Prob. 11-88 is reconsidered. The effect of the final center temperature of the block on the heating time and the amount of heat transfer is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

L=0.20 [m]
 $2r_o=0.15$ [m]
 $T_i=20$ [C]
 $T_{\infty}=1200$ [C]
 $T_{o_o}=300$ [C]
 $h=80$ [W/m²-C]

"PROPERTIES"

$k=236$ [W/m-C]
 $\rho=2702$ [kg/m³]
 $c_p=0.896$ [kJ/kg-C]
 $\alpha=9.75E-5$ [m²/s]

"ANALYSIS"

"This short cylinder can physically be formed by the intersection of a long cylinder of radius r_o and a plane wall of thickness $2L$ "

"For plane wall"

$Bi_w=(hL)/k$

"From Table 11-2 corresponding to this Bi number, we read"

$\lambda_{1w}=0.2568$ "w stands for wall"

$A_{1w}=1.0110$

$\tau_w=(\alpha \cdot \text{time})/L^2$

$\theta_{ow}=A_{1w} \cdot \exp(-\lambda_{1w}^2 \tau_w)$ " $\theta_{ow}=(T_{ow}-T_{\infty})/(T_i-T_{\infty})$ "

"For long cylinder"

$Bi_c=(hr_o)/k$ "c stands for cylinder"

"From Table 11-2 corresponding to this Bi number, we read"

$\lambda_{1c}=0.2217$

$A_{1c}=1.0063$

$\tau_c=(\alpha \cdot \text{time})/r_o^2$

$\theta_{oc}=A_{1c} \cdot \exp(-\lambda_{1c}^2 \tau_c)$ " $\theta_{oc}=(T_{oc}-T_{\infty})/(T_i-T_{\infty})$ "

$(T_{oo}-T_{\infty})/(T_i-T_{\infty})=\theta_{ow} \cdot \theta_{oc}$ "center temperature of cylinder"

$V=\pi r_o^2 L$

$m=\rho V$

$Q_{\max}=m \cdot c_p \cdot (T_{\infty}-T_i)$

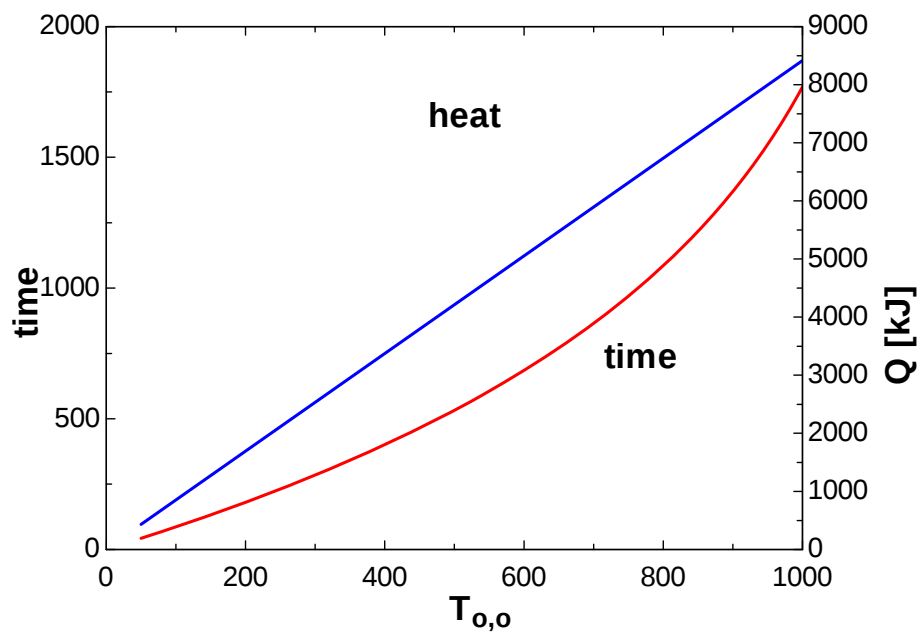
$Q_w=1-\theta_{ow} \cdot \sin(\lambda_{1w})/\lambda_{1w}$ " $Q_w=(Q/Q_{\max})_w$ "

$Q_c=1-2 \cdot \theta_{oc} \cdot J_1/\lambda_{1c}$ " $Q_c=(Q/Q_{\max})_c$ "

$J_1=0.1101$ "From Table 11-3, at λ_{1c} "

$Q/Q_{\max}=Q_w+Q_c \cdot (1-Q_w)$ "total heat transfer"

$T_{o,o}$ [C]	time [s]	Q [kJ]
50	42.43	430.3
100	86.33	850.6
150	132.3	1271
200	180.4	1691
250	231.1	2111
300	284.5	2532
350	340.9	2952
400	400.8	3372
450	464.5	3793
500	532.6	4213
550	605.8	4633
600	684.9	5053
650	770.8	5474
700	864.9	5894
750	968.9	6314
800	1085	6734
850	1217	7155
900	1369	7575
950	1549	7995
1000	1770	8416



Review Problems

11-91 Two large steel plates are stuck together because of the freezing of the water between the two plates. Hot air is blown over the exposed surface of the plate on the top to melt the ice. The length of time the hot air should be blown is to be determined.

Assumptions 1 Heat conduction in the plates is one-dimensional since the plate is large relative to its thickness and there is thermal symmetry about the center plane. 3 The thermal properties of the steel plates are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of steel plates are given to be $k = 43 \text{ W/m}\cdot^\circ\text{C}$ and $\alpha = 1.17 \times 10^{-5} \text{ m}^2/\text{s}$

Analysis The characteristic length of the plates and the Biot number are

$$L_c = \frac{V}{A_s} = L = 0.02 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(40 \text{ W/m}^2\cdot^\circ\text{C})(0.02 \text{ m})}{(43 \text{ W/m}\cdot^\circ\text{C})} = 0.019 < 0.1$$

Since $Bi < 0.1$, the lumped system analysis is applicable. Therefore,

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{40 \text{ W/m}^2\cdot^\circ\text{C}}{(3.675 \times 10^6 \text{ J/m}^3\cdot^\circ\text{C})(0.02 \text{ m})} = 0.000544 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{0 - 50}{-15 - 50} = e^{-(0.000544 \text{ s}^{-1})t} \longrightarrow t = 482 \text{ s} = 8.0 \text{ min}$$

where $\rho c_p = \frac{k}{\alpha} = \frac{43 \text{ W/m}\cdot^\circ\text{C}}{1.17 \times 10^{-5} \text{ m}^2/\text{s}} = 3.675 \times 10^6 \text{ J/m}^3\cdot^\circ\text{C}$

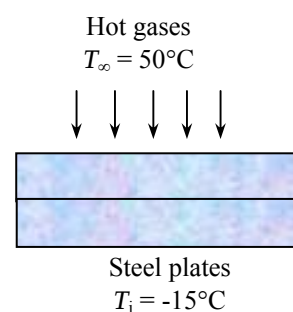
Alternative solution: This problem can also be solved using the transient chart Fig. 11-15a,

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{1}{0.019} = 52.6 \\ \frac{T_0 - T_\infty}{T_i - T_\infty} &= \frac{0 - 50}{-15 - 50} = 0.769 \end{aligned} \right\} \tau = \frac{\alpha t}{r_o^2} = 15 > 0.2$$

Then,

$$t = \frac{\tau r_o^2}{\alpha} = \frac{(15)(0.02 \text{ m})^2}{(1.17 \times 10^{-5} \text{ m}^2/\text{s})} = 513 \text{ s}$$

The difference is due to the reading error of the chart.



11-92 A curing kiln is heated by injecting steam into it and raising its inner surface temperature to a specified value. It is to be determined whether the temperature at the outer surfaces of the kiln changes during the curing period.

Assumptions 1 The temperature in the wall is affected by the thermal conditions at inner surfaces only and the convection heat transfer coefficient inside is very large. Therefore, the wall can be considered to be a semi-infinite medium with a specified surface temperature of 45°C. **2** The thermal properties of the concrete wall are constant.

Properties The thermal properties of the concrete wall are given to be $k = 0.9 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.23 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis We determine the temperature at a depth of $x = 0.3 \text{ m}$ in 2.5 h using the analytical solution,

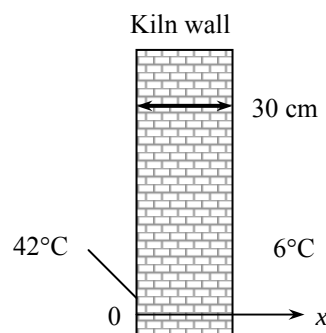
$$\frac{T(x, t) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Substituting,

$$\frac{T(x, t) - 6}{42 - 6} = \text{erfc}\left(\frac{0.3 \text{ m}}{2\sqrt{(0.23 \times 10^{-5} \text{ m}^2/\text{s})(2.5 \text{ h} \times 3600 \text{ s/h})}}\right)$$

$$= \text{erfc}(1.043) = 0.1402$$

$$T(x, t) = 11.0^\circ\text{C}$$



which is greater than the initial temperature of 6°C. Therefore, heat will propagate through the 0.3 m thick wall in 2.5 h, and thus it may be desirable to insulate the outer surface of the wall to save energy.

11-93 The water pipes are buried in the ground to prevent freezing. The minimum burial depth at a particular location is to be determined.

Assumptions 1 The temperature in the soil is affected by the thermal conditions at one surface only, and thus the soil can be considered to be a semi-infinite medium with a specified surface temperature of -10°C . **2** The thermal properties of the soil are constant.

Properties The thermal properties of the soil are given to be $k = 0.7 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.4 \times 10^{-5} \text{ m}^2/\text{s}$.

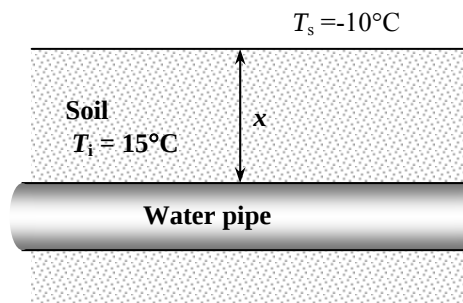
Analysis The depth at which the temperature drops to 0°C in 75 days is determined using the analytical solution,

$$\frac{T(x, t) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Substituting and using Table 11-4, we obtain

$$\frac{0 - 15}{-10 - 15} = \text{erfc}\left(\frac{x}{2\sqrt{(1.4 \times 10^{-5} \text{ m}^2/\text{s})(75 \text{ day} \times 24 \text{ h/day} \times 3600 \text{ s/h})}}\right)$$

$$\longrightarrow x = 7.05 \text{ m}$$



Therefore, the pipes must be buried at a depth of at least 7.05 m.

11-94 A hot dog is to be cooked by dropping it into boiling water. The time of cooking is to be determined.

Assumptions 1 Heat conduction in the hot dog is two-dimensional, and thus the temperature varies in both the axial x - and the radial r - directions. 2 The thermal properties of the hot dog are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the hot dog are given to be $k = 0.76 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 980 \text{ kg/m}^3$, $c_p = 3.9 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 2 \times 10^{-7} \text{ m}^2/\text{s}$.

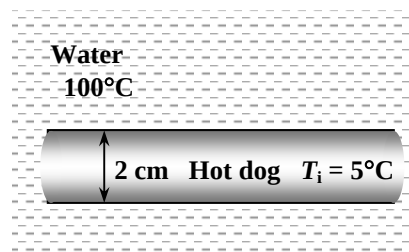
Analysis This hot dog can physically be formed by the intersection of an infinite plane wall of thickness $2L = 12 \text{ cm}$, and a long cylinder of radius $r_o = D/2 = 1 \text{ cm}$. The Biot numbers and corresponding constants are first determined to be

$$Bi = \frac{hL}{k} = \frac{(600 \text{ W/m}^2 \cdot ^\circ\text{C})(0.06 \text{ m})}{(0.76 \text{ W/m} \cdot ^\circ\text{C})} = 47.37 \longrightarrow \lambda_1 = 1.5380 \text{ and } A_1 = 1.2726$$

$$Bi = \frac{hr_o}{k} = \frac{(600 \text{ W/m}^2 \cdot ^\circ\text{C})(0.01 \text{ m})}{(0.76 \text{ W/m} \cdot ^\circ\text{C})} = 7.895 \longrightarrow \lambda_1 = 2.1249 \text{ and } A_1 = 1.5514$$

Noting that $\tau = \alpha t / L^2$ and assuming $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable, the product solution for this problem can be written as

$$\begin{aligned} \theta(0,0,t)_{block} &= \theta(0,t)_{wall} \theta(0,t)_{cyl} = \left(A_1 e^{-\lambda_1^2 \tau} \right) \left(A_1 e^{-\lambda_1^2 \tau} \right) \\ \frac{80-100}{5-100} &= \left\{ (1.2726) \exp \left[-(1.5380)^2 \frac{(2 \times 10^{-7})t}{(0.06)^2} \right] \right\} \\ &\quad \times \left\{ (1.5514) \exp \left[-(2.1249)^2 \frac{(2 \times 10^{-7})t}{(0.01)^2} \right] \right\} = 0.2105 \end{aligned}$$



which gives

$$t = 244 \text{ s} = 4.1 \text{ min}$$

Therefore, it will take about 4.1 min for the hot dog to cook. Note that

$$\tau_{cyl} = \frac{\alpha t}{r_o^2} = \frac{(2 \times 10^{-7} \text{ m}^2/\text{s})(244 \text{ s})}{(0.01 \text{ m})^2} = 0.49 > 0.2$$

and thus the assumption $\tau > 0.2$ for the applicability of the one-term approximate solution is verified.

Discussion This problem could also be solved by treating the hot dog as an infinite cylinder since heat transfer through the end surfaces will have little effect on the mid section temperature because of the large distance.

11-95 A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The temperature of the sheet metal after quenching and the rate at which heat needs to be removed from the oil in order to keep its temperature constant are to be determined.

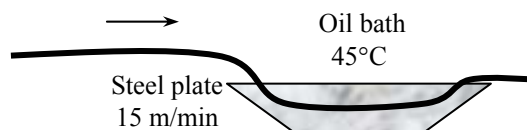
Assumptions 1 The thermal properties of the steel plate are constant. 2 The heat transfer coefficient is constant and uniform over the entire surface. 3 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be checked).

Properties The properties of the steel plate are $k = 60.5 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 7854 \text{ kg/m}^3$, and $c_p = 434 \text{ J/kg} \cdot ^\circ\text{C}$ (Table A-24).

Analysis The characteristic length of the steel plate and the Biot number are

$$L_c = \frac{V}{A_s} = L = 0.0025 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(860 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0025 \text{ m})}{60.5 \text{ W/m} \cdot ^\circ\text{C}} = 0.036 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Therefore,

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{860 \text{ W/m}^2 \cdot ^\circ\text{C}}{(7854 \text{ kg/m}^3)(434 \text{ J/kg} \cdot ^\circ\text{C})(0.0025 \text{ m})} = 0.10092 \text{ s}^{-1}$$

$$\text{time} = \frac{\text{length}}{\text{velocity}} = \frac{9 \text{ m}}{15 \text{ m/min}} = 0.6 \text{ min} = 36 \text{ s}$$

Then the temperature of the sheet metal when it leaves the oil bath is determined to be

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{T(t) - 45}{820 - 45} = e^{-(0.10092 \text{ s}^{-1})(36 \text{ s})} \longrightarrow T(t) = \mathbf{65.5^\circ\text{C}}$$

The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho \dot{V} = \rho w t V = (7854 \text{ kg/m}^3)(2 \text{ m})(0.005 \text{ m})(15 \text{ m/min}) = 1178 \text{ kg/min}$$

Then the rate of heat transfer from the sheet metal to the oil bath and thus the rate at which heat needs to be removed from the oil in order to keep its temperature constant at 45°C becomes

$$\dot{Q} = \dot{m} c_p [T_i - T(t)] = (1178 \text{ kg/min})(0.434 \text{ kJ/kg} \cdot ^\circ\text{C})(820 - 65.5)^\circ\text{C} = 385,740 \text{ kJ/min} = \mathbf{6429 \text{ kW}}$$

11-96E A stuffed turkey is cooked in an oven. The average heat transfer coefficient at the surface of the turkey, the temperature of the skin of the turkey in the oven and the total amount of heat transferred to the turkey in the oven are to be determined.

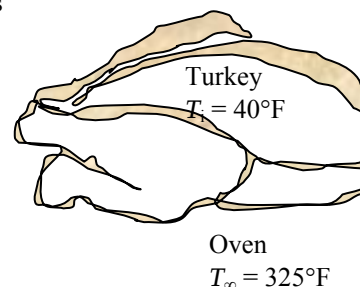
Assumptions 1 The turkey is a homogeneous spherical object. 2 Heat conduction in the turkey is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the turkey are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions are applicable (this assumption will be verified).

Properties The properties of the turkey are given to be $k = 0.26 \text{ Btu/h.ft.}^\circ\text{F}$, $\rho = 75 \text{ lbm/ft}^3$, $c_p = 0.98 \text{ Btu/lbm.}^\circ\text{F}$, and $\alpha = 0.0035 \text{ ft}^2/\text{h}$.

Analysis (a) Assuming the turkey to be spherical in shape, its radius is determined to be

$$m = \rho V \longrightarrow V = \frac{m}{\rho} = \frac{14 \text{ lbm}}{75 \text{ lbm/ft}^3} = 0.1867 \text{ ft}^3$$

$$V = \frac{4}{3}\pi r_o^3 \longrightarrow r_o = \sqrt[3]{\frac{3V}{4\pi}} = \sqrt[3]{\frac{3(0.1867 \text{ ft}^3)}{4\pi}} = 0.3545 \text{ ft}$$



The Fourier number is $\tau = \frac{\alpha t}{r_o^2} = \frac{(3.5 \times 10^{-3} \text{ ft}^2/\text{h})(5 \text{ h})}{(0.3545 \text{ ft})^2} = 0.1392$

which is close to 0.2 but a little below it. Therefore, assuming the one-term approximate solution for transient heat conduction to be applicable, the one-term solution formulation at one-third the radius from the center of the turkey can be expressed as

$$\theta(x, t)_{sph} = \frac{T(x, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_o)}{\lambda_1 r / r_o}$$

$$\frac{185 - 325}{40 - 325} = 0.491 = A_1 e^{-\lambda_1^2 (0.14)} \frac{\sin(0.333 \lambda_1)}{0.333 \lambda_1}$$

By trial and error, it is determined from Table 11-2 that the equation above is satisfied when $Bi = 20$ corresponding to $\lambda_1 = 2.9857$ and $A_1 = 1.9781$. Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.26 \text{ Btu/h.ft.}^\circ\text{F})(20)}{(0.3545 \text{ ft})} = \mathbf{14.7 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}}$$

(b) The temperature at the surface of the turkey is

$$\frac{T(r_o, t) - 325}{40 - 325} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.9781) e^{-(2.9857)^2 (0.14)} \frac{\sin(2.9857)}{2.9857} = 0.02953$$

$$\longrightarrow T(r_o, t) = \mathbf{317^\circ\text{F}}$$

(c) The maximum possible heat transfer is

$$Q_{\max} = mc_p (T_\infty - T_i) = (14 \text{ lbm})(0.98 \text{ Btu/lbm.}^\circ\text{F})(325 - 40)^\circ\text{F} = 3910 \text{ Btu}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{o, sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3} = 1 - 3(0.491) \frac{\sin(2.9857) - (2.9857) \cos(2.9857)}{(2.9857)^3} = 0.828$$

$$Q = 0.828 Q_{\max} = (0.828)(3910 \text{ Btu}) = \mathbf{3240 \text{ Btu}}$$

Discussion The temperature of the outer parts of the turkey will be greater than that of the inner parts when the turkey is taken out of the oven. Then heat will continue to be transferred from the outer parts of the turkey to the inner as a result of temperature difference. Therefore, after 5 minutes, the thermometer reading will probably be more than 185°F .

11-97 CD EES The trunks of some dry oak trees are exposed to hot gases. The time for the ignition of the trunks is to be determined.

Assumptions 1 Heat conduction in the trunks is one-dimensional since it is long and it has thermal symmetry about the center line. 2 The thermal properties of the trunks are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the trunks are given to be $k = 0.17 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.28 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis We treat the trunks of the trees as an infinite cylinder since heat transfer is primarily in the radial direction. Then the Biot number becomes

$$Bi = \frac{hr_o}{k} = \frac{(65 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1 \text{ m})}{(0.17 \text{ W/m} \cdot ^\circ\text{C})} = 38.24$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 2.3420 \quad \text{and} \quad A_1 = 1.5989$$

The Fourier number is

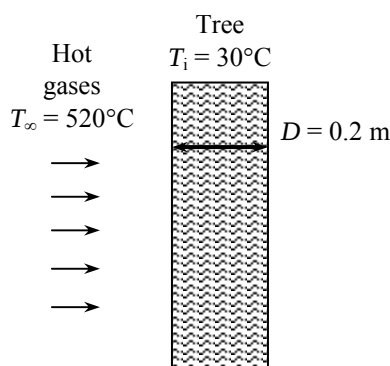
$$\tau = \frac{\alpha t}{r_o^2} = \frac{(1.28 \times 10^{-7} \text{ m}^2/\text{s})(4 \text{ h} \times 3600 \text{ s/h})}{(0.1 \text{ m})^2} = 0.184$$

which is slightly below 0.2 but close to it. Therefore, assuming the one-term approximate solution for transient heat conduction to be applicable, the temperature at the surface of the trees in 4 h becomes

$$\theta(r_o, t)_{cyl} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r / r_o)$$

$$\frac{T(r_o, t) - 520}{30 - 520} = (1.5989) e^{-(2.3420)^2 (0.184)} (0.0332) = 0.01935 \longrightarrow T(r_o, t) = 511^\circ\text{C} > 410^\circ\text{C}$$

Therefore, the trees will ignite. (Note: J_0 is read from Table 11-3).



11-98 A spherical watermelon that is cut into two equal parts is put into a freezer. The time it will take for the center of the exposed cut surface to cool from 25 to 3°C is to be determined.

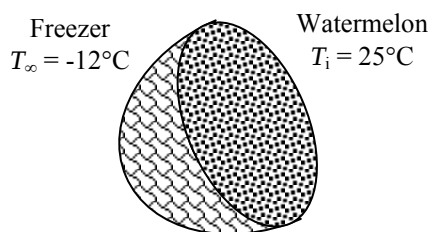
Assumptions 1 The temperature of the exposed surfaces of the watermelon is affected by the convection heat transfer at those surfaces only. Therefore, the watermelon can be considered to be a semi-infinite medium 2 The thermal properties of the watermelon are constant.

Properties The thermal properties of the water is closely approximated by those of water at room temperature, $k = 0.607 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = k / \rho c_p = 0.146 \times 10^{-6} \text{ m}^2/\text{s}$ (Table A-15).

Analysis We use the transient chart in Fig. 11-29 in this case for convenience (instead of the analytic solution),

$$\left. \begin{aligned} 1 - \frac{T(x, t) - T_\infty}{T_i - T_\infty} &= 1 - \frac{3 - (-12)}{25 - (-12)} = 0.595 \\ \xi &= \frac{x}{2\sqrt{\alpha t}} = 0 \end{aligned} \right\} \frac{h\sqrt{\alpha t}}{k} = 1$$

$$\text{Therefore, } t = \frac{(1)^2 k^2}{h^2 \alpha} = \frac{(0.607 \text{ W/m} \cdot ^\circ\text{C})^2}{(22 \text{ W/m}^2 \cdot ^\circ\text{C})^2 (0.146 \times 10^{-6} \text{ m}^2/\text{s})} = 5214 \text{ s} = 86.9 \text{ min}$$



11-99 A cylindrical rod is dropped into boiling water. The thermal diffusivity and the thermal conductivity of the rod are to be determined.

Assumptions 1 Heat conduction in the rod is one-dimensional since the rod is sufficiently long, and thus temperature varies in the radial direction only. 2 The thermal properties of the rod are constant.

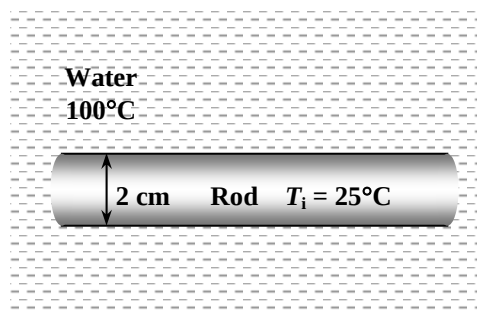
Properties The thermal properties of the rod available are given to be $\rho = 3700 \text{ kg/m}^3$ and $C_p = 920 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis From Fig. 11-16b we have

$$\left. \begin{aligned} \frac{T - T_\infty}{T_0 - T_\infty} &= \frac{93 - 100}{75 - 100} = 0.28 \\ \frac{x}{r_o} &= \frac{r_o}{r_o} = 1 \end{aligned} \right\} \frac{1}{Bi} = \frac{k}{hr_o} = 0.25$$

From Fig. 11-16a we have

$$\left. \begin{aligned} \frac{1}{Bi} &= \frac{k}{hr_o} = 0.25 \\ \frac{T_o - T_\infty}{T_i - T_\infty} &= \frac{75 - 100}{25 - 100} = 0.33 \end{aligned} \right\} \tau = \frac{\alpha t}{r_o^2} = 0.40$$



Then the thermal diffusivity and the thermal conductivity of the material become

$$\alpha = \frac{0.40 r_o^2}{t} = \frac{(0.40)(0.01 \text{ m})^2}{3 \text{ min} \times 60 \text{ s/min}} = 2.22 \times 10^{-7} \text{ m}^2/\text{s}$$

$$\alpha = \frac{k}{\rho C_p} \longrightarrow k = \alpha \rho C_p = (2.22 \times 10^{-7} \text{ m}^2/\text{s})(3700 \text{ kg/m}^3)(920 \text{ J/kg} \cdot ^\circ\text{C}) = 0.756 \text{ W/m} \cdot ^\circ\text{C}$$

11-100 The time it will take for the diameter of a raindrop to reduce to a certain value as it falls through ambient air is to be determined.

Assumptions 1 The water temperature remains constant. 2 The thermal properties of the water are constant.

Properties The density and heat of vaporization of the water are $\rho = 1000 \text{ kg/m}^3$ and $h_{fg} = 2490 \text{ kJ/kg}$ (Table A-15).

Analysis The initial and final masses of the raindrop are

$$m_i = \rho V_i = \rho \frac{4}{3} \pi r_i^3 = (1000 \text{ kg/m}^3) \frac{4}{3} \pi (0.0025 \text{ m})^3 = 0.0000654 \text{ kg}$$

$$m_f = \rho V_f = \rho \frac{4}{3} \pi r_f^3 = (1000 \text{ kg/m}^3) \frac{4}{3} \pi (0.0015 \text{ m})^3 = 0.0000141 \text{ kg}$$

whose difference is

$$m = m_i - m_f = 0.0000654 - 0.0000141 = 0.0000513 \text{ kg}$$

The amount of heat transfer required to cause this much evaporation is

$$Q = (0.0000513 \text{ kg})(2490 \text{ kJ/kg}) = 0.1278 \text{ kJ}$$

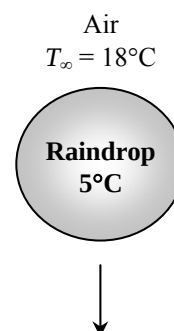
The average heat transfer surface area and the rate of heat transfer are

$$A_s = \frac{4\pi(r_i^2 + r_f^2)}{2} = \frac{4\pi[(0.0025 \text{ m})^2 + (0.0015 \text{ m})^2]}{2} = 5.341 \times 10^{-5} \text{ m}^2$$

$$\dot{Q} = h A_s (T_i - T_\infty) = (400 \text{ W/m}^2 \cdot ^\circ\text{C})(5.341 \times 10^{-5} \text{ m}^2)(18 - 5)^\circ\text{C} = 0.2777 \text{ J/s}$$

Then the time required for the raindrop to experience this reduction in size becomes

$$\dot{Q} = \frac{Q}{\Delta t} \longrightarrow \Delta t = \frac{Q}{\dot{Q}} = \frac{127.8 \text{ J}}{0.2777 \text{ J/s}} = 460 \text{ s} = 7.7 \text{ min}$$



11-101E A plate, a long cylinder, and a sphere are exposed to cool air. The center temperature of each geometry is to be determined.

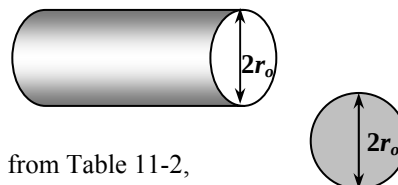
Assumptions 1 Heat conduction in each geometry is one-dimensional. 2 The thermal properties of the bodies are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of bronze are given to be $k = 15 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ and $\alpha = 0.333 \text{ ft}^2/\text{h}$.

Analysis After 5 minutes

Plate: First the Biot number is calculated to be

$$Bi = \frac{hL}{k} = \frac{(7 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.5/12 \text{ ft})}{(15 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 0.01944$$

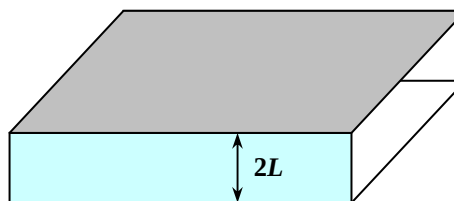


The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.1387 \quad \text{and} \quad A_1 = 1.0032$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.333 \text{ ft}^2/\text{h})(5 \text{ min}/60 \text{ min/h})}{(0.5/12 \text{ ft})^2} = 15.98 > 0.2$$



Then the center temperature of the plate becomes

$$\theta_{o,\text{wall}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_o - 75}{400 - 75} = (1.0032)e^{-(0.1387)^2(15.98)} = 0.738 \longrightarrow T_o = 315^\circ\text{F}$$

Cylinder:

$$Bi = 0.01944 \xrightarrow{\text{Table 4-2}} \lambda_1 = 0.1962 \quad \text{and} \quad A_1 = 1.0049$$

$$\theta_{o,\text{cyl}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_o - 75}{400 - 75} = (1.0049)e^{-(0.1962)^2(15.98)} = 0.543 \longrightarrow T_o = 252^\circ\text{F}$$

Sphere:

$$Bi = 0.01944 \xrightarrow{\text{Table 4-2}} \lambda_1 = 0.2405 \quad \text{and} \quad A_1 = 1.0058$$

$$\theta_{o,\text{sph}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_o - 75}{400 - 75} = (1.0058)e^{-(0.2405)^2(15.98)} = 0.399 \longrightarrow T_o = 205^\circ\text{F}$$

After 10 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.333 \text{ ft}^2/\text{h})(10 \text{ min}/60 \text{ min/h})}{(0.5/12 \text{ ft})^2} = 31.97 > 0.2$$

Plate:

$$\theta_{o,\text{wall}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_o - 75}{400 - 75} = (1.0032)e^{-(0.1387)^2(31.97)} = 0.542 \longrightarrow T_o = 251^\circ\text{F}$$

Cylinder:

$$\theta_{o,\text{cyl}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_o - 75}{400 - 75} = (1.0049)e^{-(0.1962)^2(31.97)} = 0.293 \longrightarrow T_o = 170^\circ\text{F}$$

Sphere:

$$\theta_{o,\text{sph}} = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_o - 75}{400 - 75} = (1.0058)e^{-(0.2405)^2(31.97)} = 0.158 \longrightarrow T_o = 126^\circ\text{F}$$

After 30 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.333 \text{ ft}^2/\text{h})(30 \text{ min}/60 \text{ min/h})}{(0.5/12 \text{ ft})^2} = 95.9 > 0.2$$

Plate:

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0032)e^{-(0.1387)^2 (95.9)} = 0.159 \longrightarrow T_0 = \mathbf{127^\circ\text{F}}$$

Cylinder:

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0049)e^{-(0.1962)^2 (95.9)} = 0.025 \longrightarrow T_0 = \mathbf{83^\circ\text{F}}$$

Sphere:

$$\theta_{0,\text{sph}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0058)e^{-(0.2405)^2 (95.9)} = 0.00392 \longrightarrow T_0 = \mathbf{76^\circ\text{F}}$$

The sphere has the largest surface area through which heat is transferred per unit volume, and thus the highest rate of heat transfer. Consequently, the center temperature of the sphere is always the lowest.

11-102E A plate, a long cylinder, and a sphere are exposed to cool air. The center temperature of each geometry is to be determined.

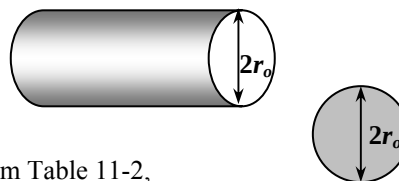
Assumptions 1 Heat conduction in each geometry is one-dimensional. 2 The thermal properties of the geometries are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of cast iron are given to be $k = 29 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ and $\alpha = 0.61 \text{ ft}^2/\text{h}$.

Analysis After 5 minutes

Plate: First the Biot number is calculated to be

$$Bi = \frac{hL}{k} = \frac{(7 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.5/12 \text{ ft})}{(29 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})} = 0.01006 \cong 0.01$$

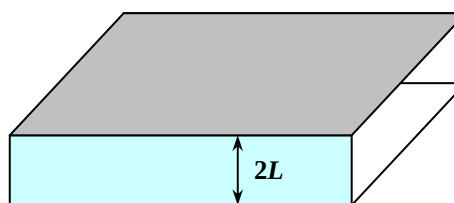


The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 0.0998 \quad \text{and} \quad A_1 = 1.0017$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.61 \text{ ft}^2/\text{h})(5 \text{ min}/60 \text{ min/h})}{(0.5/12 \text{ ft})^2} = 29.28 > 0.2$$



Then the center temperature of the plate becomes

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0017)e^{-(0.0998)^2 (29.28)} = 0.748 \longrightarrow T_0 = 318^\circ\text{F}$$

Cylinder:

$$Bi = 0.01 \xrightarrow{\text{Table 4-2}} \lambda_1 = 0.1412 \quad \text{and} \quad A_1 = 1.0025$$

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0025)e^{-(0.1412)^2 (29.28)} = 0.559 \longrightarrow T_0 = 257^\circ\text{F}$$

Sphere:

$$Bi = 0.01 \xrightarrow{\text{Table 4-2}} \lambda_1 = 0.1730 \quad \text{and} \quad A_1 = 1.0030$$

$$\theta_{0,\text{sph}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0030)e^{-(0.1730)^2 (29.28)} = 0.418 \longrightarrow T_0 = 211^\circ\text{F}$$

After 10 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.61 \text{ ft}^2/\text{h})(10 \text{ min}/60 \text{ min/h})}{(0.5/12 \text{ ft})^2} = 58.56 > 0.2$$

Plate:

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0017)e^{-(0.0998)^2 (58.56)} = 0.559 \longrightarrow T_0 = 257^\circ\text{F}$$

Cylinder:

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0025)e^{-(0.1412)^2 (58.56)} = 0.312 \longrightarrow T_0 = 176^\circ\text{F}$$

Sphere:

$$\theta_{0,\text{sph}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0030)e^{-(0.1730)^2 (58.56)} = 0.174 \longrightarrow T_0 = 132^\circ\text{F}$$

After 30 minutes

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.61 \text{ ft}^2/\text{h})(30 \text{ min}/60 \text{ min/h})}{(0.5/12 \text{ ft})^2} = 175.68 > 0.2$$

Plate:

$$\theta_{0,\text{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0017)e^{-(0.0998)^2(175.68)} = 0.174 \longrightarrow T_0 = \mathbf{132^\circ\text{F}}$$

Cylinder:

$$\theta_{0,\text{cyl}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0025)e^{-(0.1412)^2(175.68)} = 0.030 \longrightarrow T_0 = \mathbf{84.8^\circ\text{F}}$$

Sphere:

$$\theta_{0,\text{sph}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{T_0 - 75}{400 - 75} = (1.0030)e^{-(0.1730)^2(175.68)} = 0.0052 \longrightarrow T_0 = \mathbf{76.7^\circ\text{F}}$$

The sphere has the largest surface area through which heat is transferred per unit volume, and thus the highest rate of heat transfer. Consequently, the center temperature of the sphere is always the lowest.

11-103E EES Prob. 11-101E is reconsidered. The center temperature of each geometry as a function of the cooling time is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$2*L=(1/12)$ [ft]
 $2*r_{o,c}=(1/12)$ [ft] "c stands for cylinder"
 $2*r_{o,s}=(1/12)$ [ft] "s stands for sphere"
 $T_i=400$ [F]
 $T_{infinity}=75$ [F]
 $h=7$ [Btu/h-ft²-F]
 $time=5$ [min]

"PROPERTIES"

$k=15$ [Btu/h-ft-F]
 $alpha=0.333$ [ft²/h]*Convert(ft²/h, ft²/min)

"ANALYSIS"

"For plane wall"

$Bi_w=(h*L)/k$
 "From Table 11-2 corresponding to this Bi number, we read"
 $lambda_{1,w}=0.1387$
 $A_{1,w}=1.0032$
 $tau_w=(alpha*time)/L^2$
 $(T_{o,w}-T_{infinity})/(T_i-T_{infinity})=A_{1,w}*exp(-lambda_{1,w}^2*tau_w)$

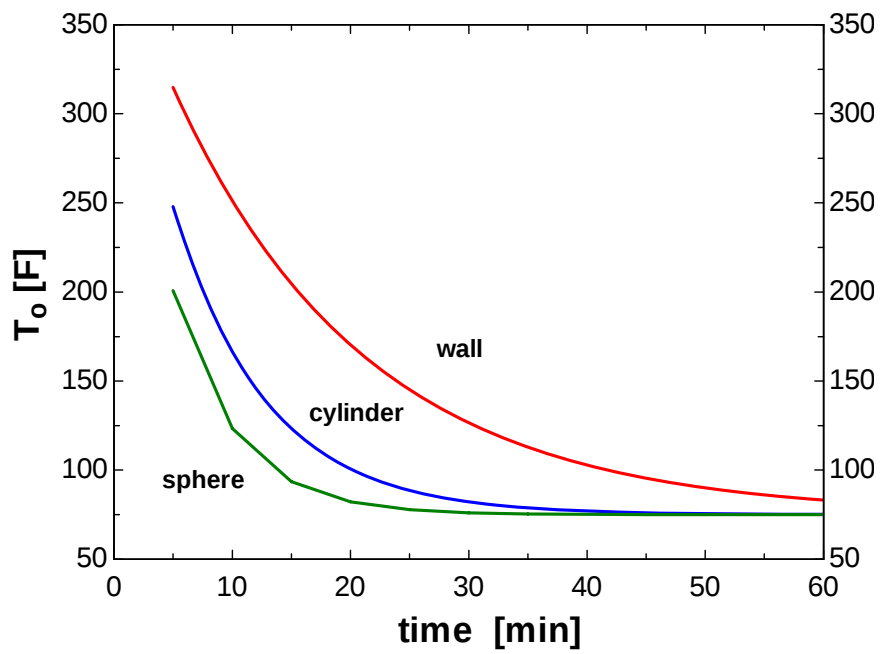
"For long cylinder"

$Bi_c=(h*r_{o,c})/k$
 "From Table 11-2 corresponding to this Bi number, we read"
 $lambda_{1,c}=0.1962$
 $A_{1,c}=1.0049$
 $tau_c=(alpha*time)/r_{o,c}^2$
 $(T_{o,c}-T_{infinity})/(T_i-T_{infinity})=A_{1,c}*exp(-lambda_{1,c}^2*tau_c)$

"For sphere"

$Bi_s=(h*r_{o,s})/k$
 "From Table 11-2 corresponding to this Bi number, we read"
 $lambda_{1,s}=0.2405$
 $A_{1,s}=1.0058$
 $tau_s=(alpha*time)/r_{o,s}^2$
 $(T_{o,s}-T_{infinity})/(T_i-T_{infinity})=A_{1,s}*exp(-lambda_{1,s}^2*tau_s)$

time [min]	$T_{o,w}$ [F]	$T_{o,c}$ [F]	$T_{o,s}$ [F]
5	314.7	251.5	204.7
10	251.3	170.4	126.4
15	204.6	126.6	95.41
20	170.3	102.9	83.1
25	145.1	90.06	78.21
30	126.5	83.14	76.27
35	112.9	79.4	75.51
40	102.9	77.38	75.2
45	95.48	76.29	75.08
50	90.06	75.69	75.03
55	86.07	75.38	75.01
60	83.14	75.2	75



11-104 Internal combustion engine valves are quenched in a large oil bath. The time it takes for the valve temperature to drop to specified temperatures and the maximum heat transfer are to be determined.

Assumptions 1 The thermal properties of the valves are constant. 2 The heat transfer coefficient is constant and uniform over the entire surface. 3 Depending on the size of the oil bath, the oil bath temperature will increase during quenching. However, an average constant temperature as specified in the problem will be used. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

Properties The thermal conductivity, density, and specific heat of the balls are given to be $k = 48$ W/m·°C, $\rho = 7840$ kg/m³, and $c_p = 440$ J/kg·°C.

Analysis (a) The characteristic length of the balls and the Biot number are

$$L_c = \frac{V}{A_s} = \frac{1.8(\pi D^2 L / 4)}{2\pi DL} = \frac{1.8D}{8} = \frac{1.8(0.008 \text{ m})}{8} = 0.0018 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(800 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0018 \text{ m})}{48 \text{ W/m} \cdot ^\circ\text{C}} = 0.03 < 0.1$$

Therefore, we can use lumped system analysis. Then the time for a final valve temperature of 400°C becomes

$$b = \frac{hA_s}{\rho c_p V} = \frac{8h}{1.8\rho c_p D} = \frac{8(800 \text{ W/m}^2 \cdot ^\circ\text{C})}{1.8(7840 \text{ kg/m}^3)(440 \text{ J/kg} \cdot ^\circ\text{C})(0.008 \text{ m})} = 0.1288 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{400 - 50}{800 - 50} = e^{-(0.1288 \text{ s}^{-1})t} \longrightarrow t = \mathbf{5.9 \text{ s}}$$

(b) The time for a final valve temperature of 200°C is

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{200 - 50}{800 - 50} = e^{-(0.1288 \text{ s}^{-1})t} \longrightarrow t = \mathbf{12.5 \text{ s}}$$

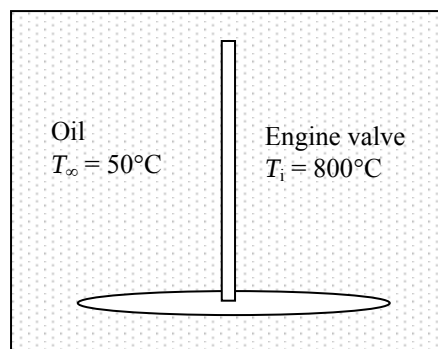
(c) The time for a final valve temperature of 51°C is

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{51 - 50}{800 - 50} = e^{-(0.1288 \text{ s}^{-1})t} \longrightarrow t = \mathbf{51.4 \text{ s}}$$

(d) The maximum amount of heat transfer from a single valve is determined from

$$m = \rho V = \rho \frac{1.8\pi D^2 L}{4} = (7840 \text{ kg/m}^3) \frac{1.8\pi(0.008 \text{ m})^2(0.10 \text{ m})}{4} = 0.0709 \text{ kg}$$

$$Q = mc_p[T_f - T_i] = (0.0709 \text{ kg})(440 \text{ J/kg} \cdot ^\circ\text{C})(800 - 50)^\circ\text{C} = 23,400 \text{ J} = \mathbf{23.4 \text{ kJ}} \text{ (per valve)}$$



11-105 A watermelon is placed into a lake to cool it. The heat transfer coefficient at the surface of the watermelon and the temperature of the outer surface of the watermelon are to be determined.

Assumptions 1 The watermelon is a homogeneous spherical object. 2 Heat conduction in the watermelon is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the watermelon are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

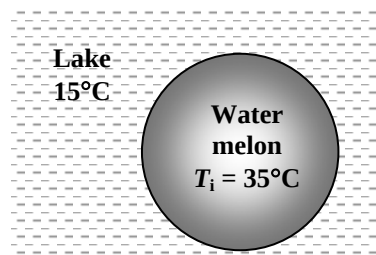
Properties The properties of the watermelon are given to be $k = 0.618 \text{ W/m} \cdot ^\circ\text{C}$, $\alpha = 0.15 \times 10^{-6} \text{ m}^2/\text{s}$, $\rho = 995 \text{ kg/m}^3$ and $c_p = 4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.15 \times 10^{-6} \text{ m}^2/\text{s})[(4 \times 60 + 40 \text{ min}) \times 60 \text{ s/min}]}{(0.10 \text{ m})^2} = 0.252$$

which is greater than 0.2. Then the one-term solution can be written in the form

$$\theta_{0,\text{sph}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \longrightarrow \frac{20 - 15}{35 - 15} = 0.25 = A_1 e^{-\lambda_1^2 (0.252)}$$



It is determined from Table 11-2 by trial and error that this equation is satisfied when $Bi = 10$, which corresponds to $\lambda_1 = 2.8363$ and $A_1 = 1.9249$. Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.618 \text{ W/m} \cdot ^\circ\text{C})(10)}{(0.10 \text{ m})} = \mathbf{61.8 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

The temperature at the surface of the watermelon is

$$\theta(r_o, t)_{\text{sph}} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.9249) e^{-(2.8363)^2 (0.252)} \frac{\sin(2.8363 \text{ rad})}{2.8363}$$

$$\frac{T(r_o, t) - 15}{35 - 15} = 0.0269 \longrightarrow T(r_o, t) = \mathbf{15.5^\circ\text{C}}$$

11-106 Large food slabs are cooled in a refrigeration room. Center temperatures are to be determined for different foods.

Assumptions 1 Heat conduction in the slabs is one-dimensional since the slab is large relative to its thickness and there is thermal symmetry about the center plane. 3 The thermal properties of the slabs are constant. 4 The heat transfer coefficient is constant and uniform over the entire surface. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of foods are given to be $k = 0.233 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.11 \times 10^{-6} \text{ m}^2/\text{s}$ for margarine, $k = 0.082 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.10 \times 10^{-6} \text{ m}^2/\text{s}$ for white cake, and $k = 0.106 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.12 \times 10^{-6} \text{ m}^2/\text{s}$ for chocolate cake.

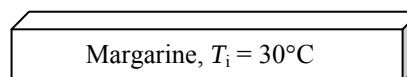
Analysis (a) In the case of margarine, the Biot number is

$$Bi = \frac{hL}{k} = \frac{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(0.05 \text{ m})}{(0.233 \text{ W/m} \cdot ^\circ\text{C})} = 5.365$$

$$\begin{array}{l} \text{Air} \\ T_\infty = 0^\circ\text{C} \end{array}$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.3269 \quad \text{and} \quad A_1 = 1.2431$$



The Fourier number is
$$\tau = \frac{\alpha t}{L^2} = \frac{(0.11 \times 10^{-6} \text{ m}^2/\text{s})(6 \text{ h} \times 3600 \text{ s/h})}{(0.05 \text{ m})^2} = 0.9504 > 0.2$$

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the temperature at the center of the box if the box contains margarine becomes

$$\theta(0, t)_{\text{wall}} = \frac{T(0, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2431)e^{-(1.3269)^2 (0.9504)}$$

$$\frac{T(0, t) - 0}{30 - 0} = 0.233 \longrightarrow T(0, t) = 7.0^\circ\text{C}$$

(b) Repeating the calculations for white cake,

$$Bi = \frac{hL}{k} = \frac{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(0.05 \text{ m})}{(0.082 \text{ W/m} \cdot ^\circ\text{C})} = 15.24 \longrightarrow \lambda_1 = 1.4641 \quad \text{and} \quad A_1 = 1.2661$$

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.10 \times 10^{-6} \text{ m}^2/\text{s})(6 \text{ h} \times 3600 \text{ s/h})}{(0.05 \text{ m})^2} = 0.864 > 0.2$$

$$\theta(0, t)_{\text{wall}} = \frac{T(0, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2661)e^{-(1.4641)^2 (0.864)}$$

$$\frac{T(0, t) - 0}{30 - 0} = 0.199 \longrightarrow T(0, t) = 6.0^\circ\text{C}$$

(c) Repeating the calculations for chocolate cake,

$$Bi = \frac{hL}{k} = \frac{(25 \text{ W/m}^2 \cdot ^\circ\text{C})(0.05 \text{ m})}{(0.106 \text{ W/m} \cdot ^\circ\text{C})} = 11.79 \longrightarrow \lambda_1 = 1.4409 \quad \text{and} \quad A_1 = 1.2634$$

$$\tau = \frac{\alpha t}{L^2} = \frac{(0.12 \times 10^{-6} \text{ m}^2/\text{s})(6 \text{ h} \times 3600 \text{ s/h})}{(0.05 \text{ m})^2} = 1.0368 > 0.2$$

$$\theta(0, t)_{\text{wall}} = \frac{T(0, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.2634)e^{-(1.4409)^2 (1.0368)}$$

$$\frac{T(0, t) - 0}{30 - 0} = 0.147 \longrightarrow T(0, t) = 4.4^\circ\text{C}$$

11-107 A cold cylindrical concrete column is exposed to warm ambient air during the day. The time it will take for the surface temperature to rise to a specified value, the amounts of heat transfer for specified values of center and surface temperatures are to be determined.

Assumptions 1 Heat conduction in the column is one-dimensional since it is long and it has thermal symmetry about the center line. 2 The thermal properties of the column are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of concrete are given to be $k = 0.79 \text{ W/m} \cdot ^\circ\text{C}$, $\alpha = 5.94 \times 10^{-7} \text{ m}^2/\text{s}$, $\rho = 1600 \text{ kg/m}^3$ and $c_p = 0.84 \text{ kJ/kg} \cdot ^\circ\text{C}$

Analysis (a) The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(14 \text{ W/m}^2 \cdot ^\circ\text{C})(0.15 \text{ m})}{(0.79 \text{ W/m} \cdot ^\circ\text{C})} = 2.658$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.7240 \quad \text{and} \quad A_1 = 1.3915$$

Once the constant $J_0 = 0.3841$ is determined from Table 11-3 corresponding to the constant λ_1 , the Fourier number is determined to be

$$\frac{T(r_o, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r_o / r_o) \longrightarrow \frac{27 - 28}{14 - 28} = (1.3915) e^{-(1.7240)^2 \tau} (0.3841) \longrightarrow \tau = 0.6771$$

which is above the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can be used. Then the time it will take for the column surface temperature to rise to 27°C becomes

$$t = \frac{\tau_o^2}{\alpha} = \frac{(0.6771)(0.15 \text{ m})^2}{5.94 \times 10^{-7} \text{ m}^2/\text{s}} = 25,650 \text{ s} = \mathbf{7.1 \text{ hours}}$$

(b) The heat transfer to the column will stop when the center temperature of column reaches to the ambient temperature, which is 28°C . That is, we are asked to determine the maximum heat transfer between the ambient air and the column.

$$m = \rho V = \rho \pi r_o^2 L = (1600 \text{ kg/m}^3)[\pi(0.15 \text{ m})^2(4 \text{ m})] = 452.4 \text{ kg}$$

$$Q_{\max} = mc_p [T_\infty - T_i] = (452.4 \text{ kg})(0.84 \text{ kJ/kg} \cdot ^\circ\text{C})(28 - 14)^\circ\text{C} = \mathbf{5320 \text{ kJ}}$$

(c) To determine the amount of heat transfer until the surface temperature reaches to 27°C , we first determine

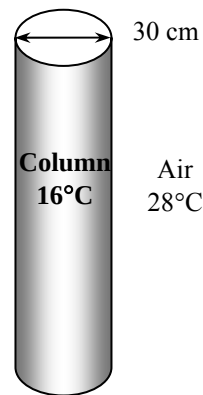
$$\frac{T(0, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.3915) e^{-(1.7240)^2 (0.6771)} = 0.1860$$

Once the constant $J_1 = 0.5787$ is determined from Table 11-3 corresponding to the constant λ_1 , the amount of heat transfer becomes

$$\left(\frac{Q}{Q_{\max}} \right)_{\text{cyl}} = 1 - 2 \left(\frac{T_o - T_\infty}{T_i - T_\infty} \right) \frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2 \times 0.1860 \times \frac{0.5787}{1.7240} = 0.875$$

$$Q = 0.875 Q_{\max}$$

$$Q = 0.875(5320 \text{ kJ}) = \mathbf{4660 \text{ kJ}}$$



11-108 Long aluminum wires are extruded and exposed to atmospheric air. The time it will take for the wire to cool, the distance the wire travels, and the rate of heat transfer from the wire are to be determined.

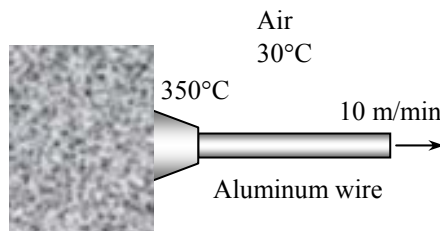
Assumptions 1 Heat conduction in the wires is one-dimensional in the radial direction. 2 The thermal properties of the aluminum are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

Properties The properties of aluminum are given to be $k = 236 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 2702 \text{ kg/m}^3$, $c_p = 0.896 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 9.75 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis (a) The characteristic length of the wire and the Biot number are

$$L_c = \frac{V}{A_s} = \frac{\pi r_o^2 L}{2\pi r_o L} = \frac{r_o}{2} = \frac{0.0015 \text{ m}}{2} = 0.00075 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(35 \text{ W/m}^2 \cdot ^\circ\text{C})(0.00075 \text{ m})}{236 \text{ W/m} \cdot ^\circ\text{C}} = 0.00011 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then,

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{35 \text{ W/m}^2 \cdot ^\circ\text{C}}{(2702 \text{ kg/m}^3)(896 \text{ J/kg} \cdot ^\circ\text{C})(0.00075 \text{ m})} = 0.0193 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{50 - 30}{350 - 30} = e^{-(0.0193 \text{ s}^{-1})t} \longrightarrow t = \mathbf{144 \text{ s}}$$

(b) The wire travels a distance of

$$\text{velocity} = \frac{\text{length}}{\text{time}} \rightarrow \text{length} = (10 / 60 \text{ m/s})(144 \text{ s}) = \mathbf{24 \text{ m}}$$

This distance can be reduced by cooling the wire in a water or oil bath.

(c) The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{V} = \rho(\pi r_o^2)V = (2702 \text{ kg/m}^3)\pi(0.0015 \text{ m})^2(10 \text{ m/min}) = 0.191 \text{ kg/min}$$

Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.191 \text{ kg/min})(0.896 \text{ kJ/kg} \cdot ^\circ\text{C})(350 - 50)^\circ\text{C} = 51.3 \text{ kJ/min} = \mathbf{856 \text{ W}}$$

11-109 Long copper wires are extruded and exposed to atmospheric air. The time it will take for the wire to cool, the distance the wire travels, and the rate of heat transfer from the wire are to be determined.

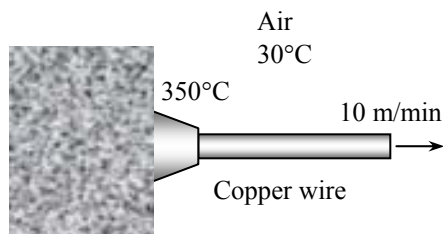
Assumptions 1 Heat conduction in the wires is one-dimensional in the radial direction. 2 The thermal properties of the copper are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

Properties The properties of copper are given to be $k = 386 \text{ W/m} \cdot ^\circ\text{C}$, $\rho = 8950 \text{ kg/m}^3$, $c_p = 0.383 \text{ kJ/kg} \cdot ^\circ\text{C}$, and $\alpha = 1.13 \times 10^{-4} \text{ m}^2/\text{s}$.

Analysis (a) The characteristic length of the wire and the Biot number are

$$L_c = \frac{V}{A_s} = \frac{\pi r_o^2 L}{2\pi r_o L} = \frac{r_o}{2} = \frac{0.0015 \text{ m}}{2} = 0.00075 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(35 \text{ W/m}^2 \cdot ^\circ\text{C})(0.00075 \text{ m})}{386 \text{ W/m} \cdot ^\circ\text{C}} = 0.000068 < 0.1$$



Since $Bi < 0.1$ the lumped system analysis is applicable. Then,

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{35 \text{ W/m}^2 \cdot ^\circ\text{C}}{(8950 \text{ kg/m}^3)(0.383 \text{ J/kg} \cdot ^\circ\text{C})(0.00075 \text{ m})} = 0.0136 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{50 - 30}{350 - 30} = e^{-(0.0136 \text{ s}^{-1})t} \longrightarrow t = \mathbf{204 \text{ s}}$$

(b) The wire travels a distance of

$$\text{velocity} = \frac{\text{length}}{\text{time}} \longrightarrow \text{length} = \left(\frac{10 \text{ m/min}}{60 \text{ s/min}} \right) (204 \text{ s}) = \mathbf{34 \text{ m}}$$

This distance can be reduced by cooling the wire in a water or oil bath.

(c) The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{V} = \rho(\pi r_o^2)V = (8950 \text{ kg/m}^3)\pi(0.0015 \text{ m})^2(10 \text{ m/min}) = 0.633 \text{ kg/min}$$

Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.633 \text{ kg/min})(0.383 \text{ kJ/kg} \cdot ^\circ\text{C})(350 - 50)^\circ\text{C} = 72.7 \text{ kJ/min} = \mathbf{1212 \text{ W}}$$

11-110 A brick house made of brick that was initially cold is exposed to warm atmospheric air at the outer surfaces. The time it will take for the temperature of the inner surfaces of the house to start changing is to be determined.

Assumptions 1 The temperature in the wall is affected by the thermal conditions at outer surfaces only, and thus the wall can be considered to be a semi-infinite medium with a specified outer surface temperature of 18°C. 2 The thermal properties of the brick wall are constant.

Properties The thermal properties of the brick are given to be $k = 0.72 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.45 \times 10^{-6} \text{ m}^2/\text{s}$.

Analysis The exact analytical solution to this problem is

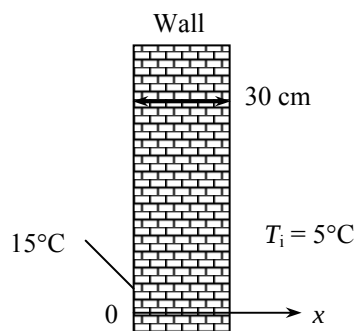
$$\frac{T(x, t) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{\sqrt{\alpha t}}\right)$$

Substituting,

$$\frac{5.1 - 5}{15 - 5} = 0.01 = \text{erfc}\left(\frac{0.3 \text{ m}}{2\sqrt{(0.45 \times 10^{-6} \text{ m}^2/\text{s})t}}\right)$$

Noting from Table 11-4 that $0.01 = \text{erfc}(1.8215)$, the time is determined to be

$$\left(\frac{0.3 \text{ m}}{2\sqrt{(0.45 \times 10^{-6} \text{ m}^2/\text{s})t}}\right) = 1.8215 \longrightarrow t = 15,070 \text{ s} = \mathbf{251 \text{ min}}$$



11-111 A thick wall is exposed to cold outside air. The wall temperatures at distances 15, 30, and 40 cm from the outer surface at the end of 2-hour cooling period are to be determined.

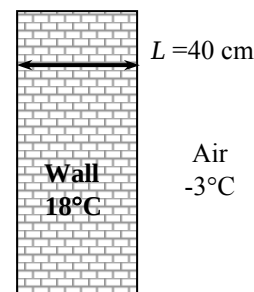
Assumptions 1 The temperature in the wall is affected by the thermal conditions at outer surfaces only. Therefore, the wall can be considered to be a semi-infinite medium **2** The thermal properties of the wall are constant.

Properties The thermal properties of the brick are given to be $k = 0.72 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.6 \times 10^{-7} \text{ m}^2/\text{s}$.

Analysis For a 15 cm distance from the outer surface, from Fig. 11-29 we have

$$\left. \begin{aligned} \frac{h\sqrt{\alpha t}}{k} &= \frac{(20 \text{ W/m}^2 \cdot ^\circ\text{C})\sqrt{(1.6 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}}{0.72 \text{ W/m} \cdot ^\circ\text{C}} = 2.98 \\ \eta &= \frac{x}{2\sqrt{\alpha t}} = \frac{0.15 \text{ m}}{2\sqrt{(1.6 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}} = 0.70 \end{aligned} \right\} 1 - \frac{T - T_\infty}{T_i - T_\infty} = 0.25$$

$$1 - \frac{T - (-3)}{18 - (-3)} = 0.25 \longrightarrow T = \mathbf{12.8^\circ\text{C}}$$



For a 30 cm distance from the outer surface, from Fig. 11-29 we have

$$\left. \begin{aligned} \frac{h\sqrt{\alpha t}}{k} &= \frac{(20 \text{ W/m}^2 \cdot ^\circ\text{C})\sqrt{(1.6 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}}{0.72 \text{ W/m} \cdot ^\circ\text{C}} = 2.98 \\ \eta &= \frac{x}{2\sqrt{\alpha t}} = \frac{0.3 \text{ m}}{2\sqrt{(1.6 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}} = 1.40 \end{aligned} \right\} 1 - \frac{T - T_\infty}{T_i - T_\infty} = 0.038$$

$$1 - \frac{T - (-3)}{18 - (-3)} = 0.038 \longrightarrow T = \mathbf{17.2^\circ\text{C}}$$

For a 40 cm distance from the outer surface, that is for the inner surface, from Fig. 11-29 we have

$$\left. \begin{aligned} \frac{h\sqrt{\alpha t}}{k} &= \frac{(20 \text{ W/m}^2 \cdot ^\circ\text{C})\sqrt{(1.6 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}}{0.72 \text{ W/m} \cdot ^\circ\text{C}} = 2.98 \\ \eta &= \frac{x}{2\sqrt{\alpha t}} = \frac{0.4 \text{ m}}{2\sqrt{(1.6 \times 10^{-7} \text{ m}^2/\text{s})(2 \times 3600 \text{ s})}} = 1.87 \end{aligned} \right\} 1 - \frac{T - T_\infty}{T_i - T_\infty} = 0$$

$$1 - \frac{T - (-3)}{18 - (-3)} = 0 \longrightarrow T = \mathbf{18.0^\circ\text{C}}$$

Discussion This last result shows that the semi-infinite medium assumption is a valid one.

11-112 The engine block of a car is allowed to cool in atmospheric air. The temperatures at the center of the top surface and at the corner after a specified period of cooling are to be determined.

Assumptions 1 Heat conduction in the block is three-dimensional, and thus the temperature varies in all three directions. 2 The thermal properties of the block are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of cast iron are given to be $k = 52 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 1.7 \times 10^{-5} \text{ m}^2/\text{s}$.

Analysis This rectangular block can physically be formed by the intersection of two infinite plane walls of thickness $2L = 40 \text{ cm}$ (call planes A and B) and an infinite plane wall of thickness $2L = 80 \text{ cm}$ (call plane C). We measure x from the center of the block.

(a) The Biot number is calculated for each of the plane wall to be

$$Bi_A = Bi_B = \frac{hL}{k} = \frac{(6 \text{ W/m}^2 \cdot ^\circ\text{C})(0.2 \text{ m})}{(52 \text{ W/m} \cdot ^\circ\text{C})} = 0.0231$$

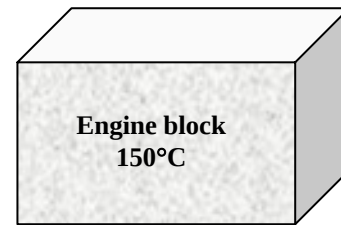
$$Bi_C = \frac{hL}{k} = \frac{(6 \text{ W/m}^2 \cdot ^\circ\text{C})(0.4 \text{ m})}{(52 \text{ W/m} \cdot ^\circ\text{C})} = 0.0462$$

The constants λ_1 and A_1 corresponding to these Biot numbers are, from Table 11-2,

$$\lambda_{1(A,B)} = 0.150 \quad \text{and} \quad A_{1(A,B)} = 1.0038$$

$$\lambda_{1(C)} = 0.212 \quad \text{and} \quad A_{1(C)} = 1.0076$$

Air
17°C



The Fourier numbers are

$$\tau_{A,B} = \frac{\alpha t}{L^2} = \frac{(1.70 \times 10^{-5} \text{ m}^2/\text{s})(45 \text{ min} \times 60 \text{ s/min})}{(0.2 \text{ m})^2} = 1.1475 > 0.2$$

$$\tau_C = \frac{\alpha t}{L^2} = \frac{(1.70 \times 10^{-5} \text{ m}^2/\text{s})(45 \text{ min} \times 60 \text{ s/min})}{(0.4 \text{ m})^2} = 0.2869 > 0.2$$

The center of the top surface of the block (whose sides are 80 cm and 40 cm) is at the center of the plane wall with $2L = 80 \text{ cm}$, at the center of the plane wall with $2L = 40 \text{ cm}$, and at the surface of the plane wall with $2L = 40 \text{ cm}$. The dimensionless temperatures are

$$\theta_{o, \text{wall (A)}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.0038) e^{-(0.150)^2 (1.1475)} = 0.9782$$

$$\theta(L, t)_{\text{wall (B)}} = \frac{T(x, t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L / L) = (1.0038) e^{-(0.150)^2 (1.1475)} \cos(0.150) = 0.9672$$

$$\theta_{o, \text{wall (C)}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.0076) e^{-(0.212)^2 (0.2869)} = 0.9947$$

Then the center temperature of the top surface of the cylinder becomes

$$\left[\frac{T(L, 0, 0, t) - T_\infty}{T_i - T_\infty} \right]_{\text{short cylinder}} = \theta(L, t)_{\text{wall (B)}} \times \theta_{o, \text{wall (A)}} \times \theta_{o, \text{wall (C)}} = 0.9672 \times 0.9782 \times 0.9947 = 0.9411$$

$$\frac{T(L, 0, 0, t) - 17}{150 - 17} = 0.9411 \longrightarrow T(L, 0, 0, t) = \mathbf{142.2^\circ\text{C}}$$

(b) The corner of the block is at the surface of each plane wall. The dimensionless temperature for the surface of the plane walls with $2L = 40$ cm is determined in part (a). The dimensionless temperature for the surface of the plane wall with $2L = 80$ cm is determined from

$$\theta(L, t)_{\text{wall (C)}} = \frac{T(x, t) - T_{\infty}}{T_i - T_{\infty}} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L / L) = (1.0076) e^{-(0.212)^2 (0.2869)} \cos(0.212) = 0.9724$$

Then the corner temperature of the block becomes

$$\left[\frac{T(L, L, L, t) - T_{\infty}}{T_i - T_{\infty}} \right]_{\text{short cylinder}} = \theta(L, t)_{\text{wall, C}} \times \theta(L, t)_{\text{wall, B}} \times \theta(L, t)_{\text{wall, A}} = 0.9724 \times 0.9672 \times 0.9672 = 0.9097$$

$$\frac{T(L, L, L, t) - 17}{150 - 17} = 0.9097 \longrightarrow T(L, L, L, t) = \mathbf{138.0^{\circ}\text{C}}$$

11-113 A man is found dead in a room. The time passed since his death is to be estimated.

Assumptions 1 Heat conduction in the body is two-dimensional, and thus the temperature varies in both radial r - and x - directions. 2 The thermal properties of the body are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface. 4 The human body is modeled as a cylinder. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of body are given to be $k = 0.62 \text{ W/m} \cdot ^\circ\text{C}$ and $\alpha = 0.15 \times 10^{-6} \text{ m}^2/\text{s}$.

Analysis A short cylinder can be formed by the intersection of a long cylinder of radius $D/2 = 14 \text{ cm}$ and a plane wall of thickness $2L = 180 \text{ cm}$. We measure x from the midplane. The temperature of the body is specified at a point that is at the center of the plane wall but at the surface of the cylinder. The Biot numbers and the corresponding constants are first determined to be

$$Bi_{\text{wall}} = \frac{hL}{k} = \frac{(9 \text{ W/m}^2 \cdot ^\circ\text{C})(0.90 \text{ m})}{(0.62 \text{ W/m} \cdot ^\circ\text{C})} = 13.06$$

$$\longrightarrow \lambda_1 = 1.4495 \quad \text{and} \quad A_1 = 1.2644$$

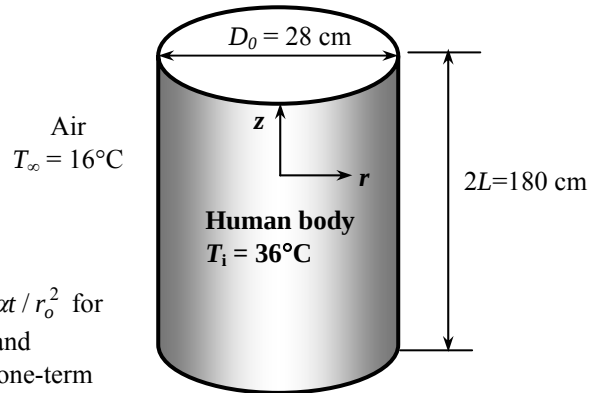
$$Bi_{\text{cyl}} = \frac{hr_o}{k} = \frac{(9 \text{ W/m}^2 \cdot ^\circ\text{C})(0.14 \text{ m})}{(0.62 \text{ W/m} \cdot ^\circ\text{C})} = 2.03$$

$$\longrightarrow \lambda_1 = 1.6052 \quad \text{and} \quad A_1 = 1.3408$$

Noting that $\tau = \alpha t / L^2$ for the plane wall and $\tau = \alpha t / r_o^2$ for cylinder and $J_0(1.6052) = 0.4524$ from Table 11-3, and assuming that $\tau > 0.2$ in all dimensions so that the one-term approximate solution for transient heat conduction is applicable, the product solution method can be written for this problem as

$$\begin{aligned} \theta(0, r_0, t)_{\text{block}} &= \theta(0, t)_{\text{wall}} \theta(r_0, t)_{\text{cyl}} \\ \frac{23-16}{36-16} &= (A_1 e^{-\lambda_1^2 \tau}) \left[A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r / r_o) \right] \\ 0.40 &= \left\{ (1.2644) \exp \left[- (1.4495)^2 \frac{(0.15 \times 10^{-6})t}{(0.90)^2} \right] \right\} \\ &\quad \times \left\{ (1.3408) \exp \left[- (1.6052)^2 \frac{(0.15 \times 10^{-6})t}{(0.14)^2} \right] (0.4524) \right\} \end{aligned}$$

$$\longrightarrow t = 32,404 \text{ s} = \mathbf{9.0 \text{ hours}}$$



11-114 An exothermic process occurs uniformly throughout a sphere. The variation of temperature with time is to be obtained. The steady-state temperature of the sphere and the time needed for the sphere to reach the average of its initial and final (steady) temperatures are to be determined.

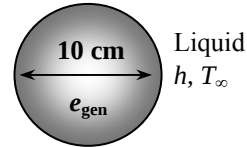
Assumptions 1 The sphere may be approximated as a lumped system. 2 The thermal properties of the sphere are constant. 3 The heat transfer coefficient is constant and uniform over the entire surface.

Properties The properties of sphere are given to be $k = 300 \text{ W/m}\cdot\text{K}$, $c_p = 400 \text{ J/kg}\cdot\text{K}$, $\rho = 7500 \text{ kg/m}^3$.

Analysis (a) First, we check the applicability of lumped system as follows:

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{\pi D^3 / 6}{\pi D^2} = \frac{D}{6} = \frac{0.10 \text{ m}}{6} = 0.0167 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(250 \text{ W/m}^2\cdot\text{K})(0.0167 \text{ m})}{300 \text{ W/m}\cdot\text{K}} = 0.014 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. An energy balance on the system may be written to give

$$\dot{e}_{\text{gen}} V = hA(T - T_{\infty}) + mc \frac{dT}{dt}$$

$$\dot{e}_{\text{gen}} (\pi D^3 / 6) = h\pi D^2 (T - T_{\infty}) + \rho(\pi D^3 / 6) \frac{dT}{dt}$$

$$(1.2 \times 10^6) \pi (0.10)^3 / 6 = (250) \pi (0.10)^2 (T - 20) + (7500) [\pi (0.10)^3 / 6] (400) \frac{dT}{dt}$$

$$20,000 = 250T - 5000 + 50,000 \frac{dT}{dt}$$

$$\frac{dT}{dt} = 0.5 - 0.005T$$

(b) Now, we use integration to get the variation of sphere temperature with time

$$\frac{dT}{dt} = 0.5 - 0.005T$$

$$\frac{dT}{0.5 - 0.005T} = dt \longrightarrow \int_{20}^T \frac{dT}{0.5 - 0.005T} = \int_0^t dt$$

$$-\frac{1}{0.005} \ln(0.5 - 0.005T) \Big|_{20}^T = t \Big|_0^t = t$$

$$\ln\left(\frac{0.5 - 0.005T}{0.5 - 0.005 \times 20}\right) = -0.005t \longrightarrow \frac{0.5 - 0.005T}{0.4} = e^{-0.005t}$$

$$0.005T = 0.5 - 0.4e^{-0.005t} \longrightarrow T = 100 - 80e^{-0.005t}$$

We obtain the steady-state temperature by setting time to infinity:

$$T = 100 - 80e^{-0.005t} = 100 - e^{-\infty} = \mathbf{100^{\circ}\text{C}}$$

or $\frac{dT}{dt} = 0 \longrightarrow 0.5 - 0.005T = 0 \longrightarrow T = 100^{\circ}\text{C}$

(c) The time needed for the sphere to reach the average of its initial and final (steady) temperatures is determined from

$$T = 100 - 80e^{-0.005t}$$

$$\frac{20 + 100}{2} = 100 - 80e^{-0.005t} \longrightarrow t = \mathbf{139 \text{ s}}$$

11-115 Large steel plates are quenched in an oil reservoir. The quench time is to be determined.

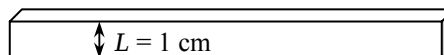
Assumptions 1 The thermal properties of the plates are constant. 2 The heat transfer coefficient is constant and uniform over the entire surface.

Properties The properties of steel plates are given to be $k = 45 \text{ W/m}\cdot\text{K}$, $\rho = 7800 \text{ kg/m}^3$, and $c_p = 470 \text{ J/kg}\cdot\text{K}$.

Analysis For sphere, the characteristic length and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{L}{2} = \frac{0.01 \text{ m}}{2} = 0.005 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(400 \text{ W/m}^2\cdot\text{C})(0.005 \text{ m})}{45 \text{ W/m}\cdot\text{C}} = 0.044 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then the cooling time is determined from

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{400 \text{ W/m}^2\cdot\text{C}}{(7800 \text{ kg/m}^3)(470 \text{ J/kg}\cdot\text{C})(0.005 \text{ m})} = 0.02182 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{100 - 30}{600 - 30} = e^{-(0.02182 \text{ s}^{-1})t} \longrightarrow t = 96 \text{ s} = \mathbf{1.6 \text{ min}}$$

11-116 Aluminum wires leaving the extruder at a specified rate are cooled in air. The necessary length of the wire is to be determined.

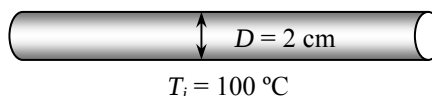
Assumptions 1 The thermal properties of the geometry are constant. 2 The heat transfer coefficient is constant and uniform over the entire surface.

Properties The properties of aluminum are $k = 237 \text{ W/m}\cdot\text{C}$, $\rho = 2702 \text{ kg/m}^3$, and $c_p = 0.903 \text{ kJ/kg}\cdot\text{C}$ (Table A-24).

Analysis For a long cylinder, the characteristic length and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{(\pi D^2 / 4)L}{\pi DL} = \frac{D}{4} = \frac{0.003 \text{ m}}{4} = 0.00075 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(50 \text{ W/m}^2\cdot\text{C})(0.00075 \text{ m})}{237 \text{ W/m}\cdot\text{C}} = 0.00016 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then the cooling time is determined from

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{50 \text{ W/m}^2\cdot\text{C}}{(2702 \text{ kg/m}^3)(903 \text{ J/kg}\cdot\text{C})(0.00075 \text{ m})} = 0.02732 \text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{50 - 25}{350 - 25} = e^{-(0.02732 \text{ s}^{-1})t} \longrightarrow t = 93.9 \text{ s}$$

Then the necessary length of the wire in the cooling section is determined to be

$$\text{Length} = \frac{t}{V} = \frac{(93.9 / 60) \text{ min}}{10 \text{ m/min}} = \mathbf{0.157 \text{ m}}$$

11-117 ... 11-120 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 12
EXTERNAL FORCED CONVECTION

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Mechanism and Types of Convection

12-1C In forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect that manifests itself as the rise of the warmer fluid and the fall of the cooler fluid. The convection caused by winds is natural convection for the earth, but it is forced convection for bodies subjected to the winds since for the body it makes no difference whether the air motion is caused by a fan or by the winds.

12-2C If the fluid is forced to flow over a surface, it is called external forced convection. If it is forced to flow in a tube, it is called internal forced convection. A heat transfer system can involve both internal and external convection simultaneously. Example: A pipe transporting a fluid in a windy area.

12-3C The convection heat transfer coefficient will usually be higher in forced convection since heat transfer coefficient depends on the fluid velocity, and forced convection involves higher fluid velocities.

12-4C The potato will normally cool faster by blowing warm air to it despite the smaller temperature difference in this case since the fluid motion caused by blowing enhances the heat transfer coefficient considerably.

12-5C Nusselt number is the dimensionless convection heat transfer coefficient, and it represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer. It is defined as $Nu = \frac{hL_c}{k}$ where L_c is the characteristic length of the surface and k is the thermal conductivity of the fluid.

12-6C Heat transfer through a fluid is conduction in the absence of bulk fluid motion, and convection in the presence of it. The rate of heat transfer is higher in convection because of fluid motion. The value of the convection heat transfer coefficient depends on the fluid motion as well as the fluid properties. Thermal conductivity is a fluid property, and its value does not depend on the flow.

12-7C A fluid flow during which the density of the fluid remains nearly constant is called *incompressible flow*. A fluid whose density is practically independent of pressure (such as a liquid) is called an incompressible fluid. The flow of compressible fluid (such as air) is not necessarily compressible since the density of a compressible fluid may still remain constant during flow.

12-8 Heat transfer coefficients at different air velocities are given during air cooling of potatoes. The initial rate of heat transfer from a potato and the temperature gradient at the potato surface are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Potato is spherical in shape. 3 Convection heat transfer coefficient is constant over the entire surface.

Properties The thermal conductivity of the potato is given to be $k = 0.49 \text{ W/m} \cdot ^\circ\text{C}$.

Analysis The initial rate of heat transfer from a potato is

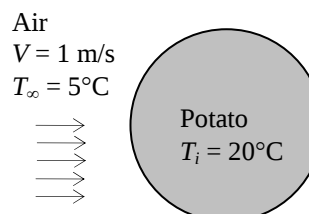
$$A_s = \pi D^2 = \pi (0.08 \text{ m})^2 = 0.02011 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (19.1 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02011 \text{ m}^2)(20 - 5)^\circ\text{C} = \mathbf{5.8 \text{ W}}$$

where the heat transfer coefficient is obtained from the table at 1 m/s velocity. The initial value of the temperature gradient at the potato surface is

$$\dot{q}_{\text{conv}} = \dot{q}_{\text{cond}} = -k \left(\frac{\partial T}{\partial r} \right)_{r=R} = h(T_s - T_\infty)$$

$$\left. \frac{\partial T}{\partial r} \right|_{r=R} = -\frac{h(T_s - T_\infty)}{k} = -\frac{(19.1 \text{ W/m}^2 \cdot ^\circ\text{C})(20 - 5)^\circ\text{C}}{0.49 \text{ W/m} \cdot ^\circ\text{C}} = \mathbf{-585^\circ\text{C/m}}$$



12-9 The rate of heat loss from an average man walking in still air is to be determined at different walking velocities.

Assumptions 1 Steady operating conditions exist. 2 Convection heat transfer coefficient is constant over the entire surface.

Analysis The convection heat transfer coefficients and the rate of heat losses at different walking velocities are

$$(a) \ h = 8.6V^{0.53} = 8.6(0.5 \text{ m/s})^{0.53} = 5.956 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.956 \text{ W/m}^2 \cdot ^\circ\text{C})(1.8 \text{ m}^2)(30 - 10)^\circ\text{C} = \mathbf{214.4 \text{ W}}$$

$$(b) \ h = 8.6V^{0.53} = 8.6(1.0 \text{ m/s})^{0.53} = 8.60 \text{ W/m}^2 \cdot ^\circ\text{C}$$

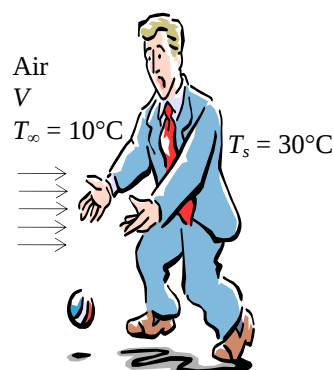
$$\dot{Q} = hA_s(T_s - T_\infty) = (8.60 \text{ W/m}^2 \cdot ^\circ\text{C})(1.8 \text{ m}^2)(30 - 10)^\circ\text{C} = \mathbf{309.6 \text{ W}}$$

$$(c) \ h = 8.6V^{0.53} = 8.6(1.5 \text{ m/s})^{0.53} = 10.66 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (10.66 \text{ W/m}^2 \cdot ^\circ\text{C})(1.8 \text{ m}^2)(30 - 10)^\circ\text{C} = \mathbf{383.8 \text{ W}}$$

$$(d) \ h = 8.6V^{0.53} = 8.6(2.0 \text{ m/s})^{0.53} = 12.42 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (12.42 \text{ W/m}^2 \cdot ^\circ\text{C})(1.8 \text{ m}^2)(30 - 10)^\circ\text{C} = \mathbf{447.0 \text{ W}}$$



12-10 The rate of heat loss from an average man walking in windy air is to be determined at different wind velocities.

Assumptions 1 Steady operating conditions exist. 2 Convection heat transfer coefficient is constant over the entire surface.

Analysis The convection heat transfer coefficients and the rate of heat losses at different wind velocities are

$$(a) \ h = 14.8V^{0.69} = 14.8(0.5 \text{ m/s})^{0.69} = 9.174 \text{ W/m}^2 \cdot ^\circ\text{C}$$

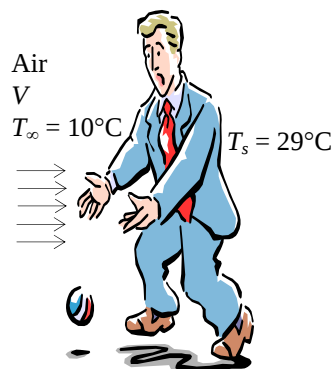
$$\dot{Q} = hA_s(T_s - T_\infty) = (9.174 \text{ W/m}^2 \cdot ^\circ\text{C})(1.7 \text{ m}^2)(29 - 10)^\circ\text{C} = \mathbf{296.3 \text{ W}}$$

$$(b) \ h = 14.8V^{0.69} = 14.8(1.0 \text{ m/s})^{0.69} = 14.8 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (14.8 \text{ W/m}^2 \cdot ^\circ\text{C})(1.7 \text{ m}^2)(29 - 10)^\circ\text{C} = \mathbf{478.0 \text{ W}}$$

$$(c) \ h = 14.8V^{0.69} = 14.8(1.5 \text{ m/s})^{0.69} = 19.58 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (19.58 \text{ W/m}^2 \cdot ^\circ\text{C})(1.7 \text{ m}^2)(29 - 10)^\circ\text{C} = \mathbf{632.4 \text{ W}}$$



12-11 The expression for the heat transfer coefficient for air cooling of some fruits is given. The initial rate of heat transfer from an orange, the temperature gradient at the orange surface, and the value of the Nusselt number are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Orange is spherical in shape. 3 Convection heat transfer coefficient is constant over the entire surface. 4 Properties of water is used for orange.

Properties The thermal conductivity of the orange is given to be $k = 0.50 \text{ W/m} \cdot ^\circ\text{C}$. The thermal conductivity and the kinematic viscosity of air at the film temperature of $(T_s + T_\infty)/2 = (15 + 5)/2 = 10^\circ\text{C}$ are (Table A-22)

$$k = 0.02439 \text{ W/m} \cdot ^\circ\text{C}, \quad \nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

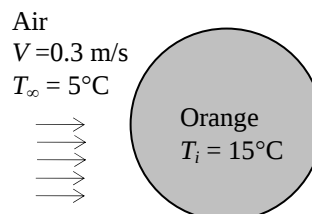
Analysis (a) The Reynolds number, the heat transfer coefficient, and the initial rate of heat transfer from an orange are

$$A_s = \pi D^2 = \pi(0.07 \text{ m})^2 = 0.01539 \text{ m}^2$$

$$\text{Re} = \frac{VD}{\nu} = \frac{(0.3 \text{ m/s})(0.07 \text{ m})}{1.426 \times 10^{-5} \text{ m}^2/\text{s}} = 1473$$

$$h = \frac{5.05k_{\text{air}} \text{Re}^{1/3}}{D} = \frac{5.05(0.02439 \text{ W/m} \cdot ^\circ\text{C})(1473)^{1/3}}{0.07 \text{ m}} = 20.02 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (20.02 \text{ W/m}^2 \cdot ^\circ\text{C})(0.01539 \text{ m}^2)(15 - 5)^\circ\text{C} = \mathbf{3.08 \text{ W}}$$



(b) The temperature gradient at the orange surface is determined from

$$\dot{q}_{\text{conv}} = \dot{q}_{\text{cond}} = -k \left(\frac{\partial T}{\partial r} \right)_{r=R} = h(T_s - T_\infty)$$

$$\left. \frac{\partial T}{\partial r} \right|_{r=R} = -\frac{h(T_s - T_\infty)}{k} = -\frac{(20.02 \text{ W/m}^2 \cdot ^\circ\text{C})(15 - 5)^\circ\text{C}}{0.50 \text{ W/m} \cdot ^\circ\text{C}} = \mathbf{-400^\circ\text{C/m}}$$

(c) The Nusselt number is

$$\text{Nu} = \frac{hD}{k} = \frac{(20.02 \text{ W/m}^2 \cdot ^\circ\text{C})(0.07 \text{ m})}{0.02439 \text{ W/m} \cdot ^\circ\text{C}} = \mathbf{57.5}$$

Velocity and Thermal Boundary Layers

12-12C Viscosity is a measure of the “stickiness” or “resistance to deformation” of a fluid. It is due to the internal frictional force that develops between different layers of fluids as they are forced to move relative to each other. Viscosity is caused by the cohesive forces between the molecules in liquids, and by the molecular collisions in gases. Liquids have higher dynamic viscosities than gases.

12-13C The fluids whose shear stress is proportional to the velocity gradient are called *Newtonian fluids*. Most common fluids such as water, air, gasoline, and oil are Newtonian fluids.

12-14C A fluid in direct contact with a solid surface sticks to the surface and there is no slip. This is known as the *no-slip condition*, and it is due to the viscosity of the fluid.

12-15C The ball reaches the bottom of the container first in water due to lower viscosity of water compared to oil.

12-16C (a) The dynamic viscosity of liquids decreases with temperature. (b) The dynamic viscosity of gases increases with temperature.

12-17C The fluid viscosity is responsible for the development of the velocity boundary layer. For the idealized inviscid fluids (fluids with zero viscosity), there will be no velocity boundary layer.

12-18C The Prandtl number $Pr = \nu / \alpha$ is a measure of the relative magnitudes of the diffusivity of momentum (and thus the development of the velocity boundary layer) and the diffusivity of heat (and thus the development of the thermal boundary layer). The Pr is a fluid property, and thus its value is independent of the type of flow and flow geometry. The Pr changes with temperature, but not pressure.

12-19C A thermal boundary layer will not develop in flow over a surface if both the fluid and the surface are at the same temperature since there will be no heat transfer in that case.

Laminar and Turbulent Flows

12-20C A fluid motion is laminar when it involves smooth streamlines and highly ordered motion of molecules, and turbulent when it involves velocity fluctuations and highly disordered motion. The heat transfer coefficient is higher in turbulent flow.

12-21C Reynolds number is the ratio of the inertial forces to viscous forces, and it serves as a criterion for determining the flow regime. For flow over a plate of length L it is defined as $Re = VL/\nu$ where V is flow velocity and ν is the kinematic viscosity of the fluid.

12-22C The friction coefficient represents the resistance to fluid flow over a flat plate. It is proportional to the drag force acting on the plate. The drag coefficient for a flat surface is equivalent to the mean friction coefficient.

12-23C In turbulent flow, it is the *turbulent eddies* due to enhanced mixing that cause the friction factor to be larger.

12-24C Turbulent viscosity μ_t is caused by turbulent eddies, and it accounts for momentum transport by turbulent eddies. It is expressed as $\tau_t = -\rho \overline{u'v'} = \mu_t \frac{\partial \bar{u}}{\partial y}$ where $\bar{u}$ is the mean value of velocity in the flow direction and u' and v' are the fluctuating components of velocity.

12-25C Turbulent thermal conductivity k_t is caused by turbulent eddies, and it accounts for thermal energy transport by turbulent eddies. It is expressed as $\dot{q}_t = \rho c_p \overline{v'T'} = -k_t \frac{\partial \bar{T}}{\partial y}$ where T' is the eddy temperature relative to the mean value, and $\dot{q}_t = \rho c_p v'T'$ the rate of thermal energy transport by turbulent eddies.

Drag Force and Heat Transfer in External Flow

12-26C The velocity of the fluid relative to the immersed solid body sufficiently far away from a body is called the *free-stream velocity*, V_∞ . The *upstream* (or *approach*) *velocity* V is the velocity of the approaching fluid far ahead of the body. These two velocities are equal if the flow is uniform and the body is small relative to the scale of the free-stream flow.

12-27C A body is said to be *streamlined* if a conscious effort is made to align its shape with the anticipated streamlines in the flow. Otherwise, a body tends to block the flow, and is said to be *blunt*. A tennis ball is a blunt body (unless the velocity is very low and we have “creeping flow”).

12-28C The force a flowing fluid exerts on a body in the flow direction is called *drag*. Drag is caused by friction between the fluid and the solid surface, and the pressure difference between the front and back of the body. We try to minimize drag in order to reduce fuel consumption in vehicles, improve safety and durability of structures subjected to high winds, and to reduce noise and vibration.

12-29C The force a flowing fluid exerts on a body in the normal direction to flow that tend to move the body in that direction is called *lift*. It is caused by the components of the pressure and wall shear forces in the normal direction to flow. The wall shear also contributes to lift (unless the body is very slim), but its contribution is usually small.

12-30C When the drag force F_D , the upstream velocity V , and the fluid density ρ are measured during flow over a body, the drag coefficient can be determined from

$$C_D = \frac{F_D}{\frac{1}{2} \rho V^2 A}$$

where A is ordinarily the *frontal area* (the area projected on a plane normal to the direction of flow) of the body.

12-31C The *frontal area* of a body is the area seen by a person when looking from upstream. The frontal area is appropriate to use in drag and lift calculations for blunt bodies such as cars, cylinders, and spheres.

12-32C The part of drag that is due directly to wall shear stress τ_w is called the *skin friction drag* $F_{D, \text{friction}}$ since it is caused by frictional effects, and the part that is due directly to pressure P and depends strongly on the shape of the body is called the *pressure drag* $F_{D, \text{pressure}}$. For slender bodies such as airfoils, the friction drag is usually more significant.

12-33C The friction drag coefficient is independent of surface roughness in *laminar flow*, but is a strong function of surface roughness in *turbulent flow* due to surface roughness elements protruding further into the highly viscous laminar sublayer.

12-34C As a result of streamlining, (a) friction drag increases, (b) pressure drag decreases, and (c) total drag decreases at high Reynolds numbers (the general case), but increases at very low Reynolds numbers since the friction drag dominates at low Reynolds numbers.

12-35C At sufficiently high velocities, the fluid stream detaches itself from the surface of the body. This is called *separation*. It is caused by a fluid flowing over a curved surface at a high velocity (or technically, by adverse pressure gradient). Separation increases the drag coefficient drastically.

Flow over Flat Plates

12-36C The friction coefficient represents the resistance to fluid flow over a flat plate. It is proportional to the drag force acting on the plate. The drag coefficient for a flat surface is equivalent to the mean friction coefficient.

12-37C The friction and the heat transfer coefficients change with position in laminar flow over a flat plate.

12-38C The average friction and heat transfer coefficients in flow over a flat plate are determined by integrating the local friction and heat transfer coefficients over the entire plate, and then dividing them by the length of the plate.

12-39 Hot engine oil flows over a flat plate. The total drag force and the rate of heat transfer per unit width of the plate are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible.

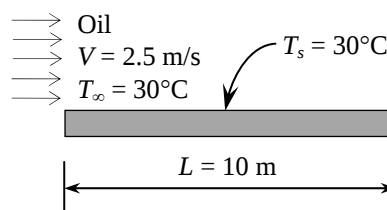
Properties The properties of engine oil at the film temperature of $(T_s + T_\infty)/2 = (80 + 30)/2 = 55^\circ\text{C}$ are (Table A-19)

$$\rho = 867 \text{ kg/m}^3 \quad \nu = 7.045 \times 10^{-5} \text{ m}^2/\text{s}$$

$$k = 0.1414 \text{ W/m}\cdot^\circ\text{C} \quad Pr = 1551$$

Analysis Noting that $L = 10 \text{ m}$, the Reynolds number at the end of the plate is

$$Re_L = \frac{VL}{\nu} = \frac{(2.5 \text{ m/s})(10 \text{ m})}{7.045 \times 10^{-5} \text{ m}^2/\text{s}} = 3.549 \times 10^5$$



which is less than the critical Reynolds number. Thus we have laminar flow over the entire plate. The average friction coefficient and the drag force per unit width are determined from

$$C_f = 1.33 Re_L^{-0.5} = 1.33(3.549 \times 10^5)^{-0.5} = 0.002233$$

$$F_D = C_f A_s \frac{\rho V^2}{2} = (0.002233)(10 \times 1 \text{ m}^2) \frac{(867 \text{ kg/m}^3)(2.5 \text{ m/s})^2}{2} = \mathbf{60.5 \text{ N}}$$

Similarly, the average Nusselt number and the heat transfer coefficient are determined using the laminar flow relations for a flat plate,

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664(3.549 \times 10^5)^{0.5} (1551)^{1/3} = 4579$$

$$h = \frac{k}{L} Nu = \frac{0.1414 \text{ W/m}\cdot^\circ\text{C}}{10 \text{ m}} (4579) = 64.75 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat transfer is then determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_\infty - T_s) = (64.75 \text{ W/m}^2\cdot^\circ\text{C})(10 \times 1 \text{ m}^2)(80 - 30)^\circ\text{C} = 3.24 \times 10^4 \text{ W} = \mathbf{32.4 \text{ kW}}$$

12-40 The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties.

Properties The atmospheric pressure in atm is

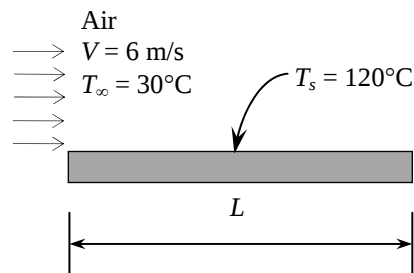
$$P = (83.4 \text{ kPa}) \frac{1 \text{ atm}}{101.325 \text{ kPa}} = 0.823 \text{ atm}$$

For an ideal gas, the thermal conductivity and the Prandtl number are independent of pressure, but the kinematic viscosity is inversely proportional to the pressure. With these considerations, the properties of air at 0.823 atm and at the film temperature of $(120+30)/2=75^\circ\text{C}$ are (Table A-22)

$$k = 0.02917 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = \nu_{@1\text{atm}} / P_{\text{atm}} = (2.046 \times 10^{-5} \text{ m}^2/\text{s}) / 0.823 = 2.486 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7166$$



Analysis (a) If the air flows parallel to the 8 m side, the Reynolds number in this case becomes

$$Re_L = \frac{VL}{\nu} = \frac{(6 \text{ m/s})(8 \text{ m})}{2.486 \times 10^{-5} \text{ m}^2/\text{s}} = 1.931 \times 10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(1.931 \times 10^6)^{0.8} - 871](0.7166)^{1/3} = 2757$$

$$h = \frac{k}{L} Nu = \frac{0.02917 \text{ W/m}\cdot^\circ\text{C}}{8 \text{ m}} (2757) = 10.05 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = wL = (2.5 \text{ m})(8 \text{ m}) = 20 \text{ m}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (10.05 \text{ W/m}^2\cdot^\circ\text{C})(20 \text{ m}^2)(120 - 30)^\circ\text{C} = 18,100 \text{ W} = \mathbf{18.10 \text{ kW}}$$

(b) If the air flows parallel to the 2.5 m side, the Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(6 \text{ m/s})(2.5 \text{ m})}{2.486 \times 10^{-5} \text{ m}^2/\text{s}} = 6.034 \times 10^5$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(6.034 \times 10^5)^{0.8} - 871](0.7166)^{1/3} = 615.1$$

$$h = \frac{k}{L} Nu = \frac{0.02917 \text{ W/m}\cdot^\circ\text{C}}{2.5 \text{ m}} (615.1) = 7.177 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (7.177 \text{ W/m}^2\cdot^\circ\text{C})(20 \text{ m}^2)(120 - 30)^\circ\text{C} = 12,920 \text{ W} = \mathbf{12.92 \text{ kW}}$$

12-41 Wind is blowing parallel to the wall of a house. The rate of heat loss from that wall is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties.

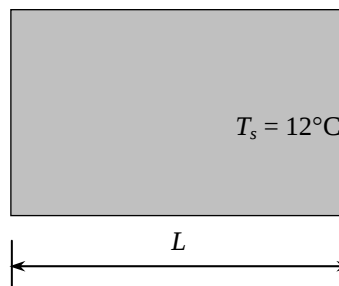
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (12+5)/2 = 8.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02428 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.413 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7340$$

Air
 $V = 55 \text{ km/h}$
 $T_\infty = 5^\circ\text{C}$



Analysis Air flows parallel to the 10 m side:

The Reynolds number in this case is

$$Re_L = \frac{VL}{\nu} = \frac{[(55 \times 1000 / 3600) \text{ m/s}](10 \text{ m})}{1.413 \times 10^{-5} \text{ m}^2/\text{s}} = 1.081 \times 10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient and then heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(1.081 \times 10^7)^{0.8} - 871](0.7340)^{1/3} = 1.336 \times 10^4$$

$$h = \frac{k}{L} Nu = \frac{0.02428 \text{ W/m} \cdot ^\circ\text{C}}{10 \text{ m}} (1.336 \times 10^4) = 32.43 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$A_s = wL = (4 \text{ m})(10 \text{ m}) = 40 \text{ m}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (32.43 \text{ W/m}^2 \cdot ^\circ\text{C})(40 \text{ m}^2)(12 - 5)^\circ\text{C} = 9080 \text{ W} = \mathbf{9.08 \text{ kW}}$$

If the wind velocity is doubled:

$$Re_L = \frac{VL}{\nu} = \frac{[(110 \times 1000 / 3600) \text{ m/s}](10 \text{ m})}{1.413 \times 10^{-5} \text{ m}^2/\text{s}} = 2.162 \times 10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(2.162 \times 10^7)^{0.8} - 871](0.7340)^{1/3} = 2.384 \times 10^4$$

$$h = \frac{k}{L} Nu = \frac{0.02428 \text{ W/m} \cdot ^\circ\text{C}}{10 \text{ m}} (2.384 \times 10^4) = 57.88 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (57.88 \text{ W/m}^2 \cdot ^\circ\text{C})(40 \text{ m}^2)(12 - 5)^\circ\text{C} = 16,210 \text{ W} = \mathbf{16.21 \text{ kW}}$$

12-42 EES Prob. 12-41 is reconsidered. The effects of wind velocity and outside air temperature on the rate of heat loss from the wall by convection are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

Vel=55 [km/h]
height=4 [m]
L=10 [m]
T_infinity=5 [C]
T_s=12 [C]

"PROPERTIES"

Fluid\$='air'
k=Conductivity(Fluid\$, T=T_film)
Pr=Prandtl(Fluid\$, T=T_film)
rho=Density(Fluid\$, T=T_film, P=101.3)
mu=Viscosity(Fluid\$, T=T_film)
nu=mu/rho
T_film=1/2*(T_s+T_infinity)

"ANALYSIS"

Re=(Vel*Convert(km/h, m/s)*L)/nu

"We use combined laminar and turbulent flow relation for Nusselt number"

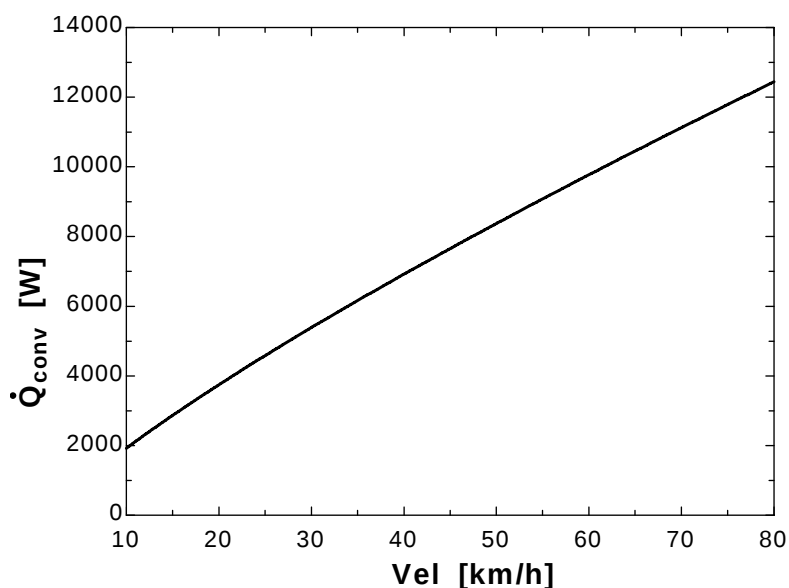
Nusselt=(0.037*Re^{0.8}-871)*Pr^(1/3)

h=k/L*Nusselt

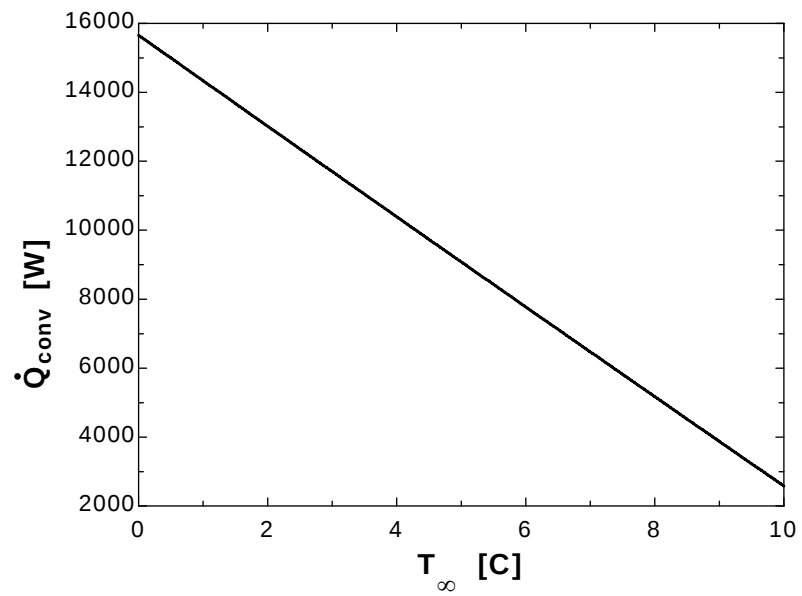
A=height*L

Q_dot_conv=h*A*(T_s-T_infinity)

Vel [km/h]	Q _{conv} [W]
10	1924
15	2866
20	3746
25	4583
30	5386
35	6163
40	6918
45	7655
50	8375
55	9081
60	9774
65	10455
70	11126
75	11788
80	12441



T_{∞} [C]	$\dot{Q}_{\text{conv}}$ [W]
0	15658
0.5	14997
1	14336
1.5	13677
2	13018
2.5	12360
3	11702
3.5	11046
4	10390
4.5	9735
5	9081
5.5	8427
6	7774
6.5	7122
7	6471
7.5	5821
8	5171
8.5	4522
9	3874
9.5	3226
10	2579



12-43E Air flows over a flat plate. The local friction and heat transfer coefficients at intervals of 1 ft are to be determined and plotted against the distance from the leading edge.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and 60°F are (Table A-22E)

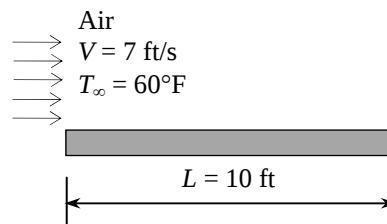
$$k = 0.01433 \text{ Btu/h.ft.}^\circ\text{F}$$

$$\nu = 0.1588 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$Pr = 0.7321$$

Analysis For the first 1 ft interval, the Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(7 \text{ ft/s})(1 \text{ ft})}{0.1588 \times 10^{-3} \text{ ft}^2/\text{s}} = 4.408 \times 10^4$$



which is less than the critical value of 5×10^5 . Therefore, the flow is laminar. The local Nusselt number is

$$Nu_x = \frac{hx}{k} = 0.332 Re_x^{0.5} Pr^{1/3} = 0.332(4.408 \times 10^4)^{0.5} (0.7321)^{1/3} = 62.82$$

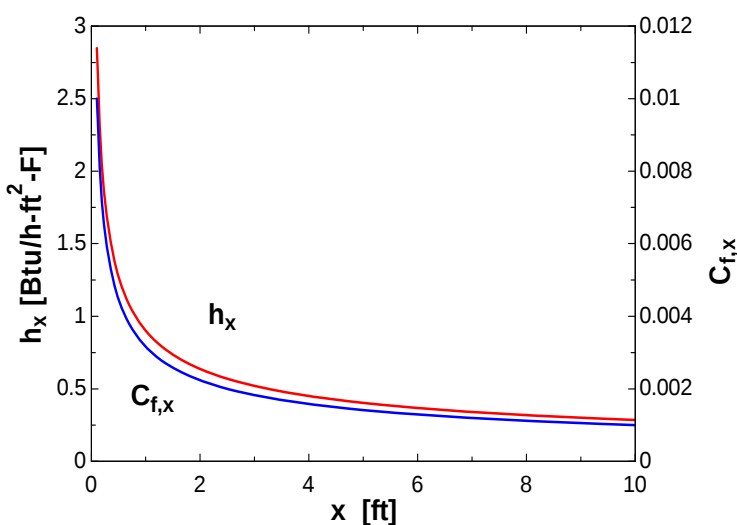
The local heat transfer and friction coefficients are

$$h_x = \frac{k}{x} Nu = \frac{0.01433 \text{ Btu/h.ft.}^\circ\text{F}}{1 \text{ ft}} (62.82) = 0.9002 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

$$C_{f,x} = \frac{0.664}{Re^{0.5}} = \frac{0.664}{(4.408 \times 10^4)^{0.5}} = 0.00316$$

We repeat calculations for all 1-ft intervals. The results are

x [ft]	h_x [Btu/h.ft ² ·F]	$C_{f,x}$
1	0.9005	0.003162
2	0.6367	0.002236
3	0.5199	0.001826
4	0.4502	0.001581
5	0.4027	0.001414
6	0.3676	0.001291
7	0.3404	0.001195
8	0.3184	0.001118
9	0.3002	0.001054
10	0.2848	0.001



12-44E EES Prob. 12-43E is reconsidered. The local friction and heat transfer coefficients along the plate are to be plotted against the distance from the leading edge.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$T_{\text{air}} = 60 \text{ [F]}$

$x = 10 \text{ [ft]}$

$\text{Vel} = 7 \text{ [ft/s]}$

"PROPERTIES"

$\text{Fluid\$} = \text{'air'}$

$k = \text{Conductivity}(\text{Fluid\$}, T = T_{\text{air}})$

$\text{Pr} = \text{Prandtl}(\text{Fluid\$}, T = T_{\text{air}})$

$\rho = \text{Density}(\text{Fluid\$}, T = T_{\text{air}}, P = 14.7)$

$\mu = \text{Viscosity}(\text{Fluid\$}, T = T_{\text{air}}) * \text{Convert}(\text{lbm/ft-h}, \text{lbm/ft-s})$

$\nu = \mu / \rho$

"ANALYSIS"

$\text{Re}_x = (\text{Vel} * x) / \nu$

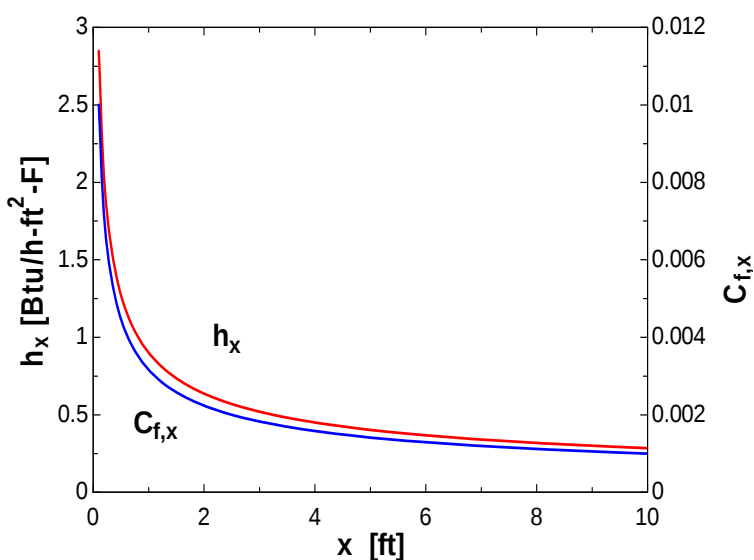
"Reynolds number is calculated to be smaller than the critical Re number. The flow is laminar."

$\text{Nusselt}_x = 0.332 * \text{Re}_x^{0.5} * \text{Pr}^{(1/3)}$

$h_x = k / x * \text{Nusselt}_x$

$C_{f,x} = 0.664 / \text{Re}_x^{0.5}$

x [ft]	h_x [Btu/h.ft ² .F]	$C_{f,x}$
0.1	2.848	0.01
0.2	2.014	0.007071
0.3	1.644	0.005774
0.4	1.424	0.005
0.5	1.273	0.004472
0.6	1.163	0.004083
0.7	1.076	0.00378
0.8	1.007	0.003536
0.9	0.9492	0.003333
1	0.9005	0.003162
...	...	...
...	...	...
9.1	0.2985	0.001048
9.2	0.2969	0.001043
9.3	0.2953	0.001037
9.4	0.2937	0.001031
9.5	0.2922	0.001026
9.6	0.2906	0.001021
9.7	0.2891	0.001015
9.8	0.2877	0.00101
9.9	0.2862	0.001005
10	0.2848	0.001



12-45 Air flows over the top and bottom surfaces of a thin, square plate. The flow regime and the total heat transfer rate are to be determined and the average gradients of the velocity and temperature at the surface are to be estimated.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible.

Properties The properties of air at the film temperature of $(T_s + T_\infty)/2 = (54 + 10)/2 = 32^\circ\text{C}$ are (Table A-22)

$$\begin{aligned}\rho &= 1.156 \text{ kg/m}^3 & \nu &= 1.627 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg}\cdot^\circ\text{C} & \text{Pr} &= 0.7276 \\ k &= 0.02603 \text{ W/m}\cdot^\circ\text{C}\end{aligned}$$

Analysis (a) The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(60 \text{ m/s})(0.5 \text{ m})}{1.627 \times 10^{-5} \text{ m}^2/\text{s}} = 1.844 \times 10^6$$

which is greater than the critical Reynolds number. Thus we have turbulent flow at the end of the plate.

(b) We use modified Reynolds analogy to determine the heat transfer coefficient and the rate of heat transfer

$$\tau_s = \frac{F}{A} = \frac{1.5 \text{ N}}{2(0.5 \text{ m})^2} = 3 \text{ N/m}^2$$

$$C_f = \frac{\tau_s}{0.5\rho V^2} = \frac{3 \text{ N/m}^2}{0.5(1.156 \text{ kg/m}^3)(60 \text{ m/s})^2} = 1.442 \times 10^{-3}$$

$$\frac{C_f}{2} = St \text{Pr}^{2/3} = \frac{Nu_L}{Re_L \text{Pr}} \text{Pr}^{2/3} = \frac{Nu_L}{Re_L \text{Pr}^{1/3}}$$

$$Nu = Re_L \text{Pr}^{1/3} \frac{C_f}{2} = (1.844 \times 10^6)(0.7276)^{1/3} \frac{(1.442 \times 10^{-3})}{2} = 1196$$

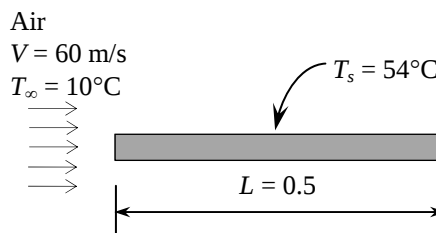
$$h = \frac{k}{L} Nu = \frac{0.02603 \text{ W/m}\cdot^\circ\text{C}}{0.5 \text{ m}} (1196) = 62.26 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (62.26 \text{ W/m}^2\cdot^\circ\text{C})[2 \times (0.5 \text{ m})^2](54 - 10)^\circ\text{C} = \mathbf{1370 \text{ W}}$$

(c) Assuming a uniform distribution of heat transfer and drag parameters over the plate, the average gradients of the velocity and temperature at the surface are determined to be

$$\tau_s = \mu \left. \frac{\partial u}{\partial y} \right|_0 \longrightarrow \left. \frac{\partial u}{\partial y} \right|_0 = \frac{\tau_s}{\rho \nu} = \frac{3 \text{ N/m}^2}{(1.156 \text{ kg/m}^3)(1.627 \times 10^{-5} \text{ m}^2/\text{s})} = \mathbf{1.60 \times 10^5 \text{ s}^{-1}}$$

$$h = \frac{-k \left. \frac{\partial T}{\partial y} \right|_0}{T_s - T_\infty} \longrightarrow \left. \frac{\partial T}{\partial y} \right|_0 = \frac{-h(T_s - T_\infty)}{k} = \frac{(62.26 \text{ W/m}^2\cdot^\circ\text{C})(54 - 10)^\circ\text{C}}{0.02603 \text{ W/m}\cdot^\circ\text{C}} = \mathbf{1.05 \times 10^5 \text{ }^\circ\text{C/m}}$$



12-46 Water flows over a large plate. The rate of heat transfer per unit width of the plate is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible.

Properties The properties of water at the film temperature of $(T_s + T_\infty)/2 = (10 + 43.3)/2 = 27^\circ\text{C}$ are (Table A-15)

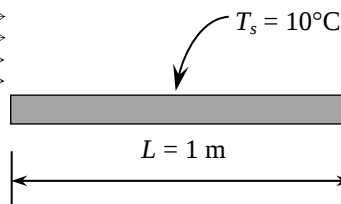
$$\rho = 996.6 \text{ kg/m}^3$$

$$k = 0.610 \text{ W/m}\cdot^\circ\text{C}$$

$$\mu = 0.854 \times 10^{-3} \text{ kg/m}\cdot\text{s}$$

$$Pr = 5.85$$

Water $\longrightarrow$
 $V = 30 \text{ cm/s}$ $\longrightarrow$
 $T_\infty = 43.3^\circ\text{C}$ $\longrightarrow$



Analysis (a) The Reynolds number is

$$Re_L = \frac{VL\rho}{\mu} = \frac{(0.3 \text{ m/s})(1.0 \text{ m})(996.6 \text{ kg/m}^3)}{0.854 \times 10^{-3} \text{ m}^2/\text{s}} = 3.501 \times 10^5$$

which is smaller than the critical Reynolds number. Thus we have laminar flow for the entire plate. The Nusselt number and the heat transfer coefficient are

$$Nu = 0.664 Re_L^{1/2} Pr^{1/3} = 0.664(3.501 \times 10^5)^{1/2} (5.85)^{1/3} = 707.9$$

$$h = \frac{k}{L} Nu = \frac{0.610 \text{ W/m}\cdot^\circ\text{C}}{1.0 \text{ m}} (707.9) = 431.8 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the rate of heat transfer per unit width of the plate is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) = (431.8 \text{ W/m}^2\cdot^\circ\text{C})(1 \text{ m})(1 \text{ m})(43.3 - 10)^\circ\text{C} = \mathbf{14,400 \text{ W}}$$

12-47 Mercury flows over a flat plate that is maintained at a specified temperature. The rate of heat transfer from the entire plate is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Atmospheric pressure is taken 1 atm.

Properties The properties of mercury at the film temperature of $(75+25)/2=50^\circ\text{C}$ are (Table A-20)

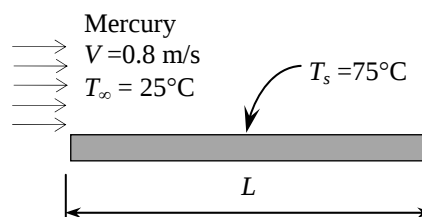
$$k = 8.83632 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.056 \times 10^{-7} \text{ m}^2/\text{s}$$

$$Pr = 0.0223$$

Analysis The local Nusselt number relation for liquid metals is given by Eq. 12-25 to be

$$Nu_x = \frac{h_x x}{k} = 0.565(Re_x Pr)^{1/2}$$



The average heat transfer coefficient for the entire surface can be determined from

$$h = \frac{1}{L} \int_0^L h_x dx$$

Substituting the local Nusselt number relation into the above equation and performing the integration we obtain

$$Nu = 1.13(Re_L Pr)^{1/2}$$

The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(0.8 \text{ m/s})(3 \text{ m})}{1.056 \times 10^{-7} \text{ m}^2/\text{s}} = 2.273 \times 10^7$$

Using the relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 1.13(Re_L Pr)^{1/2} = 1.13[(2.273 \times 10^7)(0.0223)]^{1/2} = 804.5$$

$$h = \frac{k}{L} Nu = \frac{8.83632 \text{ W/m}\cdot^\circ\text{C}}{3 \text{ m}} (804.5) = 2369 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A = wL = (2 \text{ m})(3 \text{ m}) = 6 \text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) = (2369 \text{ W/m}^2\cdot^\circ\text{C})(6 \text{ m}^2)(75 - 25)^\circ\text{C} = 710,800 \text{ W} = \mathbf{710.8 \text{ kW}}$$

12-48 Ambient air flows over parallel plates of a solar collector that is maintained at a specified temperature. The rates of convection heat transfer from the first and third plate are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Atmospheric pressure is taken 1 atm.

Properties The properties of air at the film temperature of $(15+10)/2=12.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02458 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.448 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7330$$

Analysis (a) The critical length of the plate is first determined to be

$$x_{cr} = \frac{Re_{cr} \nu}{V} = \frac{(5 \times 10^5)(1.448 \times 10^{-5} \text{ m}^2/\text{s})}{2 \text{ m/s}} = 3.62 \text{ m}$$

Therefore, both plates are under laminar flow. The Reynolds number for the first plate is

$$Re_1 = \frac{VL_1}{\nu} = \frac{(2 \text{ m/s})(1 \text{ m})}{1.448 \times 10^{-5} \text{ m}^2/\text{s}} = 1.381 \times 10^5$$

Using the relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu_1 = 0.664 Re_1^{1/2} Pr^{1/3} = 0.664(1.381 \times 10^5)^{1/2} (0.7330)^{1/3} = 222.5$$

$$h_1 = \frac{k}{L_1} Nu = \frac{0.02458 \text{ W/m}\cdot^\circ\text{C}}{1 \text{ m}} (222.5) = 5.47 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A = wL = (4 \text{ m})(1 \text{ m}) = 4 \text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) = (5.47 \text{ W/m}^2\cdot^\circ\text{C})(4 \text{ m}^2)(15 - 10)^\circ\text{C} = \mathbf{109 \text{ W}}$$

(b) Repeating the calculations for the second and third plates,

$$Re_2 = \frac{VL_2}{\nu} = \frac{(2 \text{ m/s})(2 \text{ m})}{1.448 \times 10^{-5} \text{ m}^2/\text{s}} = 2.762 \times 10^5$$

$$Nu_2 = 0.664 Re_2^{1/2} Pr^{1/3} = 0.664(2.762 \times 10^5)^{1/2} (0.7330)^{1/3} = 314.7$$

$$h_2 = \frac{k}{L_2} Nu = \frac{0.02458 \text{ W/m}\cdot^\circ\text{C}}{2 \text{ m}} (314.7) = 3.87 \text{ W/m}^2\cdot^\circ\text{C}$$

$$Re_3 = \frac{VL_3}{\nu} = \frac{(2 \text{ m/s})(3 \text{ m})}{1.448 \times 10^{-5} \text{ m}^2/\text{s}} = 4.144 \times 10^5$$

$$Nu_3 = 0.664 Re_3^{1/2} Pr^{1/3} = 0.664(4.144 \times 10^5)^{1/2} (0.7330)^{1/3} = 385.4$$

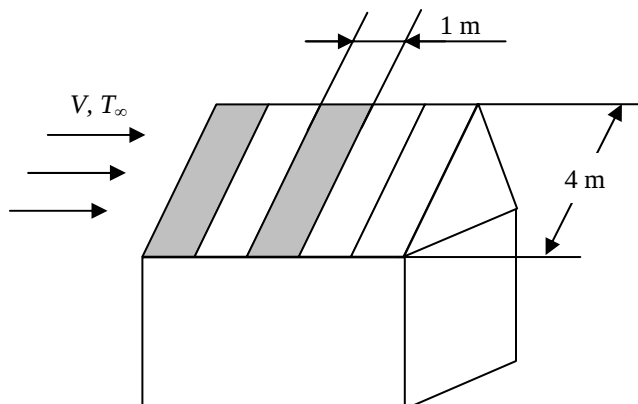
$$h_3 = \frac{k}{L_3} Nu = \frac{0.02458 \text{ W/m}\cdot^\circ\text{C}}{3 \text{ m}} (385.4) = 3.16 \text{ W/m}^2\cdot^\circ\text{C}$$

Then

$$h_{2-3} = \frac{h_3 L_3 - h_2 L_2}{L_3 - L_2} = \frac{3.16 \times 3 - 3.87 \times 2}{3 - 2} = 1.74 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat loss from the third plate is

$$\dot{Q} = hA(T_s - T_\infty) = (1.74 \text{ W/m}^2\cdot^\circ\text{C})(4 \text{ m}^2)(15 - 10)^\circ\text{C} = \mathbf{34.8 \text{ W}}$$



12-49 A car travels at a velocity of 80 km/h. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Air is an ideal gas with constant properties. 4 The flow is turbulent over the entire surface because of the constant agitation of the engine block.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (100 + 20)/2 = 60^\circ\text{C}$ are (Table A-22)

$$k = 0.02808 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.896 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7202$$

Analysis Air flows parallel to the 0.4 m side. The Reynolds number in this case is

$$Re_L = \frac{V_\infty L}{\nu} = \frac{[(80 \times 1000 / 3600) \text{ m/s}](0.8 \text{ m})}{1.896 \times 10^{-5} \text{ m}^2/\text{s}} = 9.376 \times 10^5$$

which is greater than the critical Reynolds number and thus the flow is laminar + turbulent. But the flow is assumed to be turbulent over the entire surface because of the constant agitation of the engine block. Using the proper relations, the Nusselt number, the heat transfer coefficient, and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.037 Re_L^{0.8} Pr^{1/3} = 0.037(9.376 \times 10^5)^{0.8} (0.7202)^{1/3} = 1988$$

$$h = \frac{k}{L} Nu = \frac{0.02808 \text{ W/m}\cdot^\circ\text{C}}{0.8 \text{ m}} (1988) = 69.78 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = wL = (0.8 \text{ m})(0.4 \text{ m}) = 0.32 \text{ m}^2$$

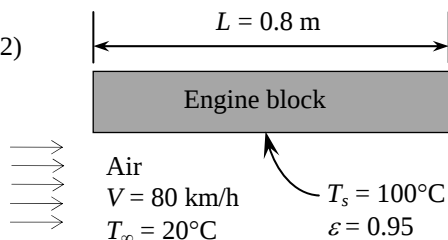
$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (69.78 \text{ W/m}^2\cdot^\circ\text{C})(0.32 \text{ m}^2)(100 - 20)^\circ\text{C} = \mathbf{1786 \text{ W}}$$

The radiation heat transfer from the same surface is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.95)(0.32 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(100 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] = \mathbf{198 \text{ W}} \end{aligned}$$

Then the total rate of heat transfer from that surface becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = (1786 + 198) \text{ W} = \mathbf{1984 \text{ W}}$$



12-50 Air flows on both sides of a continuous sheet of plastic. The rate of heat transfer from the plastic sheet is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90 + 30)/2 = 60^\circ\text{C}$ are (Table A-22)

$$\begin{aligned}\rho &= 1.059 \text{ kg/m}^3 \\ k &= 0.02808 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= 1.896 \times 10^{-5} \text{ m}^2/\text{s} \\ Pr &= 0.7202\end{aligned}$$

Analysis The width of the cooling section is first determined from

$$W = V\Delta t = [(15 / 60) \text{ m/s}](2 \text{ s}) = 0.5 \text{ m}$$

The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(3 \text{ m/s})(1.2 \text{ m})}{1.896 \times 10^{-5} \text{ m}^2/\text{s}} = 1.899 \times 10^5$$

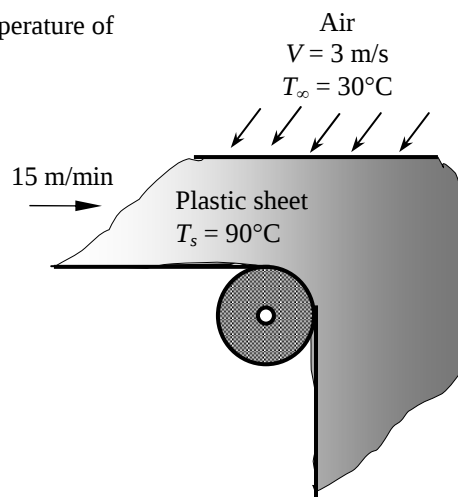
which is less than the critical Reynolds number. Thus the flow is laminar. Using the proper relation in laminar flow for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664(1.899 \times 10^5)^{0.5} (0.7202)^{1/3} = 259.3$$

$$h = \frac{k}{L} Nu = \frac{0.02808 \text{ W/m}\cdot^\circ\text{C}}{1.2 \text{ m}} (259.3) = 6.07 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = 2LW = 2(1.2 \text{ m})(0.5 \text{ m}) = 1.2 \text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (6.07 \text{ W/m}^2\cdot^\circ\text{C})(1.2 \text{ m}^2)(90 - 30)^\circ\text{C} = \mathbf{437 \text{ W}}$$



12-51 The top surface of the passenger car of a train in motion is absorbing solar radiation. The equilibrium temperature of the top surface is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation heat exchange with the surroundings is negligible. 4 Air is an ideal gas with constant properties.

Properties The properties of air at 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7282$$

Analysis The rate of convection heat transfer from the top surface of the car to the air must be equal to the solar radiation absorbed by the same surface in order to reach steady operation conditions. The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{[70 \times 1000/3600] \text{ m/s}(8 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 9.674 \times 10^6$$

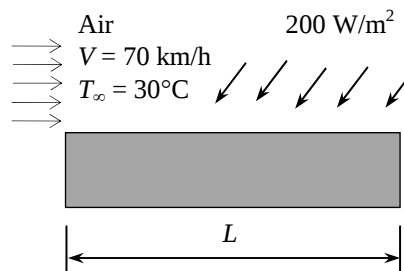
which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(9.674 \times 10^6)^{0.8} - 871](0.7282)^{1/3} = 1.212 \times 10^4$$

$$h = \frac{k}{L} Nu = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{8 \text{ m}} (1.212 \times 10^4) = 39.21 \text{ W/m}^2\cdot^\circ\text{C}$$

The equilibrium temperature of the top surface is then determined by taking convection and radiation heat fluxes to be equal to each other

$$\dot{q}_{rad} = \dot{q}_{conv} = h(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}_{conv}}{h} = 30^\circ\text{C} + \frac{200 \text{ W/m}^2}{39.21 \text{ W/m}^2\cdot^\circ\text{C}} = 35.1^\circ\text{C}$$



12-52 EES Prob. 12-51 is reconsidered. The effects of the train velocity and the rate of absorption of solar radiation on the equilibrium temperature of the top surface of the car are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

Vel=70 [km/h]

w=2.8 [m]

L=8 [m]

q_dot_rad=200 [W/m^2]

T_infinity=30 [C]

"PROPERTIES"

Fluid\$='air'

k=Conductivity(Fluid\$, T=T_film)

Pr=Prandtl(Fluid\$, T=T_film)

rho=Density(Fluid\$, T=T_film, P=101.3)

mu=Viscosity(Fluid\$, T=T_film)

nu=mu/rho

T_film=1/2*(T_s+T_infinity)

"ANALYSIS"

Re=(Vel*Convert(km/h, m/s)*L)/nu

"Reynolds number is greater than the critical Reynolds number. We use combined laminar and turbulent flow relation for Nusselt number"

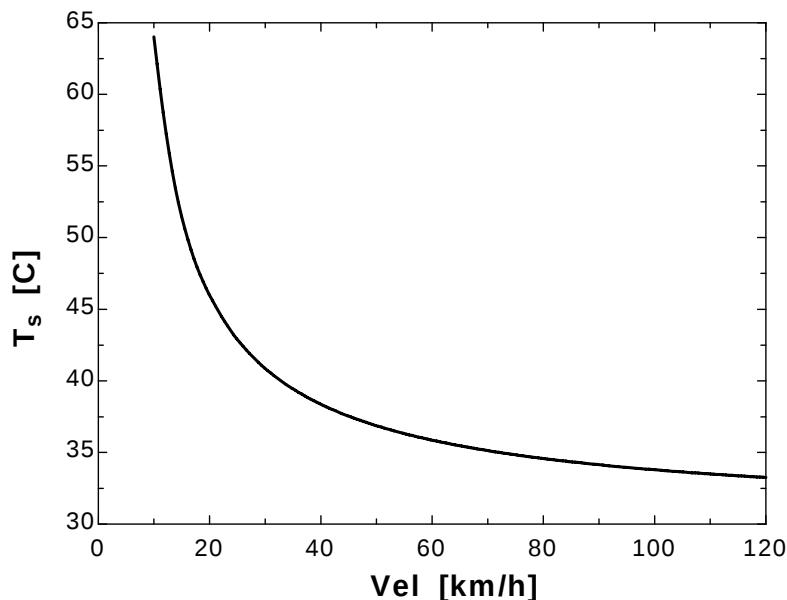
Nusselt=(0.037*Re^0.8-871)*Pr^(1/3)

h=k/L*Nusselt

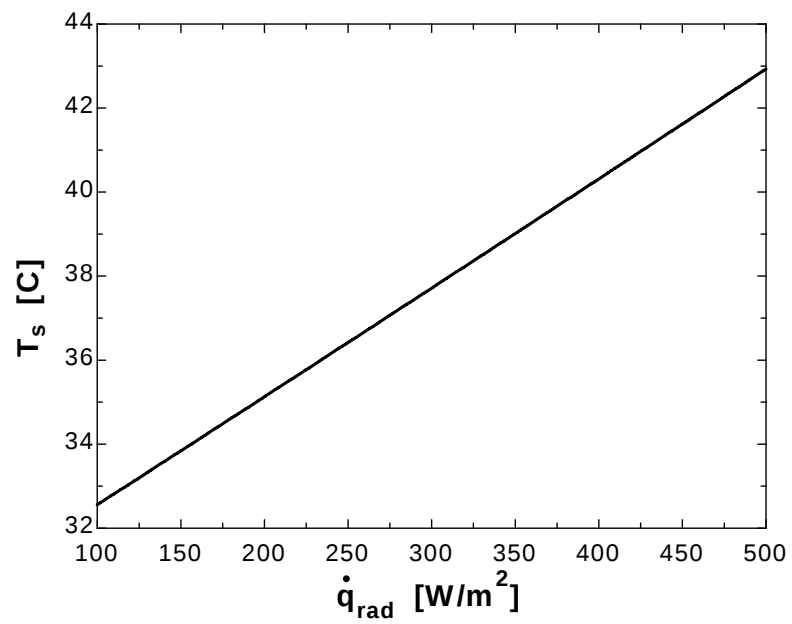
q_dot_conv=h*(T_s-T_infinity)

q_dot_conv=q_dot_rad

Vel [km/h]	T _s [C]
10	64.01
15	51.44
20	45.99
25	42.89
30	40.86
35	39.43
40	38.36
45	37.53
50	36.86
55	36.32
60	35.86
65	35.47
70	35.13
75	34.83
80	34.58
85	34.35
90	34.14
95	33.96
100	33.79
105	33.64
110	33.5
115	33.37
120	33.25



$\dot{Q}_{\text{rad}}$ [W/m ²]	T_s [C]
100	32.56
125	33.2
150	33.84
175	34.48
200	35.13
225	35.77
250	36.42
275	37.07
300	37.71
325	38.36
350	39.01
375	39.66
400	40.31
425	40.97
450	41.62
475	42.27
500	42.93



12-53 A circuit board is cooled by air. The surface temperatures of the electronic components at the leading edge and the end of the board are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Any heat transfer from the back surface of the board is disregarded. 5 Air is an ideal gas with constant properties.

Properties Assuming the film temperature to be approximately 35°C , the properties of air are evaluated at this temperature to be (Table A-22)

$$k = 0.0265 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.655 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7268$$

Analysis (a) The convection heat transfer coefficient at the leading edge approaches infinity, and thus the surface temperature there must approach the air temperature, which is 20°C .

(b) The Reynolds number is

$$Re_x = \frac{Vx}{\nu} = \frac{(6 \text{ m/s})(0.15 \text{ m})}{1.655 \times 10^{-5} \text{ m}^2/\text{s}} = 5.438 \times 10^4$$

which is less than the critical Reynolds number but we assume the flow to be turbulent since the electronic components are expected to act as turbulators. Using the Nusselt number uniform heat flux, the local heat transfer coefficient at the end of the board is determined to be

$$Nu_x = \frac{h_x x}{k} = 0.0308 Re_x^{0.8} Pr^{1/3} = 0.0308 (5.438 \times 10^4)^{0.8} (0.7268)^{1/3} = 170.1$$

$$h_x = \frac{k_x}{x} Nu_x = \frac{0.02625 \text{ W/m}\cdot^\circ\text{C}}{0.15 \text{ m}} (170.1) = 29.77 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x (T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20^\circ\text{C} + \frac{(20 \text{ W})/(0.15 \text{ m})^2}{29.77 \text{ W/m}^2\cdot^\circ\text{C}} = 49.9^\circ\text{C}$$

Discussion The heat flux can also be determined approximately using the relation for isothermal surfaces,

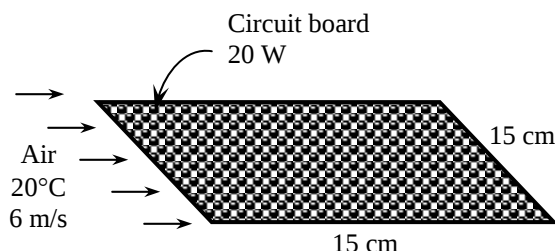
$$Nu_x = \frac{h_x x}{k} = 0.0296 Re_x^{0.8} Pr^{1/3} = 0.0296 (5.438 \times 10^4)^{0.8} (0.7268)^{1/3} = 163.5$$

$$h_x = \frac{k_x}{x} Nu_x = \frac{0.02625 \text{ W/m}\cdot^\circ\text{C}}{0.15 \text{ m}} (163.5) = 28.61 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x (T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20^\circ\text{C} + \frac{(20 \text{ W})/(0.15 \text{ m})^2}{28.61 \text{ W/m}^2\cdot^\circ\text{C}} = 51.1^\circ\text{C}$$

Note that the two results are close to each other.



12-54 Laminar flow of a fluid over a flat plate is considered. The change in the drag force and the rate of heat transfer are to be determined when the free-stream velocity of the fluid is doubled.

Analysis For the laminar flow of a fluid over a flat plate maintained at a constant temperature the drag force is given by

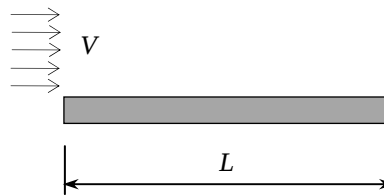
$$F_{D1} = C_f A_s \frac{\rho V^2}{2} \quad \text{where} \quad C_f = \frac{1.33}{\text{Re}^{0.5}}$$

Therefore

$$F_{D1} = \frac{1.33}{\text{Re}^{0.5}} A_s \frac{\rho V^2}{2}$$

Substituting Reynolds number relation, we get

$$F_{D1} = \frac{1.33}{\left(\frac{VL}{\nu}\right)^{0.5}} A_s \frac{\rho V^2}{2} = 0.664 V^{3/2} A_s \frac{\nu^{0.5}}{L^{0.5}}$$



When the free-stream velocity of the fluid is doubled, the new value of the drag force on the plate becomes

$$F_{D2} = \frac{1.33}{\left(\frac{(2V)L}{\nu}\right)^{0.5}} A_s \frac{\rho (2V)^2}{2} = 0.664 (2V)^{3/2} A_s \frac{\nu^{0.5}}{L^{0.5}}$$

The ratio of drag forces corresponding to V and $2V$ is

$$\frac{F_{D2}}{F_{D1}} = \frac{(2V)^{3/2}}{V^{3/2}} = 2^{3/2}$$

We repeat similar calculations for heat transfer rate ratio corresponding to V and $2V$

$$\begin{aligned} \dot{Q}_1 &= h A_s (T_s - T_\infty) = \left(\frac{k}{L} Nu\right) A_s (T_s - T_\infty) = \left(\frac{k}{L}\right) (0.664 \text{Re}^{0.5} \text{Pr}^{1/3}) A_s (T_s - T_\infty) \\ &= \frac{k}{L} 0.664 \left(\frac{VL}{\nu}\right)^{0.5} \text{Pr}^{1/3} A_s (T_s - T_\infty) \\ &= 0.664 V^{0.5} \frac{k}{L^{0.5} \nu^{0.5}} \text{Pr}^{1/3} A_s (T_s - T_\infty) \end{aligned}$$

When the free-stream velocity of the fluid is doubled, the new value of the heat transfer rate between the fluid and the plate becomes

$$\dot{Q}_2 = 0.664 (2V)^{0.5} \frac{k}{L^{0.5} \nu^{0.5}} \text{Pr}^{1/3} A_s (T_s - T_\infty)$$

Then the ratio is

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2V)^{0.5}}{V^{0.5}} = 2^{0.5} = \sqrt{2}$$

12-55E A refrigeration truck is traveling at 55 mph. The average temperature of the outer surface of the refrigeration compartment of the truck is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties. 5 The local atmospheric pressure is 1 atm.

Properties Assuming the film temperature to be approximately 80°F, the properties of air at this temperature and 1 atm are (Table A-22E)

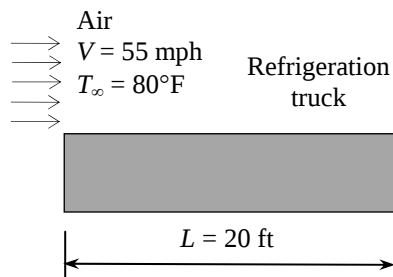
$$k = 0.01481 \text{ Btu/h.ft.}^\circ\text{F}$$

$$\nu = 1.697 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$Pr = 0.7290$$

Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{[55 \times 5280/3600] \text{ ft/s}](20 \text{ ft})}{1.697 \times 10^{-4} \text{ ft}^2/\text{s}} = 9.507 \times 10^6$$



We assume the air flow over the entire outer surface to be turbulent. Therefore using the proper relation in turbulent flow for Nusselt number, the average heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.037 Re_L^{0.8} Pr^{1/3} = 0.037(9.507 \times 10^6)^{0.8} (0.7290)^{1/3} = 1.273 \times 10^4$$

$$h = \frac{k}{L} Nu = \frac{0.01481 \text{ Btu/h.ft.}^\circ\text{F}}{20 \text{ ft}} (1.273 \times 10^4) = 9.428 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

Since the refrigeration system is operated at half the capacity, we will take half of the heat removal rate

$$\dot{Q} = \frac{(600 \times 60) \text{ Btu/h}}{2} = 18,000 \text{ Btu/h}$$

The total heat transfer surface area and the average surface temperature of the refrigeration compartment of the truck are determined from

$$A = 2[(20 \text{ ft})(9 \text{ ft}) + (20 \text{ ft})(8 \text{ ft}) + (9 \text{ ft})(8 \text{ ft})] = 824 \text{ ft}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) \longrightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 80^\circ\text{F} - \frac{18,000 \text{ Btu/h}}{(9.428 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(824 \text{ ft}^2)} = 77.7^\circ\text{F}$$

12-56 Solar radiation is incident on the glass cover of a solar collector. The total rate of heat loss from the collector, the collector efficiency, and the temperature rise of water as it flows through the collector are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Heat exchange on the back surface of the absorber plate is negligible. 4 Air is an ideal gas with constant properties. 5 The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of $(35 + 25) / 2 = 30^\circ\text{C}$ are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7282$$

Analysis (a) Assuming wind flows across 2 m surface, the Reynolds number is determined from

$$Re_L = \frac{VL}{\nu} = \frac{(30 \times 1000 / 3600 \text{ m/s})(2 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 1.036 \times 10^6$$

which is greater than the critical Reynolds number. Using the Nusselt number relation for combined laminar and turbulent flow, the average heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = (0.037 Re^{0.8} - 871) Pr^{1/3} = [0.037(1.036 \times 10^6)^{0.8} - 871](0.7282)^{1/3} = 1378$$

$$h = \frac{k}{L} Nu = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{2 \text{ m}} (1378) = 17.83 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the rate of heat loss from the collector by convection is

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (17.83 \text{ W/m}^2\cdot^\circ\text{C})(2 \times 1.2 \text{ m}^2)(35 - 25)^\circ\text{C} = 427.9 \text{ W}$$

The rate of heat loss from the collector by radiation is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.90)(2 \times 1.2 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot^\circ\text{C}) [(35 + 273 \text{ K})^4 - (-40 + 273 \text{ K})^4] \\ &= 741.2 \text{ W} \end{aligned}$$

and

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 427.9 + 741.2 = \mathbf{1169 \text{ W}}$$

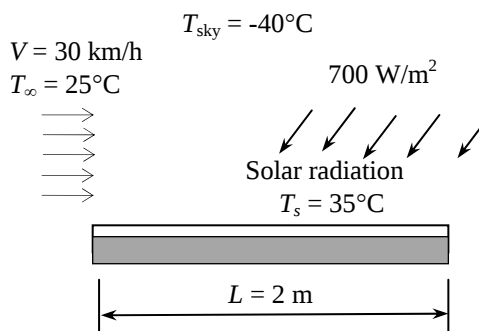
(b) The net rate of heat transferred to the water is

$$\begin{aligned} \dot{Q}_{net} &= \dot{Q}_{in} - \dot{Q}_{out} = \alpha AI - \dot{Q}_{out} \\ &= (0.88)(2 \times 1.2 \text{ m}^2)(700 \text{ W/m}^2) - 1169 \text{ W} \\ &= 1478 - 1169 = 309 \text{ W} \end{aligned}$$

$$\eta_{collector} = \frac{\dot{Q}_{net}}{\dot{Q}_{in}} = \frac{309 \text{ W}}{1478 \text{ W}} = \mathbf{0.209}$$

(c) The temperature rise of water as it flows through the collector is

$$\dot{Q}_{net} = \dot{m} c_p \Delta T \longrightarrow \Delta T = \frac{\dot{Q}_{net}}{\dot{m} c_p} = \frac{309.4 \text{ W}}{(1/60 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})} = \mathbf{4.44^\circ\text{C}}$$



12-57 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum free-stream velocity that the fan should provide to avoid overheating is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) 5 Air is an ideal gas with constant properties. 6 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (60 + 25)/2 = 42.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02681 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.726 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7248$$

Analysis The total heat transfer surface area for this finned surface is

$$A_{s,\text{finned}} = (2 \times 7)(0.1 \text{ m})(0.005 \text{ m}) = 0.007 \text{ m}^2$$

$$A_{s,\text{unfinned}} = (0.1 \text{ m})(0.062 \text{ m}) - 7 \times (0.002 \text{ m})(0.1 \text{ m}) = 0.0048 \text{ m}^2$$

$$A_{s,\text{total}} = A_{s,\text{finned}} + A_{s,\text{unfinned}} = 0.007 \text{ m}^2 + 0.0048 \text{ m}^2 = 0.0118 \text{ m}^2$$

The convection heat transfer coefficient can be determined from Newton's law of cooling relation for a finned surface.

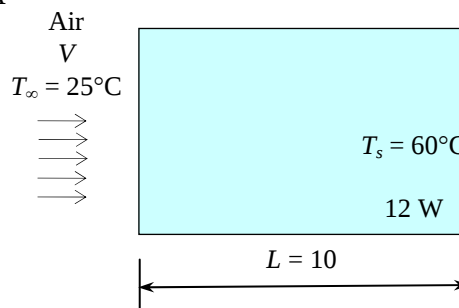
$$\dot{Q} = \eta h A_s (T_\infty - T_s) \longrightarrow h = \frac{\dot{Q}}{\eta A_s (T_\infty - T_s)} = \frac{12 \text{ W}}{(1)(0.0118 \text{ m}^2)(60 - 25)^\circ\text{C}} = 29.06 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Starting from heat transfer coefficient, Nusselt number, Reynolds number and finally free-stream velocity will be determined. We assume the flow is laminar over the entire finned surface of the transformer.

$$Nu = \frac{hL}{k} = \frac{(29.06 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1 \text{ m})}{0.02681 \text{ W/m}\cdot^\circ\text{C}} = 108.4$$

$$Nu = 0.664 Re_L^{0.5} Pr^{1/3} \longrightarrow Re_L = \frac{Nu^2}{0.664^2 Pr^{2/3}} = \frac{(108.4)^2}{(0.664)^2 (0.7248)^{2/3}} = 3.302 \times 10^4$$

$$Re_L = \frac{VL}{\nu} \longrightarrow V = \frac{Re_L \nu}{L} = \frac{(3.302 \times 10^4)(1.726 \times 10^{-5} \text{ m}^2/\text{s})}{0.1 \text{ m}} = \mathbf{5.70 \text{ m/s}}$$



12-58 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum free-stream velocity that the fan should provide to avoid overheating is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) 4 Air is an ideal gas with constant properties. 5 The local atmospheric pressure is 1 atm.

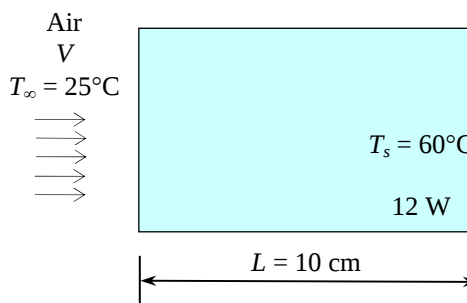
Properties The properties of air at the film temperature of $(T_s + T_\infty)/2 = (60 + 25)/2 = 42.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02681 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.726 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7248$$

Analysis We first need to determine radiation heat transfer rate. Note that we will use the base area and we assume the temperature of the surrounding surfaces are at the same temperature with the air ($T_{surr} = 25^\circ\text{C}$)



$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.90)[(0.1 \text{ m})(0.062 \text{ m})](5.67 \times 10^{-8} \text{ W/m}^2 \cdot ^\circ\text{C})[(60 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] \\ &= 1.4 \text{ W}\end{aligned}$$

The heat transfer rate by convection will be 1.4 W less than total rate of heat transfer from the transformer. Therefore

$$\dot{Q}_{conv} = \dot{Q}_{total} - \dot{Q}_{rad} = 12 - 1.4 = 10.6 \text{ W}$$

The total heat transfer surface area for this finned surface is

$$\begin{aligned}A_{s,finned} &= (2 \times 7)(0.1 \text{ m})(0.005 \text{ m}) = 0.007 \text{ m}^2 \\ A_{s,unfinned} &= (0.1 \text{ m})(0.062 \text{ m}) - 7 \times (0.002 \text{ m})(0.1 \text{ m}) = 0.0048 \text{ m}^2 \\ A_{s,total} &= A_{s,finned} + A_{s,unfinned} = 0.007 \text{ m}^2 + 0.0048 \text{ m}^2 = 0.0118 \text{ m}^2\end{aligned}$$

The convection heat transfer coefficient can be determined from Newton's law of cooling relation for a finned surface.

$$\dot{Q}_{conv} = \eta h A_s (T_\infty - T_s) \longrightarrow h = \frac{\dot{Q}_{conv}}{\eta A_s (T_\infty - T_s)} = \frac{10.6 \text{ W}}{(1)(0.0118 \text{ m}^2)(60 - 25)^\circ\text{C}} = 25.67 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Starting from heat transfer coefficient, Nusselt number, Reynolds number and finally free-stream velocity will be determined. We assume the flow is laminar over the entire finned surface of the transformer.

$$\begin{aligned}Nu &= \frac{hL}{k} = \frac{(25.67 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1 \text{ m})}{0.02681 \text{ W/m}\cdot^\circ\text{C}} = 95.73 \\ Nu &= 0.664 Re_L^{0.5} Pr^{1/3} \longrightarrow Re_L = \frac{Nu^2}{0.664^2 Pr^{2/3}} = \frac{(95.73)^2}{(0.664)^2 (0.7248)^{2/3}} = 2.576 \times 10^4 \\ Re_L &= \frac{VL}{\nu} \longrightarrow V = \frac{Re_L \nu}{L} = \frac{(2.576 \times 10^4)(1.726 \times 10^{-5} \text{ m}^2/\text{s})}{0.1 \text{ m}} = \mathbf{4.45 \text{ m/s}}\end{aligned}$$

12-59 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Heat transfer from the back side of the plate is negligible. 5 Air is an ideal gas with constant properties. 6 The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of $(T_s + T_\infty)/2 = (65 + 35)/2 = 50^\circ\text{C}$ are (Table A-22)

$$k = 0.02735 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7228$$

Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(4 \text{ m/s})(0.25 \text{ m})}{1.798 \times 10^{-5} \text{ m}^2/\text{s}} = 55,617$$

which is less than the critical Reynolds number. Thus the flow is laminar. Using the proper relation in laminar flow for Nusselt number, heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664(55,617)^{0.5} (0.7228)^{1/3} = 140.5$$

$$h = \frac{k}{L} Nu = \frac{0.02735 \text{ W/m}\cdot^\circ\text{C}}{0.25 \text{ m}} (140.5) = 15.37 \text{ W/m}^2\cdot^\circ\text{C}$$

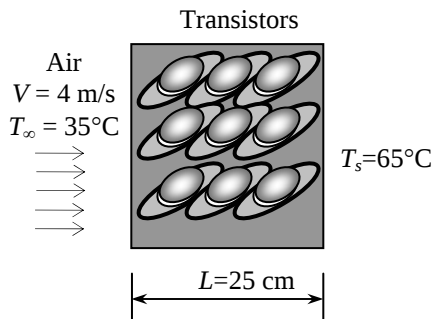
$$A_s = wL = (0.25 \text{ m})(0.25 \text{ m}) = 0.0625 \text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (15.37 \text{ W/m}^2\cdot^\circ\text{C})(0.0625 \text{ m}^2)(65 - 35)^\circ\text{C} = 28.83 \text{ W}$$

Considering that each transistor dissipates 6 W of power, the number of transistors that can be placed on this plate becomes

$$n = \frac{28.8 \text{ W}}{6 \text{ W}} = 4.8 \longrightarrow \mathbf{4}$$

This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.



12-60 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Heat transfer from the backside of the plate is negligible. 5 Air is an ideal gas with constant properties. 6 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65 + 35)/2 = 50^\circ\text{C}$ are (Table A-22)

$$k = 0.02735 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7228$$

Note that the atmospheric pressure will only affect the kinematic viscosity. The atmospheric pressure in atm is

$$P = (83.4 \text{ kPa}) \frac{1 \text{ atm}}{101.325 \text{ kPa}} = 0.823 \text{ atm}$$

The kinematic viscosity at this atmospheric pressure will be

$$\nu = (1.798 \times 10^{-5} \text{ m}^2/\text{s}) / 0.823 = 2.184 \times 10^{-5} \text{ m}^2/\text{s}$$

Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(4 \text{ m/s})(0.25 \text{ m})}{2.184 \times 10^{-5} \text{ m}^2/\text{s}} = 4.579 \times 10^4$$

which is less than the critical Reynolds number. Thus the flow is laminar. Using the proper relation in laminar flow for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664(4.579 \times 10^4)^{0.5} (0.7228)^{1/3} = 127.5$$

$$h = \frac{k}{L} Nu = \frac{0.02735 \text{ W/m}\cdot^\circ\text{C}}{0.25 \text{ m}} (127.5) = 13.95 \text{ W/m}^2\cdot^\circ\text{C}$$

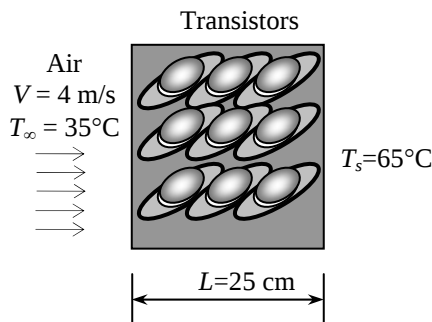
$$A_s = wL = (0.25 \text{ m})(0.25 \text{ m}) = 0.0625 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = hA_s(T_\infty - T_s) = (13.95 \text{ W/m}^2\cdot^\circ\text{C})(0.0625 \text{ m}^2)(65 - 35)^\circ\text{C} = 26.2 \text{ W}$$

Considering that each transistor dissipates 6 W of power, the number of transistors that can be placed on this plate becomes

$$n = \frac{26.2 \text{ W}}{6 \text{ W}} = 4.4 \longrightarrow 4$$

This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.



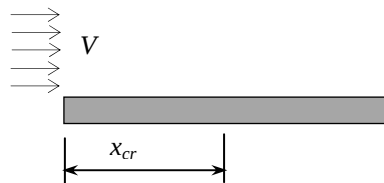
12-61 Air is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

Assumptions 1 The flow is steady and incompressible. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Air is an ideal gas. 4 The surface of the plate is smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184 \text{ kg/m}^3$ and $\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$ (Table A-22).

Analysis The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,

$$Re_{cr} = \frac{Vx_{cr}}{\nu} \rightarrow x_{cr} = \frac{\nu Re_{cr}}{V} = \frac{(1.562 \times 10^{-5} \text{ m}^2/\text{s})(5 \times 10^5)}{8 \text{ m/s}} = \mathbf{0.976 \text{ m}}$$



The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation,

$$\delta_x = \frac{5x}{Re_x^{1/2}} \rightarrow \delta_{cr} = \frac{5x_{cr}}{Re_{cr}^{1/2}} = \frac{5(0.976 \text{ m})}{(5 \times 10^5)^{1/2}} = 0.006903 \text{ m} = \mathbf{0.69 \text{ cm}}$$

Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

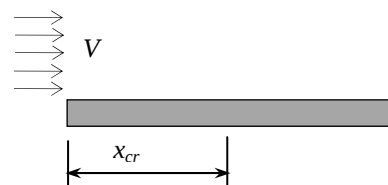
12-62 Water is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

Assumptions 1 The flow is steady and incompressible. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 The surface of the plate is smooth.

Properties The density and dynamic viscosity of water at 1 atm and 25°C are $\rho = 997 \text{ kg/m}^3$ and $\mu = 0.891 \times 10^{-3} \text{ kg/m} \cdot \text{s}$ (Table A-15).

Analysis The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,

$$Re_{cr} = \frac{\rho V x_{cr}}{\mu} \rightarrow x_{cr} = \frac{\mu Re_{cr}}{\rho V} = \frac{(0.891 \times 10^{-3} \text{ kg/m} \cdot \text{s})(5 \times 10^5)}{(997 \text{ kg/m}^3)(8 \text{ m/s})} = 0.056 \text{ m} = \mathbf{5.6 \text{ cm}}$$



The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation,

$$\delta_x = \frac{5x}{Re_x^{1/2}} \rightarrow \delta_{cr} = \frac{5x_{cr}}{Re_{cr}^{1/2}} = \frac{5(0.056 \text{ m})}{(5 \times 10^5)^{1/2}} = 0.00040 \text{ m} = \mathbf{0.4 \text{ mm}}$$

Therefore, the flow becomes turbulent after about 5 cm from the leading edge of the plate, and the thickness of the boundary layer at that location is 0.4 mm.

Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

12-63 The weight of a thin flat plate exposed to air flow on both sides is balanced by a counterweight. The mass of the counterweight that needs to be added in order to balance the plate is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. **3** Air is an ideal gas. **4** The surfaces of the plate are smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184 \text{ kg/m}^3$ and $\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$ (Table A-22).

Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{(10 \text{ m/s})(0.4 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 2.561 \times 10^5$$

which is less than the critical Reynolds number of 5×10^5 . Therefore the flow is laminar. The average friction coefficient, drag force and the corresponding mass are

$$C_f = \frac{1.33}{Re_L^{0.5}} = \frac{1.33}{(2.561 \times 10^5)^{0.5}} = 0.002628$$

$$F_D = C_f A_s \frac{\rho V^2}{2}$$

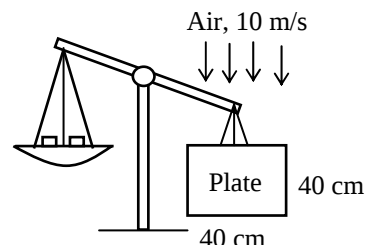
$$= (0.002628)[(2 \times 0.4 \times 0.4) \text{ m}^2] \frac{(1.184 \text{ kg/m}^3)(10 \text{ m/s})^2}{2} = 0.0498 \text{ kg} \cdot \text{m/s}^2 = 0.0498 \text{ N}$$

The mass whose weight is 0.0497 N is

$$m = \frac{F_D}{g} = \frac{0.0498 \text{ kg} \cdot \text{m/s}^2}{9.81 \text{ m/s}^2} = 0.00508 \text{ kg} = \mathbf{5.08 \text{ g}}$$

Therefore, the mass of the counterweight must be 5 g to counteract the drag force acting on the plate.

Discussion Note that the apparatus described in this problem provides a convenient mechanism to measure drag force and thus drag coefficient.



Flow across Cylinders and Spheres

12-64C For the laminar flow, the heat transfer coefficient will be the highest at the stagnation point which corresponds to $\theta \approx 0^\circ$. In turbulent flow, on the other hand, it will be highest when θ is between 90° and 120° .

12-65C Turbulence moves the fluid separation point further back on the rear of the body, reducing the size of the wake, and thus the magnitude of the pressure drag (which is the dominant mode of drag). As a result, the drag coefficient suddenly drops. In general, turbulence increases the drag coefficient for flat surfaces, but the drag coefficient usually remains constant at high Reynolds numbers when the flow is turbulent.

12-66C Friction drag is due to the shear stress at the surface whereas the pressure drag is due to the pressure differential between the front and back sides of the body when a wake is formed in the rear.

12-67C Flow separation in flow over a cylinder is delayed in turbulent flow because of the extra mixing due to random fluctuations and the transverse motion.

12-68 A steam pipe is exposed to windy air. The rate of heat loss from the steam is to be determined. ✓

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90+7)/2 = 48.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02724 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.784 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7232$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(50 \text{ km/h})(1000 \text{ m/km})/(3600 \text{ s/h})](0.08 \text{ m})}{1.784 \times 10^{-5} \text{ m}^2/\text{s}} = 6.228 \times 10^4$$

The Nusselt number corresponding to this Reynolds number is

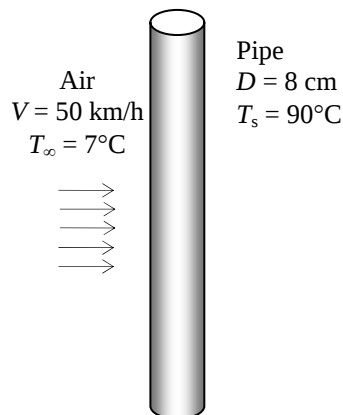
$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(6.228 \times 10^4)^{0.5} (0.7232)^{1/3}}{\left[1 + (0.4/0.7232)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{6.228 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 159.1 \end{aligned}$$

The heat transfer coefficient and the heat transfer rate become

$$h = \frac{k}{D} \text{Nu} = \frac{0.02724 \text{ W/m}\cdot^\circ\text{C}}{0.08 \text{ m}} (159.1) = 54.17 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.08 \text{ m})(1 \text{ m}) = 0.2513 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) = (54.17 \text{ W/m}^2\cdot^\circ\text{C})(0.2513 \text{ m}^2)(90 - 7)^\circ\text{C} = \mathbf{1130 \text{ W}} \text{ (per m length)}$$



12-69 The wind is blowing across a geothermal water pipe. The average wind velocity is to be determined.

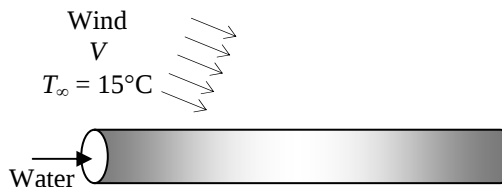
Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties The specific heat of water at the average temperature of 75°C is 4193 J/kg·°C. The properties of air at the film temperature of $(75+15)/2=45^\circ\text{C}$ are (Table A-22)

$$k = 0.02699 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.75 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7241$$



Analysis The rate of heat transfer from the pipe is the energy change of the water from inlet to exit of the pipe, and it can be determined from

$$\dot{Q} = \dot{m} c_p \Delta T = (8.5 \text{ kg/s})(4193 \text{ J/kg}\cdot^\circ\text{C})(80 - 70)^\circ\text{C} = 356,400 \text{ W}$$

The surface area and the heat transfer coefficient are

$$A = \pi DL = \pi(0.15 \text{ m})(400 \text{ m}) = 188.5 \text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A(T_s - T_\infty)} = \frac{356,400 \text{ W}}{(188.5 \text{ m}^2)(75 - 15)^\circ\text{C}} = 31.51 \text{ W/m}^2\cdot^\circ\text{C}$$

The Nusselt number is

$$Nu = \frac{hD}{k} = \frac{(31.51 \text{ W/m}^2\cdot^\circ\text{C})(0.15 \text{ m})}{0.02699 \text{ W/m}\cdot^\circ\text{C}} = 175.1$$

The Reynolds number may be obtained from the Nusselt number relation by trial-error or using an equation solver such as EES:

$$Nu = 0.3 + \frac{0.62 \text{ Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5}$$

$$175.1 = 0.3 + \frac{0.62 \text{ Re}^{0.5} (0.7241)^{1/3}}{\left[1 + (0.4/0.7241)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \longrightarrow \text{Re} = 71,900$$

The average wind velocity can be determined from Reynolds number relation

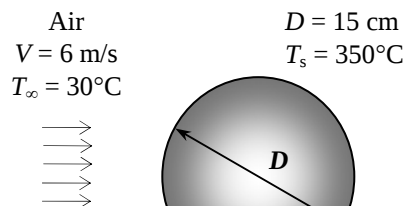
$$\text{Re} = \frac{VD}{\nu} \longrightarrow 71,900 = \frac{V(0.15 \text{ m})}{1.75 \times 10^{-5} \text{ m}^2/\text{s}} \longrightarrow V = 8.39 \text{ m/s} = \mathbf{30.2 \text{ km/h}}$$

12-70 A hot stainless steel ball is cooled by forced air. The average convection heat transfer coefficient and the cooling time are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The outer surface temperature of the ball is uniform at all times.

Properties The average surface temperature is $(350+250)/2 = 300^\circ\text{C}$, and the properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$\begin{aligned}k &= 0.02588 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= 1.608 \times 10^{-5} \text{ m}^2/\text{s} \\ \mu_\infty &= 1.872 \times 10^{-5} \text{ kg/m}\cdot\text{s} \\ \mu_{s, @ 300^\circ\text{C}} &= 2.934 \times 10^{-5} \text{ kg/m}\cdot\text{s} \\ \text{Pr} &= 0.7282\end{aligned}$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ m/s})(0.15 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 5.597 \times 10^4$$

The Nusselt number corresponding to this Reynolds number is determined to be

$$\begin{aligned}Nu &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(5.597 \times 10^4)^{0.5} + 0.06(5.597 \times 10^4)^{2/3} \right] (0.7282)^{0.4} \left(\frac{1.872 \times 10^{-5}}{2.934 \times 10^{-5}} \right)^{1/4} = 145.6\end{aligned}$$

Heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{0.15 \text{ m}} (145.6) = \mathbf{25.12 \text{ W/m}^2\cdot^\circ\text{C}}$$

The average rate of heat transfer can be determined from Newton's law of cooling by using average surface temperature of the ball

$$\begin{aligned}A_s &= \pi D^2 = \pi (0.15 \text{ m})^2 = 0.07069 \text{ m}^2 \\ \dot{Q}_{avg} &= hA_s(T_s - T_\infty) = (25.12 \text{ W/m}^2\cdot^\circ\text{C})(0.07069 \text{ m}^2)(300 - 30)^\circ\text{C} = 479.5 \text{ W}\end{aligned}$$

Assuming the ball temperature to be nearly uniform, the total heat transferred from the ball during the cooling from 350°C to 250°C can be determined from

$$Q_{total} = mc_p(T_1 - T_2)$$

$$\text{where } m = \rho V = \rho \frac{\pi D^3}{6} = (8055 \text{ kg/m}^3) \frac{\pi (0.15 \text{ m})^3}{6} = 14.23 \text{ kg}$$

$$\text{Therefore, } Q_{total} = mc_p(T_1 - T_2) = (14.23 \text{ kg})(480 \text{ J/kg}\cdot^\circ\text{C})(350 - 250)^\circ\text{C} = 683,250 \text{ J}$$

Then the time of cooling becomes

$$\Delta t = \frac{Q}{\dot{Q}_{avg}} = \frac{683,250 \text{ J}}{479.5 \text{ J/s}} = 1425 \text{ s} = \mathbf{23.7 \text{ min}}$$

12-71 EES Prob. 12-70 is reconsidered. The effect of air velocity on the average convection heat transfer coefficient and the cooling time is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D=0.15 [m]

T₁=350 [C]

T₂=250 [C]

T_{infinity}=30 [C]

P=101.3 [kPa]

Vel=6 [m/s]

rho_{ball}=8055 [kg/m³]

c_{p,ball}=480 [J/kg·C]

"PROPERTIES"

Fluid\$='air'

k=Conductivity(Fluid\$, T=T_{infinity})

Pr=Prandtl(Fluid\$, T=T_{infinity})

rho=Density(Fluid\$, T=T_{infinity}, P=P)

mu_{infinity}=Viscosity(Fluid\$, T=T_{infinity})

nu=mu_{infinity}/rho

mu_s=Viscosity(Fluid\$, T=T_{s,ave})

T_{s,ave}=1/2*(T₁+T₂)

"ANALYSIS"

Re=(Vel*D)/nu

Nusselt=2+(0.4*Re^{0.5}+0.06*Re^(2/3))*Pr^{0.4}*(mu_{infinity}/mu_s)^{0.25}

h=k/D*Nusselt

A=pi*D²

Q_{dot,ave}=h*A*(T_{s,ave}-T_{infinity})

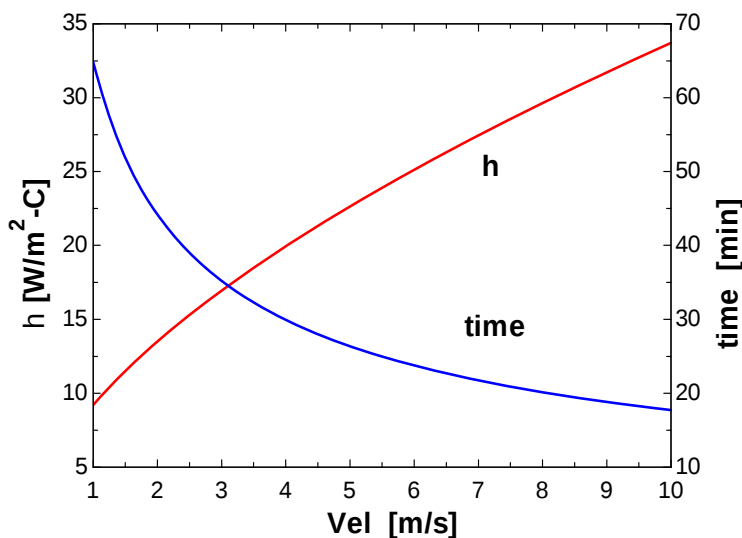
Q_{total}=m_{ball}*c_{p,ball}*(T₁-T₂)

m_{ball}=rho_{ball}*V_{ball}

V_{ball}=(pi*D³)/6

time=Q_{total}/Q_{dot,ave}*Convert(s, min)

Vel [m/s]	h [W/m ² ·C]	time [min]
1	9.204	64.83
1.5	11.5	51.86
2	13.5	44.2
2.5	15.29	39.01
3	16.95	35.21
3.5	18.49	32.27
4	19.94	29.92
4.5	21.32	27.99
5	22.64	26.36
5.5	23.9	24.96
6	25.12	23.75
6.5	26.3	22.69
7	27.44	21.74
7.5	28.55	20.9
8	29.63	20.14
8.5	30.69	19.44
9	31.71	18.81
9.5	32.72	18.24
10	33.7	17.7



12-72E A person extends his uncovered arms into the windy air outside. The rate of heat loss from the arm is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The arm is treated as a 2-ft-long and 3-in-diameter cylinder with insulated ends. 5 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (86+54)/2 = 70^\circ\text{F}$ are (Table A-22E)

$$k = 0.01457 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1643 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7306$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(20 \times 5280/3600) \text{ ft/s}](3/12) \text{ ft}}{0.1643 \times 10^{-3} \text{ ft}^2/\text{s}} = 4.463 \times 10^4$$

The Nusselt number corresponding to this Reynolds number is determined to be

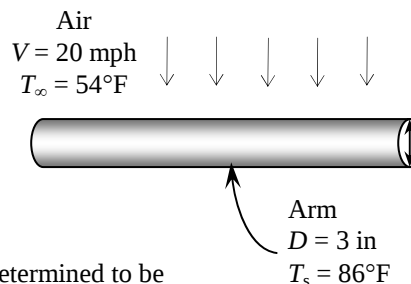
$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + \left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(4.463 \times 10^4)^{0.5} (0.7306)^{1/3}}{\left[1 + \left(\frac{0.4}{0.7306}\right)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{4.463 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 129.6 \end{aligned}$$

Then the heat transfer coefficient and the heat transfer rate from the arm becomes

$$h = \frac{k}{D} \text{Nu} = \frac{0.01457 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{(3/12) \text{ ft}} (129.6) = 7.557 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_s = \pi DL = \pi(3/12 \text{ ft})(2 \text{ ft}) = 1.571 \text{ ft}^2$$

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) = (7.557 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(1.571 \text{ ft}^2)(86 - 54)^\circ\text{F} = \mathbf{380 \text{ Btu/h}}$$



12-73E EES Prob. 12-72E is reconsidered. The effects of air temperature and wind velocity on the rate of heat loss from the arm are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$T_{\infty}=54$ [F]
 $Vel=20$ [mph]
 $T_s=86$ [F]
 $L=2$ [ft]
 $D=(3/12)$ [ft]

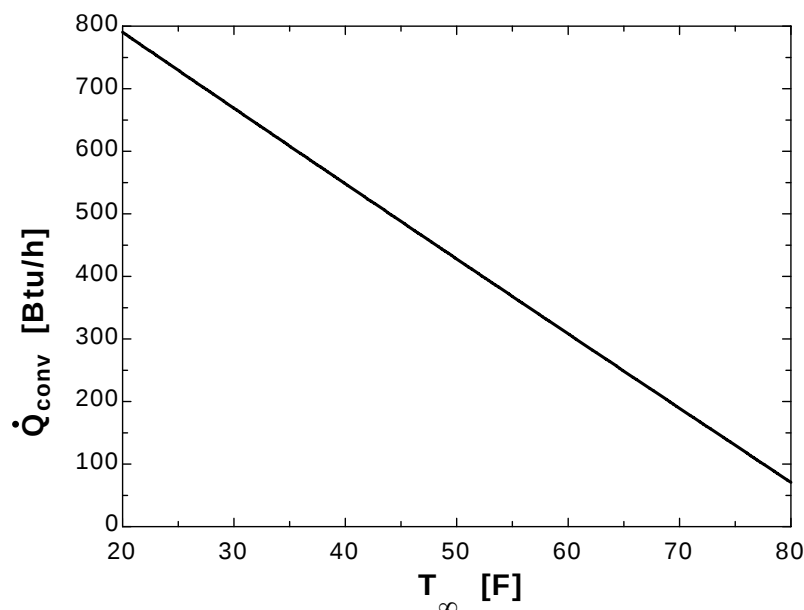
"PROPERTIES"

Fluid\$='air'
 $k=Conductivity(Fluid$, T=T_{film})$
 $Pr=Prandtl(Fluid$, T=T_{film})$
 $\rho=Density(Fluid$, T=T_{film}, P=14.7)$
 $\mu=Viscosity(Fluid$, T=T_{film}) \cdot Convert(lbm/ft-h, lbm/ft-s)$
 $\nu=\mu/\rho$
 $T_{film}=1/2 \cdot (T_s + T_{\infty})$

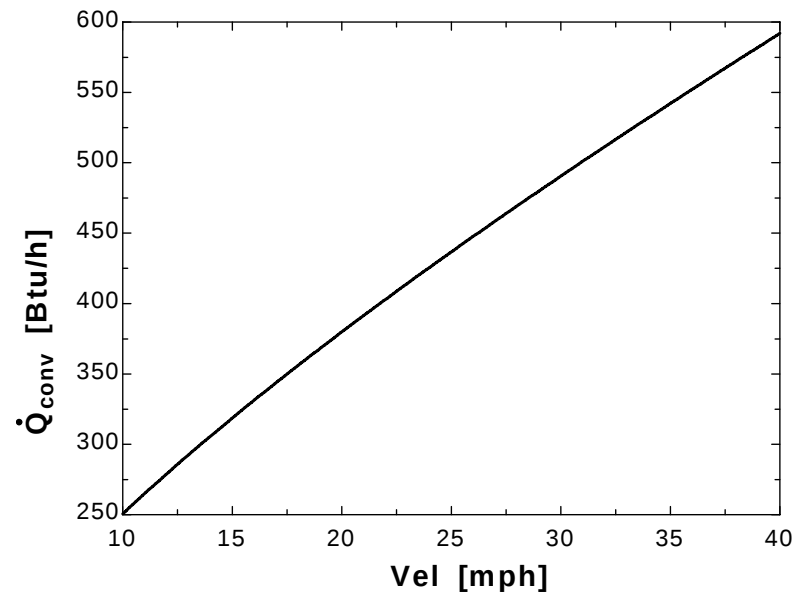
"ANALYSIS"

$Re=(Vel \cdot Convert(mph, ft/s) \cdot D)/\nu$
 $Nusselt=0.3+(0.62 \cdot Re^{0.5} \cdot Pr^{(1/3)})/(1+(0.4/Pr)^{(2/3)})^{0.25} \cdot (1+(Re/282000)^{(5/8)})^{(4/5)}$
 $h=k/D \cdot Nusselt$
 $A=\pi \cdot D \cdot L$
 $\dot{Q}_{conv}=h \cdot A \cdot (T_s - T_{\infty})$

T_{∞} [F]	$\dot{Q}_{conv}$ [Btu/h]
20	790.2
25	729.4
30	668.7
35	608.2
40	547.9
45	487.7
50	427.7
55	367.9
60	308.2
65	248.6
70	189.2
75	129.9
80	70.77



Vel [mph]	$\dot{Q}_{\text{conv}}$ [Btu/h]
10	250.6
12	278.9
14	305.7
16	331.3
18	356
20	379.8
22	403
24	425.6
26	447.7
28	469.3
30	490.5
32	511.4
34	532
36	552.2
38	572.2
40	591.9



12-74 The average surface temperature of the head of a person when it is not covered and is subjected to winds is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 One-quarter of the heat the person generates is lost from the head. 5 The head can be approximated as a 30-cm-diameter sphere. 6 The local atmospheric pressure is 1 atm.

Properties We assume the surface temperature to be 15°C for viscosity. The properties of air at 1 atm pressure and the free stream temperature of 10°C are (Table A-22)

$$k = 0.02439 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\mu_\infty = 1.778 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\mu_{s, @15^\circ\text{C}} = 1.802 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\text{Pr} = 0.7336$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25 \times 1000/3600) \text{ m/s}](0.3 \text{ m})}{1.426 \times 10^{-5} \text{ m}^2/\text{s}} = 1.461 \times 10^5$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} Nu = \frac{hD}{k} &= 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(1.461 \times 10^5)^{0.5} + 0.06(1.461 \times 10^5)^{2/3} \right] (0.7336)^{0.4} \left(\frac{1.778 \times 10^{-5}}{1.802 \times 10^{-5}} \right)^{1/4} = 283.2 \end{aligned}$$

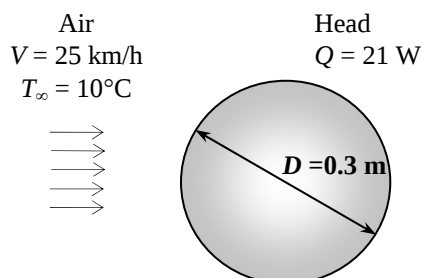
The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02439 \text{ W/m} \cdot ^\circ\text{C}}{0.3 \text{ m}} (283.2) = 23.02 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Then the surface temperature of the head is determined to be

$$A_s = \pi D^2 = \pi (0.3 \text{ m})^2 = 0.2827 \text{ m}^2$$

$$\dot{Q} = hA_s (T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10^\circ\text{C} + \frac{(84/4) \text{ W}}{(23.02 \text{ W/m}^2 \cdot ^\circ\text{C})(0.2827 \text{ m}^2)} = 13.2^\circ\text{C}$$



12-75 The flow of a fluid across an isothermal cylinder is considered. The change in the drag force and the rate of heat transfer when the free-stream velocity of the fluid is doubled is to be determined.

Analysis The drag force on a cylinder is given by

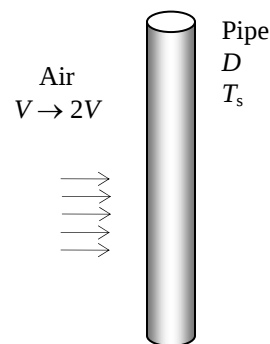
$$F_{D1} = C_D A_N \frac{\rho V^2}{2}$$

When the free-stream velocity of the fluid is doubled, the drag force becomes

$$F_{D2} = C_D A_N \frac{\rho (2V)^2}{2}$$

Taking the ratio of them yields

$$\frac{F_{D2}}{F_{D1}} = \frac{(2V)^2}{V^2} = 4$$



The rate of heat transfer between the fluid and the cylinder is given by Newton's law of cooling. We assume the Nusselt number is proportional to the n th power of the Reynolds number with $0.33 < n < 0.805$. Then,

$$\begin{aligned} \dot{Q}_1 &= h A_s (T_s - T_\infty) = \left(\frac{k}{D} Nu \right) A_s (T_s - T_\infty) = \frac{k}{D} (\text{Re})^n A_s (T_s - T_\infty) \\ &= \frac{k}{D} \left(\frac{VD}{\nu} \right)^n A_s (T_s - T_\infty) \\ &= V^n \frac{k}{D} \left(\frac{D}{\nu} \right)^n A_s (T_s - T_\infty) \end{aligned}$$

When the free-stream velocity of the fluid is doubled, the heat transfer rate becomes

$$\dot{Q}_2 = (2V)^n \frac{k}{D} \left(\frac{D}{\nu} \right)^n A_s (T_s - T_\infty)$$

Taking the ratio of them yields

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2V)^n}{V^n} = 2^n$$

12-76 The wind is blowing across the wire of a transmission line. The surface temperature of the wire is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 10°C. The properties of air at this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.246 \text{ kg/m}^3 \\ k &= 0.02439 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.426 \times 10^{-5} \text{ m}^2/\text{s} \\ \text{Pr} &= 0.7336\end{aligned}$$

$$\begin{aligned}\text{Wind} \\ V &= 40 \text{ km/h} \\ T_\infty &= 10^\circ\text{C}\end{aligned}$$



Transmission wire, T_s
 $D = 0.6 \text{ cm}$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](0.006 \text{ m})}{1.426 \times 10^{-5} \text{ m}^2/\text{s}} = 4675$$

The Nusselt number corresponding to this Reynolds number is determined to be

$$\begin{aligned}Nu &= \frac{hD}{k} = 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(4675)^{0.5} (0.7336)^{1/3}}{\left[1 + (0.4/0.7336)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{4675}{282,000}\right)^{5/8}\right]^{4/5} = 36.0\end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02439 \text{ W/m} \cdot ^\circ\text{C}}{0.006 \text{ m}} (36.0) = 146.3 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat generated in the electrical transmission lines per meter length is

$$\dot{W} = \dot{Q} = I^2 R = (50 \text{ A})^2 (0.002 \text{ Ohm}) = 5.0 \text{ W}$$

The entire heat generated in electrical transmission line has to be transferred to the ambient air. The surface temperature of the wire then becomes

$$\begin{aligned}A_s &= \pi DL = \pi(0.006 \text{ m})(1 \text{ m}) = 0.01885 \text{ m}^2 \\ \dot{Q} &= hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10^\circ\text{C} + \frac{5 \text{ W}}{(146.3 \text{ W/m}^2 \cdot ^\circ\text{C})(0.01885 \text{ m}^2)} = \mathbf{11.8^\circ\text{C}}\end{aligned}$$

12-77 EES Prob. 12-76 is reconsidered. The effect of the wind velocity on the surface temperature of the wire is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D=0.006 [m]
 L=1 [m] "unit length is considered"
 I=50 [Ampere]
 R=0.002 [Ohm]
 T_infinity=10 [C]
 Vel=40 [km/h]

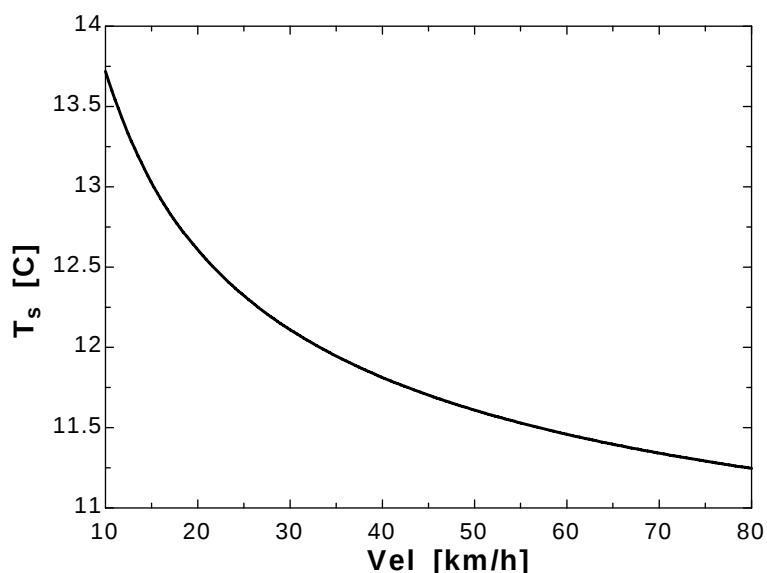
"PROPERTIES"

Fluid\$='air'
 k=Conductivity(Fluid\$, T=T_film)
 Pr=Prandtl(Fluid\$, T=T_film)
 rho=Density(Fluid\$, T=T_film, P=101.3)
 mu=Viscosity(Fluid\$, T=T_film)
 nu=mu/rho
 T_film=1/2*(T_s+T_infinity)

"ANALYSIS"

Re=(Vel*Convert(km/h, m/s)*D)/nu
 Nusselt=0.3+(0.62*Re^{0.5}*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^{0.25}*(1+(Re/282000)^(5/8))^(4/5)
 h=k/D*Nusselt
 W_dot=I²*R
 Q_dot=W_dot
 A=pi*D*L
 Q_dot=h*A*(T_s-T_infinity)

Vel [km/h]	T _s [C]
10	13.72
15	13.02
20	12.61
25	12.32
30	12.11
35	11.95
40	11.81
45	11.7
50	11.61
55	11.53
60	11.46
65	11.4
70	11.34
75	11.29
80	11.25



12-78 An aircraft is cruising at 900 km/h. A heating system keeps the wings above freezing temperatures. The average convection heat transfer coefficient on the wing surface and the average rate of heat transfer per unit surface area are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The wing is approximated as a cylinder of elliptical cross section whose minor axis is 50 cm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (0 - 55.4)/2 = -27.7^\circ\text{C}$ are (Table A-22)

$$k = 0.02152 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.106 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7421$$

Note that the atmospheric pressure will only affect the kinematic viscosity. The atmospheric pressure in atm unit is

$$P = (18.8 \text{ kPa}) \frac{1 \text{ atm}}{101.325 \text{ kPa}} = 0.1855 \text{ atm}$$

The kinematic viscosity at this atmospheric pressure is

$$\nu = (1.106 \times 10^{-5} \text{ m}^2/\text{s})/0.1855 = 5.961 \times 10^{-5} \text{ m}^2/\text{s}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(900 \times 1000/3600) \text{ m/s}](0.5 \text{ m})}{5.961 \times 10^{-5} \text{ m}^2/\text{s}} = 2.097 \times 10^6$$

The Nusselt number relation for a cylinder of elliptical cross-section is limited to $\text{Re} < 15,000$, and the relation below is not really applicable in this case. However, this relation is all we have for elliptical shapes, and we will use it with the understanding that the results may not be accurate.

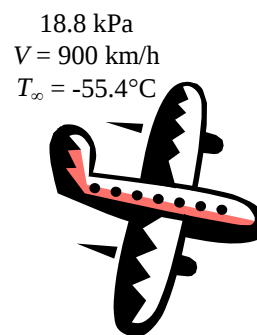
$$\text{Nu} = \frac{hD}{k} = 0.248 \text{Re}^{0.612} \text{Pr}^{1/3} = 0.248(2.097 \times 10^6)^{0.612} (0.7241)^{1/3} = 1660$$

The average heat transfer coefficient on the wing surface is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02152 \text{ W/m}\cdot^\circ\text{C}}{0.5 \text{ m}} (1660) = \mathbf{71.45 \text{ W/m}^2\cdot^\circ\text{C}}$$

Then the average rate of heat transfer per unit surface area becomes

$$\dot{q} = h(T_s - T_\infty) = (71.45 \text{ W/m}^2\cdot^\circ\text{C})[0 - (-55.4)]^\circ\text{C} = \mathbf{3958 \text{ W/m}^2}$$



12-79 A long aluminum wire is cooled by cross air flowing over it. The rate of heat transfer from the wire per meter length when it is first exposed to the air is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (370 + 30)/2 = 200^\circ\text{C}$ are (Table A-22)

$$k = 0.03779 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 3.455 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.6974$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ m/s})(0.003 \text{ m})}{3.455 \times 10^{-5} \text{ m}^2/\text{s}} = 521.0$$

The Nusselt number corresponding to this Reynolds number is determined to be

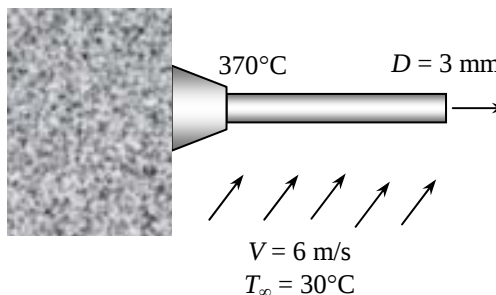
$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(521.0)^{0.5} (0.6974)^{1/3}}{\left[1 + (0.4/0.6974)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{521.0}{282,000}\right)^{5/8}\right]^{4/5} = 11.48 \end{aligned}$$

Then the heat transfer coefficient and the heat transfer rate from the wire per meter length become

$$h = \frac{k}{D} \text{Nu} = \frac{0.03779 \text{ W/m}\cdot^\circ\text{C}}{0.003 \text{ m}} (11.48) = 144.6 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.003 \text{ m})(1 \text{ m}) = 0.009425 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) = (144.6 \text{ W/m}^2\cdot^\circ\text{C})(0.009425 \text{ m}^2)(370 - 30)^\circ\text{C} = \mathbf{463 \text{ W}}$$



12-80E A fan is blowing air over the entire body of a person. The average temperature of the outer surface of the person is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The average human body can be treated as a 1-ft-diameter cylinder with an exposed surface area of 18 ft². 5 The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 100°F.

The properties of air at this temperature are (Table A-22E)

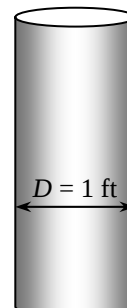
$$k = 0.01529 \text{ Btu/h.ft.}^\circ\text{F}$$

$$\nu = 1.809 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7260$$

$$V = 6 \text{ ft/s}$$

$$T_\infty = 85^\circ\text{F}$$



Person, T_s
300 Btu/h

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ ft/s})(1 \text{ ft})}{1.809 \times 10^{-4} \text{ ft}^2/\text{s}} = 3.317 \times 10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(3.317 \times 10^4)^{0.5} (0.7260)^{1/3}}{\left[1 + (0.4/0.7260)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{3.317 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 107.8 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.01529 \text{ Btu/h.ft.}^\circ\text{F}}{1 \text{ ft}} (107.8) = 1.649 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

Then the average temperature of the outer surface of the person becomes

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85^\circ\text{F} + \frac{300 \text{ Btu/h}}{(1.649 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(18 \text{ ft}^2)} = \mathbf{95.1^\circ\text{F}}$$

If the air velocity were doubled, the Reynolds number would be

$$\text{Re} = \frac{VD}{\nu} = \frac{(12 \text{ ft/s})(1 \text{ ft})}{1.809 \times 10^{-4} \text{ ft}^2/\text{s}} = 6.633 \times 10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(6.633 \times 10^4)^{0.5} (0.7260)^{1/3}}{\left[1 + (0.4/0.7260)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{6.633 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 165.9 \end{aligned}$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.01529 \text{ Btu/h.ft.}^\circ\text{F}}{1 \text{ ft}} (165.9) = 2.537 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

Then the average temperature of the outer surface of the person becomes

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85^\circ\text{F} + \frac{300 \text{ Btu/h}}{(2.537 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(18 \text{ ft}^2)} = \mathbf{91.6^\circ\text{F}}$$

12-81 A light bulb is cooled by a fan. The equilibrium temperature of the glass bulb is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The light bulb is in spherical shape. 4 The local atmospheric pressure is 1 atm.

Properties We assume the surface temperature to be 100°C for viscosity. The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$\begin{aligned}k &= 0.02588 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= 1.608 \times 10^{-5} \text{ m}^2/\text{s} \\ \mu_\infty &= 1.872 \times 10^{-5} \text{ kg/m}\cdot\text{s} \\ \mu_{s, @ 100^\circ\text{C}} &= 2.181 \times 10^{-5} \text{ kg/m}\cdot\text{s} \\ \text{Pr} &= 0.7282\end{aligned}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.1 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 1.244 \times 10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned}Nu &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(1.244 \times 10^4)^{0.5} + 0.06(1.244 \times 10^4)^{2/3} \right] (0.7282)^{0.4} \left(\frac{1.872 \times 10^{-5}}{2.181 \times 10^{-5}} \right)^{1/4} = 67.14\end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{0.1 \text{ m}} (67.14) = 17.37 \text{ W/m}^2\cdot^\circ\text{C}$$

Noting that 90 % of electrical energy is converted to heat,

$$\dot{Q} = (0.90)(100 \text{ W}) = 90 \text{ W}$$

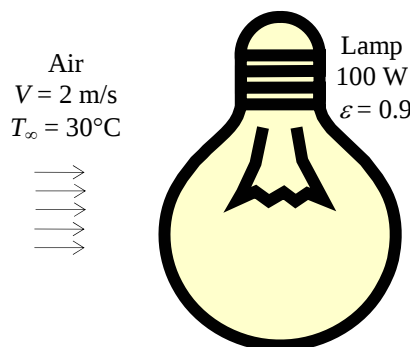
The bulb loses heat by both convection and radiation. The equilibrium temperature of the glass bulb can be determined by iteration or by an equation solver:

$$A_s = \pi D^2 = \pi (0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4)$$

$$\begin{aligned}90 \text{ W} &= (17.37 \text{ W/m}^2\cdot^\circ\text{C})(0.0314 \text{ m}^2)[T_s - (30 + 273)\text{K}] \\ &\quad + (0.9)(0.0314 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[T_s^4 - (30 + 273 \text{ K})^4]\end{aligned}$$

$$T_s = 409.9 \text{ K} = \mathbf{136.9^\circ\text{C}}$$



12-82 A steam pipe is exposed to a light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipe are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The plant operates every day of the year for 10 h a day. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of

$$(T_s + T_\infty)/2 = (75+5)/2 = 40^\circ\text{C} \text{ are (Table A-22)}$$

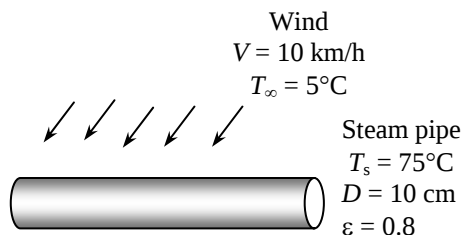
$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7255$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(10 \times 1000/3600) \text{ m/s}](0.1 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 1.632 \times 10^4$$



The Nusselt number corresponding to this Reynolds number is determined to be

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(1.632 \times 10^4)^{0.5} (0.7255)^{1/3}}{\left[1 + (0.4/0.7255)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{1.632 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 71.19 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{0.1 \text{ m}} (71.19) = 18.95 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.1 \text{ m})(12 \text{ m}) = 3.77 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (18.95 \text{ W/m}^2\cdot^\circ\text{C})(3.77 \text{ m}^2)(75 - 5)^\circ\text{C} = 5001 \text{ W}$$

The rate of heat loss by radiation is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.8)(3.77 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(75 + 273 \text{ K})^4 - (0 + 273 \text{ K})^4] = 1558 \text{ W} \end{aligned}$$

The total rate of heat loss then becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 5001 + 1558 = 6559 \text{ W}$$

The amount of heat loss from the steam during a 10-hour work day is

$$Q = \dot{Q}_{total} \Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = \mathbf{2.361 \times 10^5 \text{ kJ/day}}$$

The total amount of heat loss from the steam per year is

$$Q_{total} = \dot{Q}_{day} (\text{no. of days}) = (2.361 \times 10^5 \text{ kJ/day})(365 \text{ days/yr}) = 8.619 \times 10^7 \text{ kJ/yr}$$

Noting that the steam generator has an efficiency of 80%, the amount of gas used is

$$Q_{gas} = \frac{Q_{total}}{0.80} = \frac{8.619 \times 10^7 \text{ kJ/yr}}{0.80} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 1021 \text{ therms/yr}$$

Insulation reduces this amount by 90%. The amount of energy and money saved becomes

$$\text{Energy saved} = (0.90)Q_{gas} = (0.90)(1021 \text{ therms/yr}) = 919 \text{ therms/yr}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (919 \text{ therms/yr})(\$1.05/\text{therm}) = \mathbf{\$965}$$

12-83 A steam pipe is exposed to light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipes are to be determined.

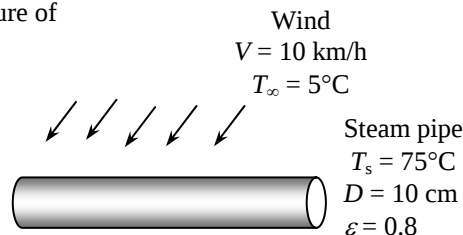
Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The plant operates every day of the year for 10 h. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40^\circ\text{C}$ are (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7255$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(10 \times 1000/3600) \text{ m/s}](0.1 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 1.632 \times 10^4$$

The Nusselt number corresponding to this Reynolds number is determined to be

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(1.632 \times 10^4)^{0.5} (0.7255)^{1/3}}{\left[1 + (0.4/0.7255)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{1.632 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 71.19 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{0.1 \text{ m}} (71.19) = 18.95 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.1 \text{ m})(12 \text{ m}) = 3.77 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (18.95 \text{ W/m}^2\cdot^\circ\text{C})(3.77 \text{ m}^2)(75 - 5)^\circ\text{C} = 5001 \text{ W}$$

For an average surrounding temperature of 0°C , the rate of heat loss by radiation and the total rate of heat loss are

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.8)(3.77 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(75 + 273 \text{ K})^4 - (0 + 273 \text{ K})^4] = 1558 \text{ W} \end{aligned}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5001 + 1558 = 6559 \text{ W}$$

If the average surrounding temperature is -20°C , the rate of heat loss by radiation and the total rate of heat loss become

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.8)(3.77 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(75 + 273 \text{ K})^4 - (-20 + 273 \text{ K})^4] \\ &= 1807 \text{ W} \end{aligned}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5001 + 1807 = 6808 \text{ W}$$

which is $6808 - 6559 = 249 \text{ W}$ more than the value for a surrounding temperature of 0°C . This corresponds to

$$\% \text{ change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total}, 0^\circ\text{C}}} \times 100 = \frac{249 \text{ W}}{6559 \text{ W}} \times 100 = \mathbf{3.8\%} \quad (\text{increase})$$

If the average surrounding temperature is 25°C, the rate of heat loss by radiation and the total rate of heat loss become

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.8)(3.77 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) \left[(75 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4 \right] \\ &= 1159 \text{ W}\end{aligned}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5001 + 1159 = 6160 \text{ W}$$

which is 6559 - 6160 = 399 W less than the value for a surrounding temperature of 0°C. This corresponds to

$$\% \text{ change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total}, 0^\circ\text{C}}} \times 100 = \frac{399 \text{ W}}{6559 \text{ W}} \times 100 = \mathbf{6.1\%} \quad (\text{decrease})$$

Therefore, the effect of the temperature variations of the surrounding surfaces on the total heat transfer is less than 6%.

12-84E An electrical resistance wire is cooled by a fan. The surface temperature of the wire is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 200°F. The properties of air at this temperature are (Table A-22E)

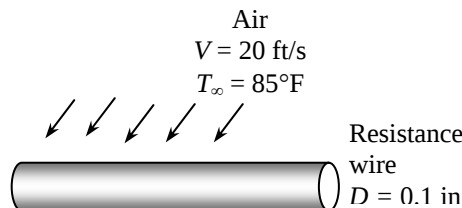
$$k = 0.01761 \text{ Btu/h.ft.}^\circ\text{F}$$

$$\nu = 2.406 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7124$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(20 \text{ ft/s})(0.1/12 \text{ ft})}{2.406 \times 10^{-4} \text{ ft}^2/\text{s}} = 692.7$$



The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(692.7)^{0.5} (0.7124)^{1/3}}{\left[1 + (0.4/0.7124)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{692.7}{282,000}\right)^{5/8}\right]^{4/5} = 13.34 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.01761 \text{ Btu/h.ft.}^\circ\text{F}}{(0.1/12 \text{ ft})} (13.34) = 28.19 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

Then the average temperature of the outer surface of the wire becomes

$$A_s = \pi DL = \pi(0.1/12 \text{ ft})(12 \text{ ft}) = 0.3142 \text{ ft}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 85^\circ\text{F} + \frac{(1500 \times 3.41214) \text{ Btu/h}}{(28.19 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(0.3142 \text{ ft}^2)} = \mathbf{662.9^\circ\text{F}}$$

Discussion Repeating the calculations at the new film temperature of $(85 + 662.9)/2 = 374^\circ\text{F}$ gives $T_s = 668.3^\circ\text{F}$.

12-85 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65 + 30)/2 = 47.5^\circ\text{C}$ are (Table A-22)

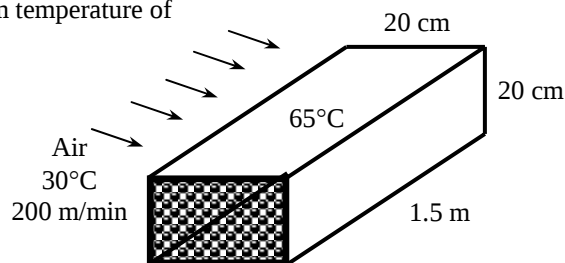
$$k = 0.02717 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.774 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7235$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(200/60) \text{ m/s}](0.2 \text{ m})}{1.774 \times 10^{-5} \text{ m}^2/\text{s}} = 3.758 \times 10^4$$



Using the relation for a square duct from Table 12-1, the Nusselt number is determined to be

$$Nu = \frac{hD}{k} = 0.102 \text{ Re}^{0.675} \text{ Pr}^{1/3} = 0.102(3.758 \times 10^4)^{0.675} (0.7235)^{1/3} = 112.2$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02717 \text{ W/m}\cdot^\circ\text{C}}{0.2 \text{ m}} (112.2) = 15.24 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the rate of heat transfer from the duct becomes

$$A_s = (4 \times 0.2 \text{ m})(1.5 \text{ m}) = 1.2 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (15.24 \text{ W/m}^2\cdot^\circ\text{C})(1.2 \text{ m}^2)(65 - 30)^\circ\text{C} = \mathbf{640 \text{ W}}$$

12-86 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined. $\checkmark$

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65 + 30)/2 = 47.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02717 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.774 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7235$$

For a location at 4000 m altitude where the atmospheric pressure is 61.66 kPa, only kinematic viscosity of air will be affected. Thus,

$$\nu_{@61.66 \text{ kPa}} = \left(\frac{101.325}{61.66} \right) (1.774 \times 10^{-5}) = 2.915 \times 10^{-5} \text{ m}^2/\text{s}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(200/60) \text{ m/s}](0.2 \text{ m})}{2.915 \times 10^{-5} \text{ m}^2/\text{s}} = 2.287 \times 10^4$$

Using the relation for a square duct from Table 12-1, the Nusselt number is determined to be

$$\text{Nu} = \frac{hD}{k} = 0.102 \text{Re}^{0.675} \text{Pr}^{1/3} = 0.102(2.287)^{0.675} (0.7235)^{1/3} = 80.21$$

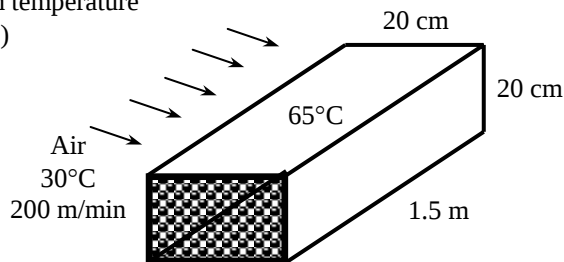
The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02717 \text{ W/m}\cdot^\circ\text{C}}{0.2 \text{ m}} (80.21) = 10.90 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the rate of heat transfer from the duct becomes

$$A_s = (4 \times 0.2 \text{ m})(1.5 \text{ m}) = 1.2 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (10.90 \text{ W/m}^2\cdot^\circ\text{C})(1.2 \text{ m}^2)(65 - 30)^\circ\text{C} = \mathbf{458 \text{ W}}$$



12-87 A cylindrical electronic component mounted on a circuit board is cooled by air flowing across it. The surface temperature of the component is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 50°C. The properties of air at 1 atm and at this temperature are (Table A-22)

$$k = 0.02735 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7228$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(240/60 \text{ m/s})(0.003 \text{ m})}{1.798 \times 10^{-5} \text{ m}^2/\text{s}} = 667.4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(667.4)^{0.5} (0.7228)^{1/3}}{\left[1 + (0.4/0.7228)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{667.4}{282,000}\right)^{5/8}\right]^{4/5} = 13.17 \end{aligned}$$

The heat transfer coefficient is

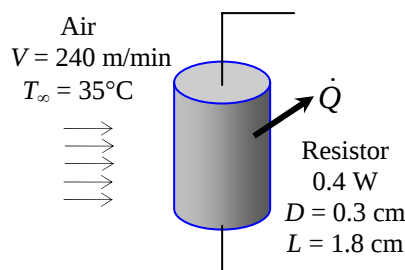
$$h = \frac{k}{D} \text{Nu} = \frac{0.02735 \text{ W/m} \cdot ^\circ\text{C}}{0.003 \text{ m}} (13.17) = 120.0 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Then the surface temperature of the component becomes

$$A_s = \pi DL = \pi(0.003 \text{ m})(0.018 \text{ m}) = 0.0001696 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 35^\circ\text{C} + \frac{0.4 \text{ W}}{(120.0 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0001696 \text{ m}^2)} = 54.6^\circ\text{C}$$

The film temperature is $(54.6 + 35)/2 = 44.8^\circ\text{C}$, which is sufficiently close to the assumed value of 50°C . Therefore, there is no need to repeat calculations.



12-88 A cylindrical hot water tank is exposed to windy air. The temperature of the tank after a 45-min cooling period is to be estimated.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The surface of the tank is at the same temperature as the water temperature. 5 The heat transfer coefficient on the top and bottom surfaces is the same as that on the side surfaces.

Properties The properties of water at 80°C are (Table A-15)

$$\rho = 971.8 \text{ kg/m}^3$$

$$c_p = 4197 \text{ J/kg} \cdot ^\circ\text{C}$$

The properties of air at 1 atm and at the anticipated film temperature of 50°C are (Table A-22)

$$k = 0.02735 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7228$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{\left(\frac{40 \times 1000}{3600} \text{ m/s}\right)(0.50 \text{ m})}{1.798 \times 10^{-5} \text{ m}^2/\text{s}} = 3.090 \times 10^5$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(3.090 \times 10^5)^{0.5} (0.7228)^{1/3}}{\left[1 + (0.4/0.7228)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{3.090 \times 10^5}{282,000}\right)^{5/8}\right]^{4/5} = 484.8 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02735 \text{ W/m} \cdot ^\circ\text{C}}{0.50 \text{ m}} (484.8) = 26.52 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The surface area of the tank is

$$A_s = \pi DL + 2\pi \frac{D^2}{4} = \pi(0.5)(0.95) + 2\pi(0.5)^2/4 = 1.885 \text{ m}^2$$

The rate of heat transfer is determined from

$$\dot{Q} = hA_s(T_s - T_\infty) = (26.52 \text{ W/m}^2 \cdot ^\circ\text{C})(1.885 \text{ m}^2) \left(\frac{80 + T_2}{2} - 18\right)^\circ\text{C} \quad (\text{Eq. 1})$$

where T_2 is the final temperature of water so that $(80 + T_2)/2$ gives the average temperature of water during the cooling process. The mass of water in the tank is

$$m = \rho V = \rho \pi \frac{D^2}{4} L = (971.8 \text{ kg/m}^3) \pi (0.50 \text{ m})^2 (0.95 \text{ m}) / 4 = 181.3 \text{ kg}$$

The amount of heat transfer from the water is determined from

$$Q = mc_p(T_2 - T_1) = (181.3 \text{ kg})(4197 \text{ J/kg} \cdot ^\circ\text{C})(80 - T_2)^\circ\text{C}$$

Then average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{(181.3 \text{ kg})(4197 \text{ J/kg} \cdot ^\circ\text{C})(80 - T_2)^\circ\text{C}}{45 \times 60 \text{ s}} \quad (\text{Eq. 2})$$

Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of water

$$\dot{Q} = (26.52 \text{ W/m}^2 \cdot ^\circ\text{C})(1.885 \text{ m}^2) \left(\frac{80 + T_2}{2} - 18\right)^\circ\text{C} = \frac{(181.3 \text{ kg})(4197 \text{ J/kg} \cdot ^\circ\text{C})(80 - T_2)^\circ\text{C}}{45 \times 60 \text{ s}}$$

$$T_2 = 69.9^\circ\text{C}$$

Water tank
D = 50 cm
L = 95 cm



Air
V = 40 km/h
 $T_\infty = 18^\circ\text{C}$

12-89 EES Prob. 12-88 is reconsidered. The temperature of the tank as a function of the cooling time is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D=0.50 [m]

L=0.95 [m]

T_w1=80 [C]

T_infinity=18 [C]

Vel=40 [km/h]

time=45 [min]

"PROPERTIES"

Fluid\$='air'

k=Conductivity(Fluid\$, T=T_film)

Pr=Prandtl(Fluid\$, T=T_film)

rho=Density(Fluid\$, T=T_film, P=101.3)

mu=Viscosity(Fluid\$, T=T_film)

nu=mu/rho

T_film=1/2*(T_w_ave+T_infinity)

rho_w=Density(water, T=T_w_ave, P=101.3)

c_p_w=CP(Water, T=T_w_ave, P=101.3)*Convert(kJ/kg-C, J/kg-C)

T_w_ave=1/2*(T_w1+T_w2)

"ANALYSIS"

Re=(Vel*Convert(km/h, m/s)*D)/nu

Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8))^(4/5)

h=k/D*Nusselt

A=pi*D*L+2*pi*D^2/4

Q_dot=h*A*(T_w_ave-T_infinity)

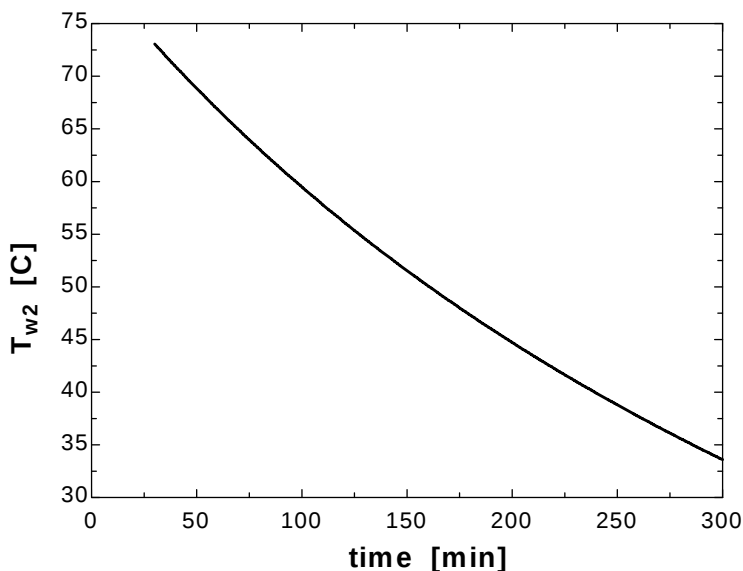
m_w=rho_w*V_w

V_w=pi*D^2/4*L

Q=m_w*c_p_w*(T_w1-T_w2)

Q_dot=Q/(time*Convert(min, s))

time [min]	T _{w2} [C]
30	73.06
45	69.86
60	66.83
75	63.96
90	61.23
105	58.63
120	56.16
135	53.8
150	51.54
165	49.39
180	47.33
195	45.36
210	43.47
225	41.65
240	39.91
255	38.24
270	36.63
285	35.09
300	33.6



12-90 Air flows over a spherical tank containing iced water. The rate of heat transfer to the tank and the rate at which ice melts are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 25°C are (Table A-22)

$$k = 0.02551 \text{ W/m} \cdot ^\circ\text{C}$$

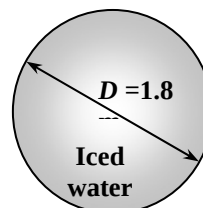
$$\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\mu_\infty = 1.849 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\mu_{s, @ 0^\circ\text{C}} = 1.729 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\text{Pr} = 0.7296$$

Air
 $V = 7 \text{ m/s}$
 $T_\infty = 25^\circ\text{C}$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(7 \text{ m/s})(1.8 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 8.067 \times 10^5$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(8.067 \times 10^5)^{0.5} + 0.06(8.067 \times 10^5)^{2/3} \right] (0.7296)^{0.4} \left(\frac{1.849 \times 10^{-5}}{1.729 \times 10^{-5}} \right)^{1/4} = 790.1 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02551 \text{ W/m} \cdot ^\circ\text{C}}{1.8 \text{ m}} (790.1) = 11.20 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Then the rate of heat transfer is determined to be

$$A_s = \pi D^2 = \pi (1.8 \text{ m})^2 = 10.18 \text{ m}^2$$

$$\dot{Q} = hA_s (T_s - T_\infty) = (11.20 \text{ W/m}^2 \cdot ^\circ\text{C})(10.18 \text{ m}^2)(25 - 0)^\circ\text{C} = \mathbf{2850 \text{ W}}$$

The rate at which ice melts is

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{2.85 \text{ kW}}{333.7 \text{ kJ/kg}} = 0.00854 \text{ kg/s} = \mathbf{0.512 \text{ kg/min}}$$

12-91 A cylindrical bottle containing cold water is exposed to windy air. The average wind velocity is to be estimated.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 Heat transfer at the top and bottom surfaces is negligible.

Properties The properties of water at the average temperature of $(T_1 + T_2)/2 = (3 + 11)/2 = 7^\circ\text{C}$ are (Table A-15)

$$\rho = 999.8 \text{ kg/m}^3$$

$$c_p = 4200 \text{ J/kg}\cdot^\circ\text{C}$$

The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (7 + 27)/2 = 17^\circ\text{C}$ are (Table A-22)

$$k = 0.02491 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.488 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7317$$

Analysis The mass of water in the bottle is

$$m = \rho V = \rho \pi \frac{D^2}{4} L = (999.8 \text{ kg/m}^3) \pi (0.10 \text{ m})^2 (0.30 \text{ m}) / 4 = 2.356 \text{ kg}$$

Then the amount of heat transfer to the water is

$$Q = mc_p (T_2 - T_1) = (2.356 \text{ kg})(4200 \text{ J/kg}\cdot^\circ\text{C})(11 - 3)^\circ\text{C} = 79,162 \text{ J}$$

The average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{79,162 \text{ J}}{45 \times 60 \text{ s}} = 29.32 \text{ W}$$

The heat transfer coefficient is

$$A_s = \pi DL = \pi (0.10 \text{ m})(0.30 \text{ m}) = 0.09425 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = hA_s (T_\infty - T_s) \longrightarrow 29.32 \text{ W} = h(0.09425 \text{ m}^2)(27 - 7)^\circ\text{C} \longrightarrow h = 15.55 \text{ W/m}^2\cdot^\circ\text{C}$$

The Nusselt number is

$$\text{Nu} = \frac{hD}{k} = \frac{(15.55 \text{ W/m}^2\cdot^\circ\text{C})(0.10 \text{ m})}{0.02491 \text{ W/m}\cdot^\circ\text{C}} = 62.42$$

Reynolds number can be obtained from the Nusselt number relation for a flow over the cylinder

$$\text{Nu} = 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5}$$

$$62.42 = 0.3 + \frac{0.62 \text{Re}^{0.5} (0.7317)^{1/3}}{\left[1 + (0.4/0.7317)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \longrightarrow \text{Re} = 12,856$$

Then using the Reynolds number relation we determine the wind velocity

$$\text{Re} = \frac{VD}{\nu} \longrightarrow 12,856 = \frac{V(0.10 \text{ m})}{1.488 \times 10^{-5} \text{ m}^2/\text{s}} \longrightarrow V = 1.91 \text{ m/s}$$

Air
V
 $T_\infty = 27^\circ\text{C}$



Bottle
 $D = 10 \text{ cm}$
 $L = 30 \text{ cm}$

Review Problems

12-92 Wind is blowing parallel to the walls of a house. The rate of heat loss from the wall is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties Assuming a film temperature of $T_f = 10^\circ\text{C}$ for the outdoors, the properties of air are evaluated to be (Table A-22)

$$k = 0.02439 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7336$$

Analysis Air flows along 8-m side. The Reynolds number in this case is

$$Re_L = \frac{VL}{\nu} = \frac{[(50 \times 1000 / 3600) \text{ m/s}](8 \text{ m})}{1.426 \times 10^{-5} \text{ m}^2/\text{s}} = 7.792 \times 10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{h_o L}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(7.792 \times 10^6)^{0.8} - 871](0.7336)^{1/3} = 10,096$$

$$h_o = \frac{k}{L} Nu = \frac{0.02439 \text{ W/m}\cdot^\circ\text{C}}{8 \text{ m}} (10,096) = 30.78 \text{ W/m}^2\cdot^\circ\text{C}$$

The thermal resistances are

$$A_s = wL = (4 \text{ m})(8 \text{ m}) = 32 \text{ m}^2$$

$$R_i = \frac{1}{h_i A_s} = \frac{1}{(8 \text{ W/m}^2\cdot^\circ\text{C})(32 \text{ m}^2)} = 0.0039^\circ\text{C/W}$$

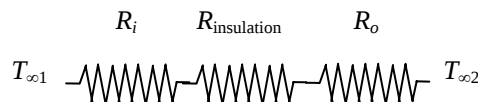
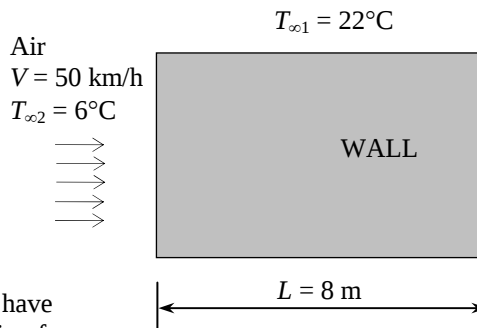
$$R_{insulation} = \frac{(R - 3.38)_{value}}{A_s} = \frac{3.38 \text{ m}^2\cdot^\circ\text{C/W}}{32 \text{ m}^2} = 0.1056^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A_s} = \frac{1}{(30.78 \text{ W/m}^2\cdot^\circ\text{C})(32 \text{ m}^2)} = 0.0010^\circ\text{C/W}$$

Then the total thermal resistance and the heat transfer rate through the wall are determined from

$$R_{total} = R_i + R_{insulation} + R_o = 0.0039 + 0.1056 + 0.0010 = 0.1105^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(22 - 6)^\circ\text{C}}{0.1105^\circ\text{C/W}} = 145 \text{ W}$$



12-93 A car travels at a velocity of 60 km/h. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm. 5 The flow is turbulent over the entire surface because of the constant agitation of the engine block. 6 The bottom surface of the engine is a flat surface.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40^\circ\text{C}$ are (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7255$$

Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{[(60 \times 1000 / 3600) \text{ m/s}](0.7 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 6.855 \times 10^5$$

which is less than the critical Reynolds number. But we will assume turbulent flow because of the constant agitation of the engine block.

$$Nu = \frac{hL}{k} = 0.037 Re_L^{0.8} Pr^{1/3} = 0.037(6.855 \times 10^5)^{0.8} (0.7255)^{1/3} = 1551$$

$$h = \frac{k}{L} Nu = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{0.7 \text{ m}} (1551) = 58.97 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (58.97 \text{ W/m}^2 \cdot ^\circ\text{C})[(0.6 \text{ m})(0.7 \text{ m})](75 - 5)^\circ\text{C} = 1734 \text{ W}$$

The heat loss by radiation is then determined from Stefan-Boltzman law to be

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.92)(0.6 \text{ m})(0.7 \text{ m})(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(75 + 273 \text{ K})^4 - (10 + 273 \text{ K})^4] = 181 \text{ W} \end{aligned}$$

Then the total rate of heat loss from the bottom surface of the engine block becomes

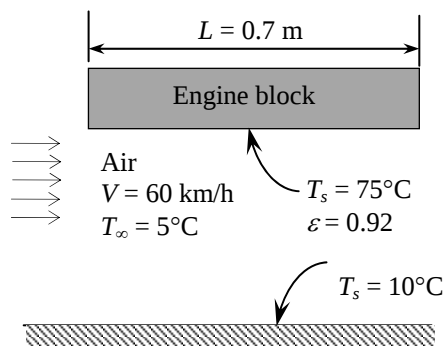
$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1734 + 181 = \mathbf{1915 \text{ W}}$$

The gunk will introduce an additional resistance to heat dissipation from the engine. The total heat transfer rate in this case can be calculated from

$$\dot{Q} = \frac{T_\infty - T_s}{\frac{1}{hA_s} + \frac{L}{kA_s}} = \frac{(75 - 5)^\circ\text{C}}{\frac{1}{(58.97 \text{ W/m}^2 \cdot ^\circ\text{C})[(0.6 \text{ m})(0.7 \text{ m})]} + \frac{(0.002 \text{ m})}{(3 \text{ W/m}\cdot^\circ\text{C})(0.6 \text{ m} \times 0.7 \text{ m})}} = 1668 \text{ W}$$

The decrease in the heat transfer rate is

$$1734 - 1668 = \mathbf{66 \text{ W (3.8\%)}}$$



12-94E A minivan is traveling at 60 mph. The rate of heat transfer to the van is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air flow is turbulent because of the intense vibrations involved. 5 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties Assuming a film temperature of $T_f = 80^\circ\text{F}$, the properties of air are evaluated to be (Table A-22E)

$$k = 0.01481 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 1.697 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$Pr = 0.7290$$

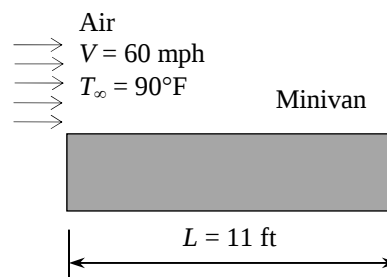
Analysis Air flows along 11 ft long side. The Reynolds number in this case is

$$Re_L = \frac{VL}{\nu} = \frac{[(60 \times 5280 / 3600) \text{ ft/s}](11 \text{ ft})}{1.697 \times 10^{-4} \text{ ft}^2/\text{s}} = 5.704 \times 10^6$$

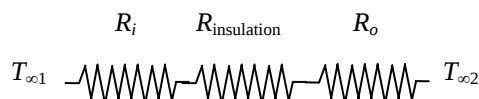
which is greater than the critical Reynolds number. The air flow is assumed to be entirely turbulent because of the intense vibrations involved. Then the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = \frac{h_o L}{k} = 0.037 Re_L^{0.8} Pr^{1/3} = 0.037 (5.704 \times 10^6)^{0.8} (0.7290)^{1/3} = 8461$$

$$h_o = \frac{k}{L} Nu = \frac{0.01481 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{11 \text{ ft}} (8461) = 11.39 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$



The thermal resistances are



$$A_s = 2[(3.2 \text{ ft})(6 \text{ ft}) + (3.2 \text{ ft})(11 \text{ ft}) + (6 \text{ ft})(11 \text{ ft})] = 240.8 \text{ ft}^2$$

$$R_i = \frac{1}{h_i A_s} = \frac{1}{(1.2 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(240.8 \text{ ft}^2)} = 0.0035 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_{insulation} = \frac{(R-3)_{value}}{A_s} = \frac{3 \text{ h}\cdot\text{ft}^2\cdot^\circ\text{F/Btu}}{(240.8 \text{ ft}^2)} = 0.0125 \text{ h}\cdot^\circ\text{F/Btu}$$

$$R_o = \frac{1}{h_o A_s} = \frac{1}{(11.39 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(240.8 \text{ ft}^2)} = 0.0004 \text{ h}\cdot^\circ\text{F/Btu}$$

Then the total thermal resistance and the heat transfer rate into the minivan are determined to be

$$R_{total} = R_i + R_{insulation} + R_o = 0.0035 + 0.0125 + 0.0004 = 0.0164 \text{ h}\cdot^\circ\text{F/Btu}$$

$$\dot{Q} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{total}} = \frac{(90 - 70)^\circ\text{F}}{0.0164 \text{ h}\cdot^\circ\text{F/Btu}} = 1220 \text{ Btu/h}$$

12-95 Wind is blowing parallel to the walls of a house with windows. The rate of heat loss through the window is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties Assuming a film temperature of 5°C , the properties of air at 1 atm and this temperature are evaluated to be (Table A-22)

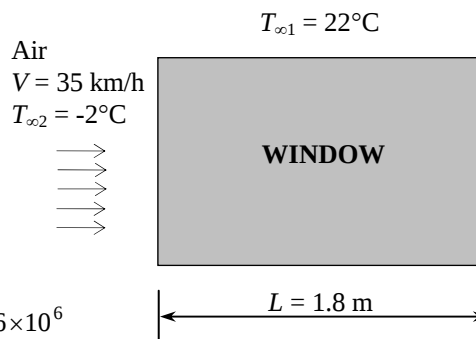
$$k = 0.02401 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.382 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7350$$

Analysis Air flows along 1.8 m side. The Reynolds number in this case is

$$Re_L = \frac{VL}{\nu} = \frac{[(35 \times 1000 / 3600) \text{ m/s}](1.8 \text{ m})}{1.382 \times 10^{-5} \text{ m}^2/\text{s}} = 1.266 \times 10^6$$



which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(1.266 \times 10^6)^{0.8} - 871](0.7350)^{1/3} = 1759$$

$$h = \frac{k}{L} Nu = \frac{0.02401 \text{ W/m}\cdot^\circ\text{C}}{1.8 \text{ m}} (1759) = 23.46 \text{ W/m}^2\cdot^\circ\text{C}$$

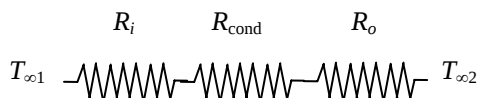
The thermal resistances are

$$A_s = 3(1.8 \text{ m})(1.5 \text{ m}) = 8.1 \text{ m}^2$$

$$R_{conv,i} = \frac{1}{h_i A_s} = \frac{1}{(8 \text{ W/m}^2\cdot^\circ\text{C})(8.1 \text{ m}^2)} = 0.0154^\circ\text{C/W}$$

$$R_{cond} = \frac{L}{k A_s} = \frac{0.005 \text{ m}}{(0.78 \text{ W/m}\cdot^\circ\text{C})(8.1 \text{ m}^2)} = 0.0008^\circ\text{C/W}$$

$$R_{conv,o} = \frac{1}{h_o A_s} = \frac{1}{(23.46 \text{ W/m}^2\cdot^\circ\text{C})(8.1 \text{ m}^2)} = 0.0053^\circ\text{C/W}$$



Then the total thermal resistance and the heat transfer rate through the 3 windows become

$$R_{total} = R_{conv,i} + R_{cond} + R_{conv,o} = 0.0154 + 0.0008 + 0.0053 = 0.0215^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[22 - (-2)]^\circ\text{C}}{0.0215^\circ\text{C/W}} = \mathbf{1116 \text{ W}}$$

12-96 A fan is blowing air over the entire body of a person. The average temperature of the outer surface of the person is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The pressure of air is 1 atm. 4 The average human body can be treated as a 30-cm-diameter cylinder with an exposed surface area of 1.7 m^2 .

Properties We assume the film temperature to be 35°C . The properties of air at 1 atm and this temperature are (Table A-22)

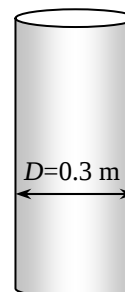
$$k = 0.02625 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.655 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7268$$

$$V = 5 \text{ m/s}$$

$$T_\infty = 32^\circ\text{C}$$



$$\begin{aligned} \text{Person, } T_s \\ 90 \text{ W} \\ \varepsilon = 0.9 \end{aligned}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(5 \text{ m/s})(0.3 \text{ m})}{1.655 \times 10^{-5} \text{ m}^2/\text{s}} = 9.063 \times 10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(9.063 \times 10^4)^{0.5} (0.7268)^{1/3}}{\left[1 + (0.4/0.7268)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{9.063 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 203.6 \end{aligned}$$

Then

$$h = \frac{k}{D} \text{Nu} = \frac{0.02655 \text{ W/m}\cdot^\circ\text{C}}{0.3 \text{ m}} (203.6) = 18.02 \text{ W/m}^2\cdot^\circ\text{C}$$

Considering that there is heat generation in that person's body at a rate of 90 W and body gains heat by radiation from the surrounding surfaces, an energy balance can be written as

$$\dot{Q}_{\text{generated}} + \dot{Q}_{\text{radiation}} = \dot{Q}_{\text{convection}}$$

Substituting values with proper units and then application of trial & error method or the use of an equation solver yields the average temperature of the outer surface of the person.

$$\begin{aligned} 90 \text{ W} + \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) &= h A_s (T_s - T_\infty) \\ 90 + (0.9)(1.7)(5.67 \times 10^{-8})[(40 + 273)^4 - T_s^4] &= (18.02)(1.7)[T_s - (32 + 273)] \\ \longrightarrow T_s &= 309.2 \text{ K} = 36.2^\circ\text{C} \end{aligned}$$

12-97 The heat generated by four transistors mounted on a thin vertical plate is dissipated by air blown over the plate on both surfaces. The temperature of the aluminum plate is to be determined.

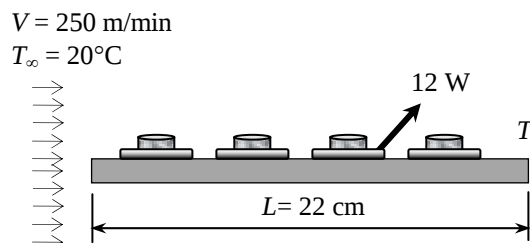
Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 The entire plate is nearly isothermal. 5 The exposed surface area of the transistor is taken to be equal to its base area. 6 Air is an ideal gas with constant properties. 7 The pressure of air is 1 atm.

Properties Assuming a film temperature of 40°C , the properties of air are evaluated to be (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7255$$



Analysis The Reynolds number in this case is

$$Re_L = \frac{VL}{\nu} = \frac{[(250/60) \text{ m/s}](0.22 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 5.386 \times 10^4$$

which is smaller than the critical Reynolds number. Thus we have laminar flow. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664(5.386 \times 10^4)^{0.5} (0.7255)^{1/3} = 138.5$$

$$h = \frac{k}{L} Nu = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{0.22 \text{ m}} (138.5) = 16.75 \text{ W/m}^2\cdot^\circ\text{C}$$

The temperature of aluminum plate then becomes

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 20^\circ\text{C} + \frac{(4 \times 12) \text{ W}}{(16.75 \text{ W/m}^2\cdot^\circ\text{C})[2(0.22 \text{ m})^2]} = 50.0^\circ\text{C}$$

Discussion In reality, the heat transfer coefficient will be higher since the transistors will cause turbulence in the air.

12-98 A spherical tank used to store iced water is subjected to winds. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Thermal resistance of the tank is negligible. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

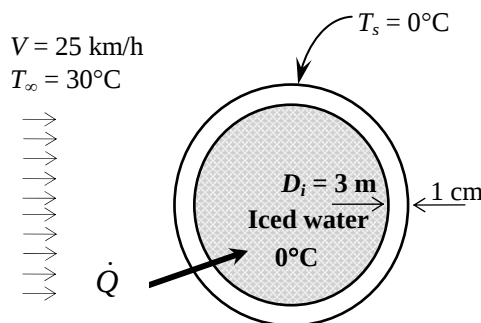
$$\mu_\infty = 1.872 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\mu_{s, @ 0^\circ\text{C}} = 1.729 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\text{Pr} = 0.7282$$

Analysis (a) The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25 \times 1000/3600) \text{ m/s}](3.02 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 1.304 \times 10^6$$



The Nusselt number corresponding to this Reynolds number is determined from

$$\begin{aligned} \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(1.304 \times 10^6)^{0.5} + 0.06(1.304 \times 10^6)^{2/3} \right] (0.7282)^{0.4} \left(\frac{1.872 \times 10^{-5}}{1.729 \times 10^{-5}} \right)^{1/4} = 1056 \end{aligned}$$

and
$$h = \frac{k}{D} \text{Nu} = \frac{0.02588 \text{ W/m} \cdot ^\circ\text{C}}{3.02 \text{ m}} (1056) = 9.05 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer to the iced water is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (9.05 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(3.02 \text{ m})^2](30 - 0)^\circ\text{C} = 7779 \text{ W}$$

(b) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}\Delta t = (7.779 \text{ kJ/s})(24 \times 3600 \text{ s}) = 672,000 \text{ kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{672,000 \text{ kJ}}{333.7 \text{ kJ/kg}} = 2014 \text{ kg}$$

12-99 A spherical tank used to store iced water is subjected to winds. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 7 The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

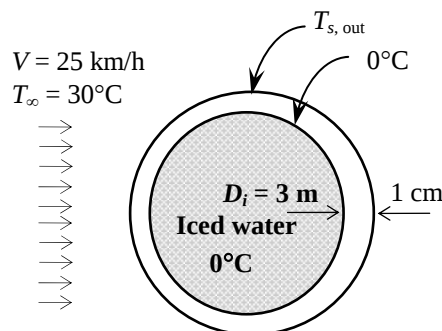
$$\mu_\infty = 1.872 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\mu_{s, @ 0^\circ\text{C}} = 1.729 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\text{Pr} = 0.7282$$

Analysis (a) The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25 \times 1000/3600) \text{ m/s}](3.02 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 1.304 \times 10^6$$



The Nusselt number corresponding to this Reynolds number is determined from

$$\begin{aligned} \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(1.304 \times 10^6)^{0.5} + 0.06(1.304 \times 10^6)^{2/3} \right] (0.7282)^{0.4} \left(\frac{1.872 \times 10^{-5}}{1.729 \times 10^{-5}} \right)^{1/4} = 1056 \end{aligned}$$

and
$$h = \frac{k}{D} \text{Nu} = \frac{0.02588 \text{ W/m} \cdot ^\circ\text{C}}{3.02 \text{ m}} (1056) = 9.05 \text{ W/m}^2 \cdot ^\circ\text{C}$$

In steady operation, heat transfer through the tank by conduction is equal to the heat transfer from the outer surface of the tank by convection and radiation. Therefore,

$$\dot{Q} = \dot{Q}_{\text{through tank}} = \dot{Q}_{\text{from tank, conv + rad}}$$

$$\dot{Q} = \frac{T_{s, \text{out}} - T_{s, \text{in}}}{R_{\text{sphere}}} = h_o A_o (T_{\text{surr}} - T_{s, \text{out}}) + \varepsilon A_o \sigma (T_{\text{surr}}^4 - T_{s, \text{out}}^4)$$

where
$$R_{\text{sphere}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.51 - 1.50) \text{ m}}{4\pi (15 \text{ W/m} \cdot ^\circ\text{C})(1.51 \text{ m})(1.50 \text{ m})} = 2.342 \times 10^{-5} \text{ } ^\circ\text{C/W}$$

$$A_o = \pi D^2 = \pi (3.02 \text{ m})^2 = 28.65 \text{ m}^2$$

Substituting,

$$\begin{aligned} \dot{Q} &= \frac{T_{s, \text{out}} - 0^\circ\text{C}}{2.34 \times 10^{-5} \text{ } ^\circ\text{C/W}} = (9.05 \text{ W/m}^2 \cdot ^\circ\text{C})(28.65 \text{ m}^2)(30 - T_{s, \text{out}})^\circ\text{C} \\ &\quad + (0.75)(28.65 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(25 + 273 \text{ K})^4 - (T_{s, \text{out}} + 273 \text{ K})^4] \end{aligned}$$

whose solution is

$$T_s = 0.25^\circ\text{C} \text{ and } \dot{Q} = 10,530 \text{ W} = \mathbf{10.53 \text{ kW}}$$

(b) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q} \Delta t = (10.531 \text{ kJ/s})(24 \times 3600 \text{ s}) = 909,880 \text{ kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = m h_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{909,880 \text{ kJ}}{333.7 \text{ kJ/kg}} = \mathbf{2727 \text{ kg}}$$

12-100E A cylindrical transistor mounted on a circuit board is cooled by air flowing over it. The maximum power rating of the transistor is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $T_f = (180 + 120) / 2 = 150^\circ\text{F}$ are (Table A-22E)

$$k = 0.01646 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 2.099 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7188$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(500/60 \text{ ft/s})(0.22/12 \text{ ft})}{2.099 \times 10^{-4} \text{ ft}^2/\text{s}} = 727.9$$

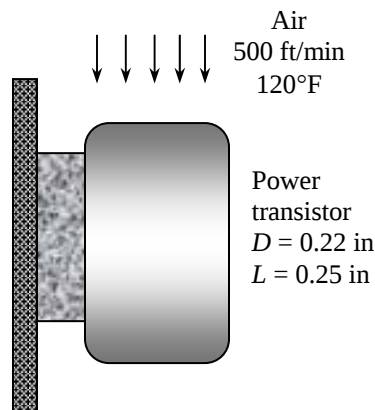
The Nusselt number corresponding to this Reynolds number is

$$\begin{aligned} Nu = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(727.9)^{0.5} (0.7188)^{1/3}}{\left[1 + (0.4/0.7188)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{727.9}{282,000}\right)^{5/8}\right]^{4/5} = 13.72 \end{aligned}$$

$$\text{and } h = \frac{k}{D} Nu = \frac{0.01646 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{(0.22/12 \text{ ft})} (13.72) = 12.32 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

Then the amount of power this transistor can dissipate safely becomes

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) = h(\pi DL)(T_s - T_\infty) \\ &= (12.32 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})[\pi(0.22/12 \text{ ft})(0.25/12 \text{ ft})](180 - 120)^\circ\text{F} \\ &= \mathbf{0.887 \text{ Btu/h} = 0.26 \text{ W}} \quad (1 \text{ W} = 3.412 \text{ Btu/h}) \end{aligned}$$



12-101 Wind is blowing over the roof of a house. The rate of heat transfer through the roof and the cost of this heat loss for 14-h period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

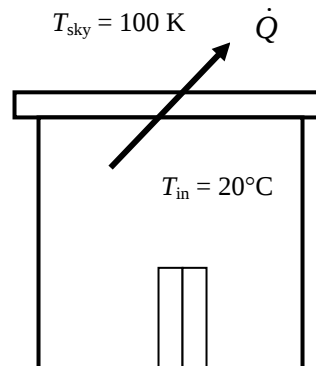
Properties Assuming a film temperature of 10°C , the properties of air are (Table A-22)

$$k = 0.02439 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7336$$

Air
 $V = 60 \text{ km/h}$
 $T_\infty = 10^\circ\text{C}$



Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{[(60 \times 1000 / 3600) \text{ m/s}](20 \text{ m})}{1.426 \times 10^{-5} \text{ m}^2/\text{s}} = 2.338 \times 10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Then the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = \frac{hL}{k} = (0.037 Re_L^{0.8} - 871) Pr^{1/3} = [0.037(2.338 \times 10^7)^{0.8} - 871](0.7336)^{1/3} = 2.542 \times 10^4$$

$$h = \frac{k}{L} Nu = \frac{0.02439 \text{ W/m}\cdot^\circ\text{C}}{20 \text{ m}} (2.542 \times 10^4) = 31.0 \text{ W/m}^2\cdot^\circ\text{C}$$

In steady operation, heat transfer from the room to the roof (by convection and radiation) must be equal to the heat transfer from the roof to the surroundings (by convection and radiation), which must be equal to the heat transfer through the roof by conduction. That is,

$$\dot{Q} = \dot{Q}_{\text{room to roof, conv+rad}} = \dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{roof to surroundings, conv+rad}}$$

Taking the inner and outer surface temperatures of the roof to be $T_{s,in}$ and $T_{s,out}$, respectively, the quantities above can be expressed as

$$\begin{aligned} \dot{Q}_{\text{room to roof, conv+rad}} &= h_i A_s (T_{\text{room}} - T_{s,in}) + \varepsilon A_s \sigma (T_{\text{room}}^4 - T_{s,in}^4) = (5 \text{ W/m}^2\cdot^\circ\text{C})(300 \text{ m}^2)(20 - T_{s,in})^\circ\text{C} \\ &\quad + (0.9)(300 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(20 + 273 \text{ K})^4 - (T_{s,in} + 273 \text{ K})^4] \end{aligned}$$

$$\dot{Q}_{\text{roof, cond}} = k A_s \frac{T_{s,in} - T_{s,out}}{L} = (2 \text{ W/m}\cdot^\circ\text{C})(300 \text{ m}^2) \frac{T_{s,in} - T_{s,out}}{0.15 \text{ m}}$$

$$\begin{aligned} \dot{Q}_{\text{roof to surr, conv+rad}} &= h_o A_s (T_{s,out} - T_{\text{surr}}) + \varepsilon A_s \sigma (T_{s,out}^4 - T_{\text{surr}}^4) = (31.0 \text{ W/m}^2\cdot^\circ\text{C})(300 \text{ m}^2)(T_{s,out} - 10)^\circ\text{C} \\ &\quad + (0.9)(300 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_{s,out} + 273 \text{ K})^4 - (100 \text{ K})^4] \end{aligned}$$

Solving the equations above simultaneously gives

$$\dot{Q} = 28,025 \text{ W} = \mathbf{28.03 \text{ kW}}, \quad T_{s,in} = 10.6^\circ\text{C}, \text{ and } T_{s,out} = 3.5^\circ\text{C}$$

The total amount of natural gas consumption during a 14-hour period is

$$Q_{\text{gas}} = \frac{Q_{\text{total}}}{0.85} = \frac{\dot{Q} \Delta t}{0.85} = \frac{(28.03 \text{ kJ/s})(14 \times 3600 \text{ s})}{0.85} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) = 15.75 \text{ therms}$$

Finally, the money lost through the roof during that period is

$$\text{Money lost} = (15.75 \text{ therms})(\$1.20 / \text{therm}) = \mathbf{\$18.9}$$

12-102 Steam is flowing in a stainless steel pipe while air is flowing across the pipe. The rate of heat loss from the steam per unit length of the pipe is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The pressure of air is 1 atm.

Properties Assuming a film temperature of 10°C, the properties of air are (Table A-22)

$$k = 0.02439 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7336$$

Analysis The outer diameter of insulated pipe is $D_o = 4.6 + 2 \times 3.5 = 11.6 \text{ cm} = 0.116 \text{ m}$. The Reynolds number is

$$Re = \frac{VD_o}{\nu} = \frac{(4 \text{ m/s})(0.116 \text{ m})}{1.426 \times 10^{-5} \text{ m}^2/\text{s}} = 3.254 \times 10^4$$

The Nusselt number for flow across a cylinder is determined from

$$\begin{aligned} Nu &= \frac{hD_o}{k} = 0.3 + \frac{0.62 Re^{0.5} Pr^{1/3}}{\left[1 + (0.4/Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(3.254 \times 10^4)^{0.5} (0.7336)^{1/3}}{\left[1 + (0.4/0.7336)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{3.254 \times 10^4}{282,000}\right)^{5/8}\right]^{4/5} = 107.0 \end{aligned}$$

$$\text{and } h_o = \frac{k}{D_o} Nu = \frac{0.02439 \text{ W/m} \cdot ^\circ\text{C}}{0.116 \text{ m}} (107.0) = 22.50 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Area of the outer surface of the pipe per m length of the pipe is

$$A_o = \pi D_o L = \pi (0.116 \text{ m})(1 \text{ m}) = 0.3644 \text{ m}^2$$

In steady operation, heat transfer from the steam through the pipe and the insulation to the outer surface (by first convection and then conduction) must be equal to the heat transfer from the outer surface to the surroundings (by simultaneous convection and radiation). That is,

$$\dot{Q} = \dot{Q}_{\text{pipe and insulation}} = \dot{Q}_{\text{surface to surroundings}}$$

Using the thermal resistance network, heat transfer from the steam to the outer surface is expressed as

$$R_{\text{conv},i} = \frac{1}{h_i A_i} = \frac{1}{(80 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.04 \text{ m})(1 \text{ m})]} = 0.0995 \text{ } ^\circ\text{C/W}$$

$$R_{\text{pipe}} = \frac{\ln(r_2/r_1)}{2\pi k L} = \frac{\ln(2.3/2)}{2\pi(15 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} = 0.0015 \text{ } ^\circ\text{C/W}$$

$$R_{\text{insulation}} = \frac{\ln(r_3/r_2)}{2\pi k L} = \frac{\ln(5.8/2.3)}{2\pi(0.038 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} = 3.874 \text{ } ^\circ\text{C/W}$$

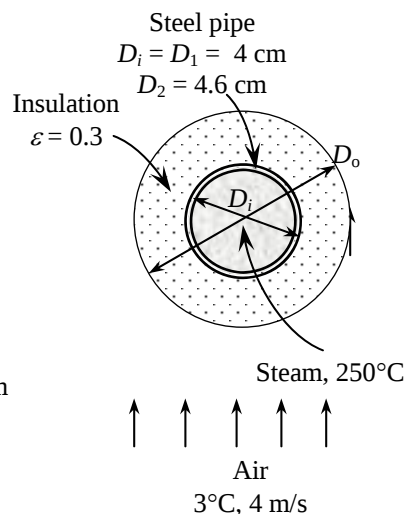
$$\text{and } \dot{Q}_{\text{pipe and ins}} = \frac{T_{\infty 1} - T_s}{R_{\text{conv},i} + R_{\text{pipe}} + R_{\text{insulation}}} = \frac{(250 - T_s)^\circ\text{C}}{(0.0995 + 0.0015 + 3.874)^\circ\text{C/W}}$$

Heat transfer from the outer surface can be expressed as

$$\begin{aligned} \dot{Q}_{\text{surface to surr, conv+rad}} &= h_o A_o (T_s - T_{\text{surr}}) + \epsilon A_o \sigma (T_s^4 - T_{\text{surr}}^4) = (22.50 \text{ W/m}^2 \cdot ^\circ\text{C})(0.3644 \text{ m}^2)(T_s - 3)^\circ\text{C} \\ &\quad + (0.3)(0.3644 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(T_s + 273 \text{ K})^4 - (3 + 273 \text{ K})^4] \end{aligned}$$

Solving the two equations above simultaneously, the surface temperature and the heat transfer rate per m length of the pipe are determined to be

$$T_s = 9.9^\circ\text{C} \text{ and } \dot{Q} = \mathbf{60.4 \text{ W}} \text{ (per m length)}$$



12-103 A spherical tank filled with liquid nitrogen is exposed to winds. The rate of evaporation of the liquid nitrogen due to heat transfer from the air is to be determined for three cases.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 20°C are (Table A-22)

$$k = 0.02514 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.516 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\mu_\infty = 1.825 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\mu_{s, @ -196^\circ\text{C}} = 5.023 \times 10^{-6} \text{ kg/m} \cdot \text{s} \text{ (from EES)}$$

$$\text{Pr} = 0.7309$$

Analysis (a) When there is no insulation, $D = D_i = 4 \text{ m}$, and the Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](4 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 2.932 \times 10^6$$

The Nusselt number is determined from

$$\begin{aligned} \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(2.932 \times 10^6)^{0.5} + 0.06(2.932 \times 10^6)^{2/3} \right] (0.7309)^{0.4} \left(\frac{1.825 \times 10^{-5}}{5.023 \times 10^{-6}} \right)^{1/4} = 2333 \end{aligned}$$

and
$$h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{4 \text{ m}} (2333) = 14.66 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (14.66 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(4 \text{ m})^2][(20 - (-196))^\circ\text{C}] = 159,200 \text{ W}$$

The rate of evaporation of liquid nitrogen then becomes

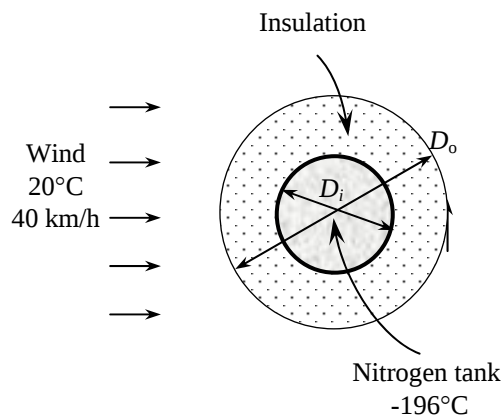
$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{159.2 \text{ kJ/s}}{198 \text{ kJ/kg}} = \mathbf{0.804 \text{ kg/s}}$$

(b) Note that after insulation the outer surface temperature and diameter will change. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C . At -100°C , $\mu = 1.189 \times 10^{-5} \text{ kg/m} \cdot \text{s}$. Noting that $D = D_0 = 4.1 \text{ m}$, the Nusselt number becomes

$$\begin{aligned} \text{Re} &= \frac{VD}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](4.1 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 3.005 \times 10^6 \\ \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(3.005 \times 10^6)^{0.5} + 0.06(3.005 \times 10^6)^{2/3} \right] (0.7309)^{0.4} \left(\frac{1.825 \times 10^{-5}}{1.189 \times 10^{-5}} \right)^{1/4} = 1910 \end{aligned}$$

and
$$h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{4.1 \text{ m}} (1910) = 11.71 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer to the liquid nitrogen is



$$\begin{aligned}
 A_s &= \pi D^2 = \pi (4.1 \text{ m})^2 = 52.81 \text{ m}^2 \\
 \dot{Q} &= \frac{T_\infty - T_{s, \text{tan } k}}{R_{\text{insulation}} + R_{\text{conv}}} = \frac{T_\infty - T_{s, \text{tan } k}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{h A_s}} \\
 &= \frac{[20 - (-196)]^\circ\text{C}}{\frac{(2.05 - 2) \text{ m}}{4\pi (0.035 \text{ W/m}\cdot^\circ\text{C})(2.05 \text{ m})(2 \text{ m})} + \frac{1}{(11.71 \text{ W/m}^2\cdot^\circ\text{C})(52.81 \text{ m}^2)}} = 7361 \text{ W}
 \end{aligned}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m} h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{7.361 \text{ kJ/s}}{198 \text{ kJ/kg}} = \mathbf{0.0372 \text{ kg/s}}$$

(c) We use the dynamic viscosity value at the new estimated surface temperature of 0°C to be $\mu = 1.729 \times 10^{-5} \text{ kg/m}\cdot\text{s}$. Noting that $D = D_0 = 4.04 \text{ m}$ in this case, the Nusselt number becomes

$$\begin{aligned}
 \text{Re} &= \frac{VD}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](4.04 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 2.961 \times 10^6 \\
 \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\
 &= 2 + \left[0.4(2.961 \times 10^6)^{0.5} + 0.06(2.961 \times 10^6)^{2/3} \right] (0.7309)^{0.4} \left(\frac{1.825 \times 10^{-5}}{1.729 \times 10^{-5}} \right)^{1/4} = 1724
 \end{aligned}$$

$$\text{and } h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m}\cdot^\circ\text{C}}{4.04 \text{ m}} (1724) = 10.73 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$\begin{aligned}
 A_s &= \pi D^2 = \pi (4.04 \text{ m})^2 = 51.28 \text{ m}^2 \\
 \dot{Q} &= \frac{T_\infty - T_{s, \text{tan } k}}{R_{\text{insulation}} + R_{\text{conv}}} = \frac{T_\infty - T_{s, \text{tan } k}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{h A_s}} \\
 &= \frac{[20 - (-196)]^\circ\text{C}}{\frac{(2.02 - 2) \text{ m}}{4\pi (0.00005 \text{ W/m}\cdot^\circ\text{C})(2.02 \text{ m})(2 \text{ m})} + \frac{1}{(10.73 \text{ W/m}^2\cdot^\circ\text{C})(51.28 \text{ m}^2)}} = 27.4 \text{ W}
 \end{aligned}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m} h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{0.0274 \text{ kJ/s}}{198 \text{ kJ/kg}} = \mathbf{1.38 \times 10^{-4} \text{ kg/s}}$$

12-104 A spherical tank filled with liquid oxygen is exposed to ambient winds. The rate of evaporation of the liquid oxygen due to heat transfer from the air is to be determined for three cases.

Assumptions 1 Steady operating conditions exist. 2 Radiation effects are negligible. 3 Air is an ideal gas with constant properties. 7 The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 20°C are (Table A-22)

$$k = 0.02514 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.516 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\mu_\infty = 1.825 \times 10^{-5} \text{ kg/m} \cdot \text{s}$$

$$\mu_{s, @ -183^\circ\text{C}} = 6.127 \times 10^{-6} \text{ kg/m} \cdot \text{s} \text{ (from EES)}$$

$$\text{Pr} = 0.7309$$

Analysis (a) When there is no insulation, $D = D_i = 4 \text{ m}$, and the Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](4 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 2.932 \times 10^6$$

The Nusselt number is determined from

$$\begin{aligned} \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(2.932 \times 10^6)^{0.5} + 0.06(2.932 \times 10^6)^{2/3} \right] (0.7309)^{0.4} \left(\frac{1.825 \times 10^{-5}}{6.127 \times 10^{-6}} \right)^{1/4} = 2220 \end{aligned}$$

and
$$h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{4 \text{ m}} (2220) = 13.95 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer to the liquid oxygen is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (13.95 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(4 \text{ m})^2][(20 - (-183))^\circ\text{C}] = 142,372 \text{ W}$$

The rate of evaporation of liquid oxygen then becomes

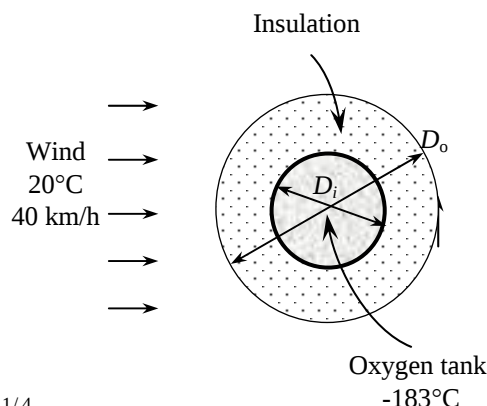
$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{142.4 \text{ kJ/s}}{213 \text{ kJ/kg}} = \mathbf{0.668 \text{ kg/s}}$$

(b) Note that after insulation the outer surface temperature and diameter will change. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C . At -100°C , $\mu = 1.189 \times 10^{-5} \text{ kg/m} \cdot \text{s}$. Noting that $D = D_0 = 4.1 \text{ m}$, the Nusselt number becomes

$$\begin{aligned} \text{Re} &= \frac{V_\infty D}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](4.1 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 3.005 \times 10^6 \\ \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(3.005 \times 10^6)^{0.5} + 0.06(3.005 \times 10^6)^{2/3} \right] (0.7309)^{0.4} \left(\frac{1.825 \times 10^{-5}}{1.189 \times 10^{-5}} \right)^{1/4} = 1910 \end{aligned}$$

and
$$h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{4.1 \text{ m}} (1910) = 11.71 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer to the liquid nitrogen is



$$\begin{aligned}
 A_s &= \pi D^2 = \pi (4.1 \text{ m})^2 = 52.81 \text{ m}^2 \\
 \dot{Q} &= \frac{T_\infty - T_{s, \text{tan } k}}{R_{\text{insulation}} + R_{\text{conv}}} = \frac{T_\infty - T_{s, \text{tan } k}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{h A_s}} \\
 &= \frac{[20 - (-183)]^\circ\text{C}}{\frac{(2.05 - 2) \text{ m}}{4\pi (0.035 \text{ W/m}\cdot^\circ\text{C})(2.05 \text{ m})(2 \text{ m})} + \frac{1}{(11.71 \text{ W/m}^2\cdot^\circ\text{C})(52.81 \text{ m}^2)}} = 6918 \text{ W}
 \end{aligned}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m} h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{6.918 \text{ kJ/s}}{213 \text{ kJ/kg}} = \mathbf{0.0325 \text{ kg/s}}$$

(c) Again we use the dynamic viscosity value at the estimated surface temperature of 0°C to be $\mu = 1.729 \times 10^{-5} \text{ kg/m}\cdot\text{s}$. Noting that $D = D_0 = 4.04 \text{ m}$ in this case, the Nusselt number becomes

$$\begin{aligned}
 \text{Re} &= \frac{VD}{\nu} = \frac{[(40 \times 1000/3600) \text{ m/s}](4.04 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 2.961 \times 10^6 \\
 \text{Nu} &= \frac{hD}{k} = 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4} \\
 &= 2 + \left[0.4(2.961 \times 10^6)^{0.5} + 0.06(2.961 \times 10^6)^{2/3} \right] (0.713)^{0.4} \left(\frac{1.825 \times 10^{-5}}{1.729 \times 10^{-5}} \right)^{1/4} = 1724
 \end{aligned}$$

$$\text{and } h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m}\cdot^\circ\text{C}}{4.04 \text{ m}} (1724) = 10.73 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$\begin{aligned}
 A_s &= \pi D^2 = \pi (4.04 \text{ m})^2 = 51.28 \text{ m}^2 \\
 \dot{Q} &= \frac{T_\infty - T_{s, \text{tan } k}}{R_{\text{insulation}} + R_{\text{conv}}} = \frac{T_\infty - T_{s, \text{tan } k}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{h A_s}} \\
 &= \frac{[20 - (-183)]^\circ\text{C}}{\frac{(2.02 - 2) \text{ m}}{4\pi (0.00005 \text{ W/m}\cdot^\circ\text{C})(2.02 \text{ m})(2 \text{ m})} + \frac{1}{(10.73 \text{ W/m}^2\cdot^\circ\text{C})(51.28 \text{ m}^2)}} = 25.8 \text{ W}
 \end{aligned}$$

The rate of evaporation of liquid oxygen then becomes

$$\dot{Q} = \dot{m} h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{0.0258 \text{ kJ/s}}{213 \text{ kJ/kg}} = \mathbf{1.21 \times 10^{-4} \text{ kg/s}}$$

12-105 A circuit board houses 80 closely spaced logic chips on one side. All the heat generated is conducted across the circuit board and is dissipated from the back side of the board to the ambient air, which is forced to flow over the surface by a fan. The temperatures on the two sides of the circuit board are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. 3 Radiation effects are negligible. 4 Air is an ideal gas with constant properties. 7 The pressure of air is 1 atm.

Properties Assuming a film temperature of 40°C , the properties of air are (Table A-22)

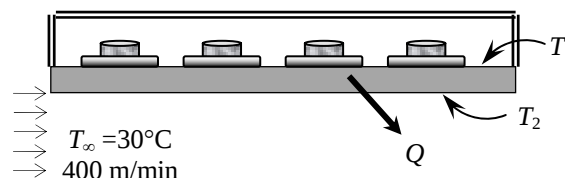
$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7255$$

Analysis The Reynolds number is

$$Re_L = \frac{VL}{\nu} = \frac{[(300/60) \text{ m/s}](0.18 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 5.288 \times 10^4$$



which is less than the critical Reynolds number. Therefore, the flow is laminar. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664(5.288 \times 10^4)^{0.5} (0.7255)^{1/3} = 137.2$$

$$h = \frac{k}{L} Nu = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{0.18 \text{ m}} (137.2) = 20.29 \text{ W/m}^2\cdot^\circ\text{C}$$

The temperatures on the two sides of the circuit board are

$$\begin{aligned} \dot{Q} &= hA_s(T_2 - T_\infty) \rightarrow T_2 = T_\infty + \frac{\dot{Q}}{hA_s} \\ &= 30^\circ\text{C} + \frac{(80 \times 0.06) \text{ W}}{(20.29 \text{ W/m}^2\cdot^\circ\text{C})(0.12 \text{ m})(0.18 \text{ m})} = \mathbf{40.95^\circ\text{C}} \end{aligned}$$

$$\begin{aligned} \dot{Q} &= \frac{kA_s}{L}(T_1 - T_2) \rightarrow T_1 = T_2 + \frac{\dot{Q}L}{kA_s} \\ &= 40.95^\circ\text{C} + \frac{(80 \times 0.06 \text{ W})(0.005 \text{ m})}{(16 \text{ W/m}\cdot^\circ\text{C})(0.12 \text{ m})(0.18 \text{ m})} = \mathbf{41.02^\circ\text{C}} \end{aligned}$$

12-106 ... 12-108 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 13
INTERNAL FORCED CONVECTION

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General Flow Analysis

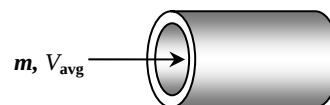
13-1C Liquids are usually transported in circular pipes because pipes with a circular cross-section can withstand large pressure differences between the inside and the outside without undergoing any distortion.

13-2C Reynolds number for flow in a circular tube of diameter D is expressed as

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} \quad \text{where} \quad V_{\infty} = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho(\pi D^2 / 4)} = \frac{4\dot{m}}{\rho \pi D^2} \quad \text{and} \quad \nu = \frac{\mu}{\rho}$$

Substituting,

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{4\dot{m} D}{\rho \pi D^2 (\mu / \rho)} = \frac{4\dot{m}}{\pi D \mu}$$



13-3C Engine oil requires a larger pump because of its much larger density.

13-4C The generally accepted value of the Reynolds number above which the flow in a smooth pipe is turbulent is 4000.

13-5C For flow through non-circular tubes, the Reynolds number as well as the Nusselt number and the friction factor are based on the hydraulic diameter D_h defined as $D_h = \frac{4A_c}{p}$ where A_c is the cross-sectional area of the tube and p is its perimeter. The hydraulic diameter is defined such that it reduces to ordinary diameter D for circular tubes since $D_h = \frac{4A_c}{p} = \frac{4\pi D^2 / 4}{\pi D} = D$.

13-6C The region from the tube inlet to the point at which the boundary layer merges at the centerline is called the *hydrodynamic entry region*, and the length of this region is called *hydrodynamic entry length*. The entry length is much longer in laminar flow than it is in turbulent flow. But at very low Reynolds numbers, L_h is very small ($L_h = 1.2D$ at $\text{Re} = 20$).

13-7C The friction factor is highest at the tube inlet where the thickness of the boundary layer is zero, and decreases gradually to the fully developed value. The same is true for turbulent flow.

13-8C In turbulent flow, the tubes with rough surfaces have much higher friction factors than the tubes with smooth surfaces. In the case of laminar flow, the effect of surface roughness on the friction factor is negligible.

13-9C The friction factor f remains constant along the flow direction in the fully developed region in both laminar and turbulent flow.

13-10C The fluid viscosity is responsible for the development of the velocity boundary layer. For the idealized inviscid fluids (fluids with zero viscosity), there will be no velocity boundary layer.

13-11C The number of transfer units NTU is a measure of the heat transfer area and effectiveness of a heat transfer system. A small value of NTU ($NTU < 5$) indicates more opportunities for heat transfer whereas a large NTU value ($NTU > 5$) indicates that heat transfer will not increase no matter how much we extend the length of the tube.

13-12C The logarithmic mean temperature difference $\Delta T_{\ln}$ is an exact representation of the average temperature difference between the fluid and the surface for the entire tube. It truly reflects the exponential decay of the local temperature difference. The error in using the arithmetic mean temperature increases to undesirable levels when ΔT_e differs from ΔT_i by great amounts. Therefore we should always use the logarithmic mean temperature.

13-13C The region of flow over which the thermal boundary layer develops and reaches the tube center is called the thermal entry region, and the length of this region is called the thermal entry length. The region in which the flow is both hydrodynamically (the velocity profile is fully developed and remains unchanged) and thermally (the dimensionless temperature profile remains unchanged) developed is called the fully developed region.

13-14C The heat flux will be higher near the inlet because the heat transfer coefficient is highest at the tube inlet where the thickness of thermal boundary layer is zero, and decreases gradually to the fully developed value.

13-15C The heat flux will be higher near the inlet because the heat transfer coefficient is highest at the tube inlet where the thickness of thermal boundary layer is zero, and decreases gradually to the fully developed value.

13-16C In the fully developed region of flow in a circular tube, the velocity profile will not change in the flow direction but the temperature profile may.

13-17C The hydrodynamic and thermal entry lengths are given as $L_h = 0.05 \text{ Re } D$ and $L_t = 0.05 \text{ Re Pr } D$ for laminar flow, and $L_h \approx L_t \approx 10D$ in turbulent flow. Noting that $\text{Pr} \gg 1$ for oils, the thermal entry length is larger than the hydrodynamic entry length in laminar flow. In turbulent, the hydrodynamic and thermal entry lengths are independent of Re or Pr numbers, and are comparable in magnitude.

13-18C The hydrodynamic and thermal entry lengths are given as $L_h = 0.05 \text{ Re } D$ and $L_t = 0.05 \text{ Re Pr } D$ for laminar flow, and $L_h \approx L_t \approx 10D$ in turbulent flow. Noting that $\text{Pr} \ll 1$ for liquid metals, the thermal entry length is smaller than the hydrodynamic entry length in laminar flow. In turbulent, the hydrodynamic and thermal entry lengths are independent of Re or Pr numbers, and are comparable in magnitude.

13-19C In fluid flow, it is convenient to work with an average or mean velocity V_{avg} and an average or mean temperature T_m which remain constant in incompressible flow when the cross-sectional area of the tube is constant. The V_{avg} and T_m represent the velocity and temperature, respectively, at a cross section if all the particles were at the same velocity and temperature.

13-20C When the surface temperature of tube is constant, the appropriate temperature difference for use in the Newton's law of cooling is logarithmic mean temperature difference that can be expressed as

$$\Delta T_{\ln} = \frac{\Delta T_e - \Delta T_i}{\ln(\Delta T_e / \Delta T_i)}$$

13-21 Air flows inside a duct and it is cooled by water outside. The exit temperature of air and the rate of heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature of the duct is constant. 3 The thermal resistance of the duct is negligible.

Properties The properties of air at the anticipated average temperature of 30°C are (Table A-22)

$$\rho = 1.164 \text{ kg/m}^3$$

$$c_p = 1007 \text{ J/kg} \cdot ^\circ\text{C}$$

Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho \left(\frac{\pi D^2}{4} \right) V_{\text{avg}} = (1.164 \text{ kg/m}^3) \frac{\pi (0.25 \text{ m})^2}{4} (7 \text{ m/s}) = 0.400 \text{ kg/s}$$

$$A_s = \pi D L = \pi (0.25 \text{ m})(12 \text{ m}) = 9.425 \text{ m}^2$$

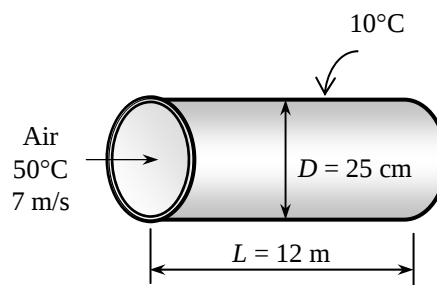
The exit temperature of air is determined from

$$T_e = T_s - (T_s - T_i) e^{-h A_s / (\dot{m} c_p)} = 10 - (10 - 50) e^{-\frac{(85)(9.425)}{(0.400)(1007)}} = 15.47^\circ\text{C}$$

The logarithmic mean temperature difference and the rate of heat transfer are

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln \left(\frac{T_s - T_e}{T_s - T_i} \right)} = \frac{15.47 - 50}{\ln \left(\frac{10 - 15.47}{10 - 50} \right)} = 17.36^\circ\text{C}$$

$$\dot{Q} = h A_s \Delta T_{\text{ln}} = (85 \text{ W/m}^2 \cdot ^\circ\text{C})(9.425 \text{ m}^2)(17.36^\circ\text{C}) = 13,900 \text{ W} = \mathbf{13.9 \text{ kW}}$$



13-22 Steam is condensed by cooling water flowing inside copper tubes. The average heat transfer coefficient and the number of tubes needed are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature of the pipe is constant. 3 The thermal resistance of the pipe is negligible.

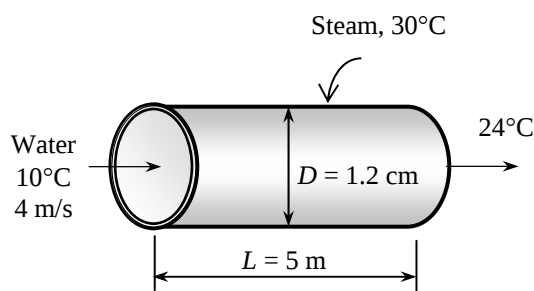
Properties The properties of water at the average temperature of $(10+24)/2=17^\circ\text{C}$ are (Table A-15)

$$\rho = 998.7 \text{ kg/m}^3$$

$$c_p = 4183.8 \text{ J/kg}\cdot^\circ\text{C}$$

Also, the heat of vaporization of water at 30°C is $h_{fg} = 2431 \text{ kJ/kg}$.

Analysis The mass flow rate of water and the surface area are



$$\dot{m} = \rho A_c V_{\text{avg}} = \rho \left(\frac{\pi D^2}{4} \right) V_{\text{avg}} = (998.7 \text{ kg/m}^3) \frac{\pi (0.012 \text{ m})^2}{4} (4 \text{ m/s}) = 0.4518 \text{ kg/s}$$

The rate of heat transfer for one tube is

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (0.4518 \text{ kg/s})(4183.8 \text{ J/kg}\cdot^\circ\text{C})(24 - 10^\circ\text{C}) = 26,460 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln \left(\frac{T_s - T_e}{T_s - T_i} \right)} = \frac{24 - 10}{\ln \left(\frac{30 - 24}{30 - 10} \right)} = 11.63^\circ\text{C}$$

$$A_s = \pi D L = \pi (0.012 \text{ m})(5 \text{ m}) = 0.1885 \text{ m}^2$$

The average heat transfer coefficient is determined from

$$\dot{Q} = h A_s \Delta T_{\ln} \longrightarrow h = \frac{\dot{Q}}{A_s \Delta T_{\ln}} = \frac{26,460 \text{ W}}{(0.1885 \text{ m}^2)(11.63^\circ\text{C})} \left(\frac{1 \text{ kW}}{1000 \text{ W}} \right) = 12.1 \text{ kW/m}^2\cdot^\circ\text{C}$$

The total rate of heat transfer is determined from

$$\dot{Q}_{\text{total}} = \dot{m}_{\text{cond}} h_{fg} = (0.15 \text{ kg/s})(2431 \text{ kJ/kg}) = 364.65 \text{ kW}$$

Then the number of tubes becomes

$$N_{\text{tube}} = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{364,650 \text{ W}}{26,460 \text{ W}} = 13.8$$

13-23 Steam is condensed by cooling water flowing inside copper tubes. The average heat transfer coefficient and the number of tubes needed are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature of the pipe is constant. 3 The thermal resistance of the pipe is negligible.

Properties The properties of water at the average temperature of $(10+24)/2=17^\circ\text{C}$ are (Table A-15)

$$\rho = 998.7 \text{ kg/m}^3$$

$$c_p = 4183.8 \text{ J/kg}\cdot^\circ\text{C}$$

Also, the heat of vaporization of water at 30°C is $h_{fg} = 2431 \text{ kJ/kg}$.

Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho \left(\frac{\pi D^2}{4} \right) V_{\text{avg}} = (998.7 \text{ kg/m}^3) \frac{\pi (0.012 \text{ m})^2}{4} (4 \text{ m/s}) = 0.4518 \text{ kg/s}$$

The rate of heat transfer for one tube is

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (0.4518 \text{ kg/s})(4183.8 \text{ J/kg}\cdot^\circ\text{C})(24 - 10^\circ\text{C}) = 26,460 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln \left(\frac{T_s - T_e}{T_s - T_i} \right)} = \frac{24 - 10}{\ln \left(\frac{30 - 24}{30 - 10} \right)} = 11.63^\circ\text{C}$$

$$A_s = \pi D L = \pi (0.012 \text{ m})(5 \text{ m}) = 0.1885 \text{ m}^2$$

The average heat transfer coefficient is determined from

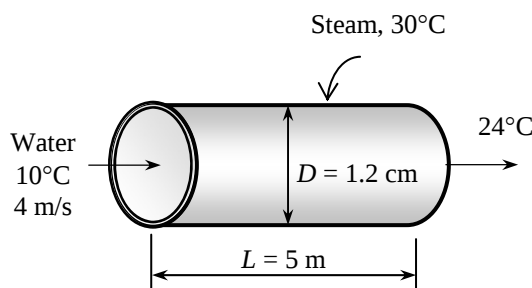
$$\dot{Q} = h A_s \Delta T_{\ln} \longrightarrow h = \frac{\dot{Q}}{A_s \Delta T_{\ln}} = \frac{26,460 \text{ W}}{(0.1885 \text{ m}^2)(11.63^\circ\text{C})} \left(\frac{1 \text{ kW}}{1000 \text{ W}} \right) = 12.1 \text{ kW/m}^2\cdot^\circ\text{C}$$

The total rate of heat transfer is determined from

$$\dot{Q}_{\text{total}} = \dot{m}_{\text{cond}} h_{fg} = (0.60 \text{ kg/s})(2431 \text{ kJ/kg}) = 1458.6 \text{ kW}$$

Then the number of tubes becomes

$$N_{\text{tube}} = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{1,458,600 \text{ W}}{26,460 \text{ W}} = 55.1$$



13-24 Combustion gases passing through a tube are used to vaporize waste water. The tube length and the rate of evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature of the pipe is constant. 3 The thermal resistance of the pipe is negligible. 4 Air properties are to be used for exhaust gases.

Properties The properties of air at the average temperature of $(250+150)/2=200^\circ\text{C}$ are (Table A-22)

$$c_p = 1023 \text{ J/kg}\cdot^\circ\text{C}$$

$$R = 0.287 \text{ kJ/kg}\cdot\text{K}$$

Also, the heat of vaporization of water at 1 atm or 100°C is $h_{fg} = 2257 \text{ kJ/kg}$ (Table A-15).

Analysis The density of air at the inlet and the mass flow rate of exhaust gases are

$$\rho = \frac{P}{RT} = \frac{115 \text{ kPa}}{(0.287 \text{ kJ/kg}\cdot\text{K})(250 + 273 \text{ K})} = 0.7662 \text{ kg/m}^3$$

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho \left(\frac{\pi D^2}{4} \right) V_{\text{avg}} = (0.7662 \text{ kg/m}^3) \frac{\pi (0.03 \text{ m})^2}{4} (5 \text{ m/s}) = 0.002708 \text{ kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m} c_p (T_i - T_e) = (0.002708 \text{ kg/s})(1023 \text{ J/kg}\cdot^\circ\text{C})(250 - 150^\circ\text{C}) = 277.0 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln \left(\frac{T_s - T_e}{T_s - T_i} \right)} = \frac{150 - 250}{\ln \left(\frac{110 - 150}{110 - 250} \right)} = 79.82^\circ\text{C}$$

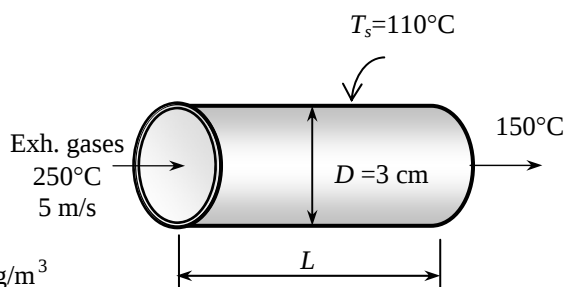
$$\dot{Q} = h A_s \Delta T_{\text{ln}} \longrightarrow A_s = \frac{\dot{Q}}{h \Delta T_{\text{ln}}} = \frac{277.0 \text{ W}}{(120 \text{ W/m}^2\cdot^\circ\text{C})(79.82^\circ\text{C})} = 0.02891 \text{ m}^2$$

Then the tube length becomes

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.02891 \text{ m}^2}{\pi (0.03 \text{ m})} = 0.3067 \text{ m} = \mathbf{30.7 \text{ cm}}$$

The rate of evaporation of water is determined from

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} \longrightarrow \dot{m}_{\text{evap}} = \frac{\dot{Q}}{h_{fg}} = \frac{(0.2770 \text{ kW})}{(2257 \text{ kJ/kg})} = 0.0001227 \text{ kg/s} = \mathbf{0.442 \text{ kg/h}}$$



13-25 Combustion gases passing through a tube are used to vaporize waste water. The tube length and the rate of evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature of the pipe is constant. 3 The thermal resistance of the pipe is negligible. 4 Air properties are to be used for exhaust gases.

Properties The properties of air at the average temperature of $(250+150)/2=200^\circ\text{C}$ are (Table A-22)

$$c_p = 1023 \text{ J/kg}\cdot^\circ\text{C}$$

$$R = 0.287 \text{ kJ/kg}\cdot\text{K}$$

Also, the heat of vaporization of water at 1 atm or 100°C is $h_{fg} = 2257 \text{ kJ/kg}$ (Table A-15).

Analysis The density of air at the inlet and the mass flow rate of exhaust gases are

$$\rho = \frac{P}{RT} = \frac{115 \text{ kPa}}{(0.287 \text{ kJ/kg}\cdot\text{K})(250 + 273 \text{ K})} = 0.7662 \text{ kg/m}^3$$

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho \left(\frac{\pi D^2}{4} \right) V_{\text{avg}} = (0.7662 \text{ kg/m}^3) \frac{\pi (0.03 \text{ m})^2}{4} (5 \text{ m/s}) = 0.002708 \text{ kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m} c_p (T_i - T_e) = (0.002708 \text{ kg/s})(1023 \text{ J/kg}\cdot^\circ\text{C})(250 - 150^\circ\text{C}) = 277.0 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln \left(\frac{T_s - T_e}{T_s - T_i} \right)} = \frac{150 - 250}{\ln \left(\frac{110 - 150}{110 - 250} \right)} = 79.82^\circ\text{C}$$

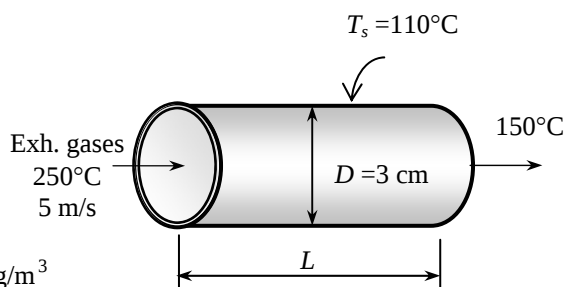
$$\dot{Q} = h A_s \Delta T_{\text{ln}} \longrightarrow A_s = \frac{\dot{Q}}{h \Delta T_{\text{ln}}} = \frac{277.0 \text{ W}}{(40 \text{ W/m}^2\cdot^\circ\text{C})(79.82^\circ\text{C})} = 0.08673 \text{ m}^2$$

Then the tube length becomes

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.08673 \text{ m}^2}{\pi (0.03 \text{ m})} = 0.920 \text{ m} = \mathbf{92.0 \text{ cm}}$$

The rate of evaporation of water is determined from

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} \longrightarrow \dot{m}_{\text{evap}} = \frac{\dot{Q}}{h_{fg}} = \frac{(0.2770 \text{ kW})}{(2257 \text{ kJ/kg})} = 0.0001227 \text{ kg/s} = \mathbf{0.442 \text{ kg/h}}$$



Laminar and Turbulent Flow in Tubes

13-26C The friction factor for flow in a tube is proportional to the pressure drop. Since the pressure drop along the flow is directly related to the power requirements of the pump to maintain flow, the friction factor is also proportional to the power requirements. The applicable relations are

$$\Delta P = f \frac{L}{D} \frac{\rho V^2}{2} \quad \text{and} \quad \dot{W}_{\text{pump}} = \frac{\dot{m} \Delta P}{\rho}$$

13-27C The shear stress at the center of a circular tube during fully developed laminar flow is zero since the shear stress is proportional to the velocity gradient, which is zero at the tube center.

13-28C Yes, the shear stress at the surface of a tube during fully developed turbulent flow is maximum since the shear stress is proportional to the velocity gradient, which is maximum at the tube surface.

13-29C In fully developed flow in a circular pipe with negligible entrance effects, if the length of the pipe is doubled, the pressure drop will also *double* (the pressure drop is proportional to length).

13-30C Yes, the volume flow rate in a circular pipe with laminar flow can be determined by measuring the velocity at the centerline in the fully developed region, multiplying it by the cross-sectional area, and dividing the result by 2 since $\dot{V} = V_{\text{avg}} A_c = (V_{\text{max}} / 2) A_c$.

13-31C No, the average velocity in a circular pipe in fully developed laminar flow **cannot** be determined by simply measuring the velocity at $R/2$ (midway between the wall surface and the centerline). The mean velocity is $V_{\text{max}}/2$, but the velocity at $R/2$ is

$$V(R/2) = V_{\text{max}} \left(1 - \frac{r^2}{R^2} \right)_{r=R/2} = \frac{3V_{\text{max}}}{4}$$

13-32C In fully developed laminar flow in a circular pipe, the pressure drop is given by

$$\Delta P = \frac{8\mu L V_{\text{avg}}}{R^2} = \frac{32\mu L V_{\text{avg}}}{D^2}$$

The mean velocity can be expressed in terms of the flow rate as $V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2 / 4}$. Substituting,

$$\Delta P = \frac{8\mu L V_{\text{avg}}}{R^2} = \frac{32\mu L V_{\text{avg}}}{D^2} = \frac{32\mu L}{D^2} \frac{\dot{V}}{\pi D^2 / 4} = \frac{128\mu L \dot{V}}{\pi D^4}$$

Therefore, at constant flow rate and pipe length, the pressure drop is inversely proportional to the 4th power of diameter, and thus reducing the pipe diameter by half will increase the pressure drop **by a factor of 16**.

13-33C In fully developed laminar flow in a circular pipe, the pressure drop is given by

$$\Delta P = \frac{8\mu L V_{\text{avg}}}{R^2} = \frac{32\mu L V_{\text{avg}}}{D^2}$$

When the flow rate and thus mean velocity are held constant, the pressure drop becomes proportional to viscosity. Therefore, pressure drop will be **reduced by half** when the viscosity is reduced by half.

13-34C The tubes with rough surfaces have much higher heat transfer coefficients than the tubes with smooth surfaces. In the case of laminar flow, the effect of surface roughness on the heat transfer coefficient is negligible.

13-35 The flow rate through a specified water pipe is given. The pressure drop and the pumping power requirements are to be determined.

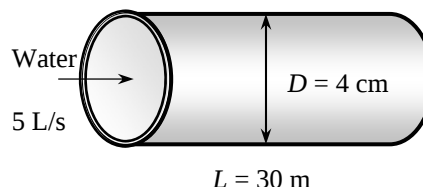
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be $\rho = 999.1 \text{ kg/m}^3$ and $\mu = 1.138 \times 10^{-3} \text{ kg/m}\cdot\text{s}$, respectively. The roughness of stainless steel is 0.002 mm (Table 13-3).

Analysis First, we calculate the mean velocity and the Reynolds number to determine the flow regime:

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2 / 4} = \frac{0.005 \text{ m}^3 / \text{s}}{\pi (0.04 \text{ m})^2 / 4} = 3.98 \text{ m/s}$$

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{(999.1 \text{ kg/m}^3)(3.98 \text{ m/s})(0.04 \text{ m})}{1.138 \times 10^{-3} \text{ kg/m}\cdot\text{s}} = 1.40 \times 10^5$$



which is greater than 10,000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D = \frac{2 \times 10^{-6} \text{ m}}{0.04 \text{ m}} = 5 \times 10^{-5}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0 \log \left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re} \sqrt{f}} \right) \rightarrow \frac{1}{\sqrt{f}} = -2.0 \log \left(\frac{5 \times 10^{-5}}{3.7} + \frac{2.51}{1.40 \times 10^5 \sqrt{f}} \right)$$

It gives $f = 0.0171$. Then the pressure drop and the required power input become

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.0171 \frac{30 \text{ m}}{0.04 \text{ m}} \frac{(999.1 \text{ kg/m}^3)(3.98 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) = 101.5 \text{ kPa}$$

$$\dot{W}_{\text{pump,u}} = \dot{V} \Delta P = (0.005 \text{ m}^3 / \text{s})(101.5 \text{ kPa}) \left(\frac{1 \text{ kW}}{1 \text{ kPa}\cdot\text{m}^3/\text{s}} \right) = \mathbf{0.508 \text{ kW}}$$

Therefore, useful power input in the amount of 0.508 kW is needed to overcome the frictional losses in the pipe.

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give $f = 0.0169$, which is sufficiently close to 0.0171. Also, the friction factor corresponding to $\varepsilon = 0$ in this case is 0.0168, which indicates that stainless steel pipes can be assumed to be smooth with an error of about 2%. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

13-36 In fully developed laminar flow in a circular pipe, the velocity at $r = R/2$ is measured. The velocity at the center of the pipe ($r = 0$) is to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is given by

$$V(r) = V_{\max} \left(1 - \frac{r^2}{R^2} \right)$$

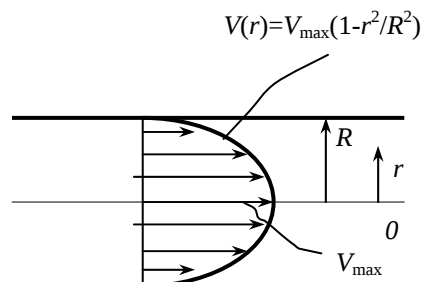
where $V_{\max}$ is the maximum velocity which occurs at pipe center, $r = 0$. At $r = R/2$,

$$V(R/2) = V_{\max} \left(1 - \frac{(R/2)^2}{R^2} \right) = V_{\max} \left(1 - \frac{1}{4} \right) = \frac{3V_{\max}}{4}$$

Solving for $V_{\max}$ and substituting,

$$V_{\max} = \frac{4V(R/2)}{3} = \frac{4(6 \text{ m/s})}{3} = \mathbf{8 \text{ m/s}}$$

which is the velocity at the pipe center.



13-37 The velocity profile in fully developed laminar flow in a circular pipe is given. The mean and maximum velocities are to be determined.

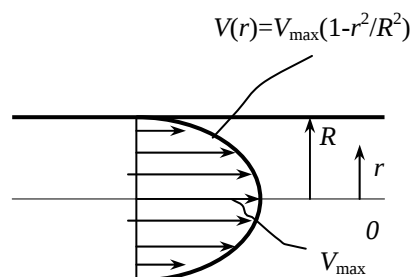
Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is given by

$$V(r) = V_{\max} \left(1 - \frac{r^2}{R^2} \right)$$

The velocity profile in this case is given by

$$V(r) = 4(1 - r^2 / R^2)$$



Comparing the two relations above gives the maximum velocity to be $V_{\max} = 4 \text{ m/s}$. Then the mean velocity and volume flow rate become

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{4 \text{ m/s}}{2} = \mathbf{2 \text{ m/s}}$$

$$\dot{V} = V_{\text{avg}} A_c = V_{\text{avg}} (\pi R^2) = (2 \text{ m/s}) [\pi (0.10 \text{ m})^2] = \mathbf{0.0628 \text{ m}^3/\text{s}}$$

13-38 The velocity profile in fully developed laminar flow in a circular pipe is given. The mean and maximum velocities are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is given by

$$V(r) = V_{\max} \left(1 - \frac{r^2}{R^2} \right)$$

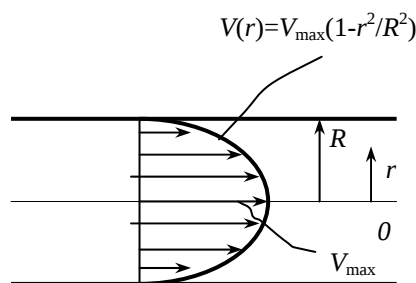
The velocity profile in this case is given by

$$V(r) = 4(1 - r^2 / R^2)$$

Comparing the two relations above gives the maximum velocity to be $V_{\max} = 4 \text{ m/s}$. Then the mean velocity and volume flow rate become

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{4 \text{ m/s}}{2} = 2 \text{ m/s}$$

$$\dot{V} = V_{\text{avg}} A_c = V_{\text{avg}} (\pi R^2) = (2 \text{ m/s}) [\pi (0.05 \text{ m})^2] = 0.0157 \text{ m}^3/\text{s}$$



13-39 The convection heat transfer coefficients for the flow of air and water are to be determined under similar conditions.

Assumptions 1 Steady flow conditions exist. 2 The surface heat flux is uniform. 3 The inner surfaces of the tube are smooth.

Properties The properties of air at 25°C are (Table A-22)

$$k = 0.02551 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7296$$

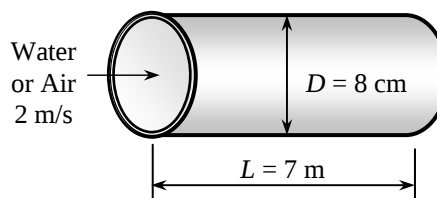
The properties of water at 25°C are (Table A-15)

$$\rho = 997 \text{ kg/m}^3$$

$$k = 0.607 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = \mu / \rho = 0.891 \times 10^{-3} / 997 = 8.937 \times 10^{-7} \text{ m}^2/\text{s}$$

$$\text{Pr} = 6.14$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.08 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 10,243$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.08 \text{ m}) = 0.8 \text{ m}$$

which is much shorter than the total length of the tube. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(10,243)^{0.8} (0.7296)^{0.4} = 32.76$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02551 \text{ W/m} \cdot ^\circ\text{C}}{0.08 \text{ m}} (32.76) = \mathbf{10.45 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

Repeating calculations for water:

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.08 \text{ m})}{8.937 \times 10^{-7} \text{ m}^2/\text{s}} = 179,035$$

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(179,035)^{0.8} (6.14)^{0.4} = 757.4$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m} \cdot ^\circ\text{C}}{0.08 \text{ m}} (757.4) = \mathbf{5747 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

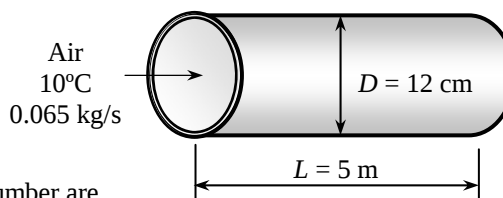
Discussion The heat transfer coefficient for water is 550 times that of air.

13-40 Air flows in a pipe whose inner surface is not smooth. The rate of heat transfer is to be determined using two different Nusselt number relations.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The pressure of air is 1 atm.

Properties Assuming a bulk mean fluid temperature of 20°C, the properties of air are (Table A-22)

$$\begin{aligned}\rho &= 1.204 \text{ kg/m}^3 \\ k &= 0.02514 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.516 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7309\end{aligned}$$



Analysis The mean velocity of air and the Reynolds number are

$$\begin{aligned}V_{\text{avg}} &= \frac{\dot{m}}{\rho A_c} = \frac{0.065 \text{ kg/s}}{(1.204 \text{ kg/m}^3)\pi(0.12 \text{ m})^2/4} = 4.773 \text{ m/s} \\ \text{Re} &= \frac{V_{\text{avg}} D}{\nu} = \frac{(4.773 \text{ m/s})(0.12 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 37,785\end{aligned}$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.12 \text{ m}) = 1.2 \text{ m}$$

which is much shorter than the total length of the pipe. Therefore, we can assume fully developed turbulent flow in the entire duct. The friction factor may be determined from Colebrook equation using EES to be

$$\frac{1}{\sqrt{f}} = -2 \log \left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re} \sqrt{f}} \right) \longrightarrow \frac{1}{\sqrt{f}} = -2 \log \left(\frac{0.00022/0.12}{3.7} + \frac{2.51}{37,785 \sqrt{f}} \right) \longrightarrow f = 0.02695$$

The Nusselt number from Eq. 13-66 is

$$\text{Nu} = 0.125 f \text{ Re Pr}^{1/3} = 0.125(0.02695)(37,785)(0.7309)^{1/3} = 114.7$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{0.12 \text{ m}} (114.7) = 24.02 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Next we determine the exit temperature of air

$$A = \pi DL = \pi(0.12 \text{ m})(5 \text{ m}) = 1.885 \text{ m}^2$$

$$T_e = T_s - (T_s - T_i) e^{-hA/(\dot{m}c_p)} = 50 - (50 - 10) e^{-\frac{(24.02)(1.885)}{(0.065)(1007)}} = 30.0^\circ\text{C}$$

This result verifies our assumption of bulk mean fluid temperature that we used for property evaluation. Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.065 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})(30.0 - 10)^\circ\text{C} = \mathbf{1307 \text{ W}}$$

Repeating the calculations using the Nusselt number from Eq. 13-70:

$$\text{Nu} = \frac{(f/8)(\text{Re} - 1000) \text{Pr}}{1 + 12.7(f/8)^{0.5}(\text{Pr}^{2/3} - 1)} = \frac{(0.02695/8)(37,785 - 1000)(0.7309)}{1 + 12.7(0.02695/8)^{0.5}(0.7309^{2/3} - 1)} = 105.2$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{0.12 \text{ m}} (105.2) = 22.04 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$T_e = T_s - (T_s - T_i) e^{-hA/(\dot{m}c_p)} = 50 - (50 - 10) e^{-\frac{(22.04)(1.885)}{(0.065)(1007)}} = 28.8^\circ\text{C}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.065 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})(28.8 - 10)^\circ\text{C} = \mathbf{1230 \text{ W}}$$

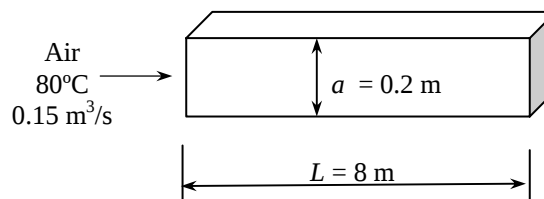
The result by Eq. 13-66 is about 6 percent greater than that by Eq. 13-70.

13-41 Air flows in a square cross section pipe. The rate of heat loss and the pressure difference between the inlet and outlet sections of the duct are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The pressure of air is 1 atm.

Properties Taking a bulk mean fluid temperature of 80°C assuming that the air does not lose much heat to the attic, the properties of air are (Table A-22)

$$\begin{aligned}\rho &= 0.9994 \text{ kg/m}^3 \\ k &= 0.02953 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 2.097 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1008 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7154\end{aligned}$$



Analysis The mean velocity of air, the hydraulic diameter, and the Reynolds number are

$$\begin{aligned}V &= \frac{\dot{V}}{A} = \frac{0.15 \text{ m}^3/\text{s}}{(0.2 \text{ m})^2} = 3.75 \text{ m/s} \\ D_h &= \frac{4A}{P} = \frac{4a^2}{4a} = a = 0.2 \text{ m} \\ \text{Re} &= \frac{VD_h}{\nu} = \frac{(3.75 \text{ m/s})(0.2 \text{ m})}{2.097 \times 10^{-5}} = 35,765\end{aligned}$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10(0.2 \text{ m}) = 2 \text{ m}$$

which is much shorter than the total length of the pipe. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(35,765)^{0.8} (0.7154)^{0.3} = 91.4$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.02953 \text{ W/m} \cdot ^\circ\text{C}}{0.2 \text{ m}} (91.4) = 13.5 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Next we determine the exit temperature of air

$$A = 4aL = 4(0.2 \text{ m})(8 \text{ m}) = 6.4 \text{ m}^2$$

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}c_p)} = 60 - (60 - 80)e^{-\frac{(13.5)(6.4)}{(0.9994)(0.15)(1008)}} = 71.3^\circ\text{C}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.9994 \text{ kg/m}^3)(0.15 \text{ m}^3/\text{s})(1008 \text{ J/kg} \cdot ^\circ\text{C})(80 - 71.3)^\circ\text{C} = \mathbf{1315 \text{ W}}$$

From Moody chart:

$$\text{Re} = 35,765 \text{ and } \varepsilon/D = 0.001 \rightarrow f = 0.026$$

Then the pressure drop is determined to be

$$\Delta P = f \frac{\rho V^2}{2D} L = (0.026) \frac{(0.9994 \text{ kg/m}^3)(3.75 \text{ m/s})^2}{2(0.2 \text{ m})} (8 \text{ m}) = \mathbf{7.3 \text{ Pa}}$$

13-42 A liquid is heated as it flows in a pipe that is wrapped by electric resistance heaters. The required surface heat flux, the surface temperature at the exit, and the pressure loss through the pipe and the minimum power required to overcome the resistance to flow are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The surface heat flux is uniform. 3 The inner surfaces of the tube are smooth. 4 Heat transfer to the surroundings is negligible.

Properties The properties of the fluid are given to be $\rho = 1000 \text{ kg/m}^3$, $c_p = 4000 \text{ J/kg}\cdot\text{K}$, $\mu = 2 \times 10^{-3} \text{ kg/s}\cdot\text{m}$, $k = 0.48 \text{ W/m}\cdot\text{K}$, and $\text{Pr} = 10$

Analysis (a) The mass flow rate of the liquid is

$$\dot{m} = \rho AV = (1000 \text{ kg/m}^3) \left(\pi (0.01 \text{ m})^2 / 4 \right) (0.8 \text{ m/s}) = 0.0628 \text{ kg/s}$$

The rate of heat transfer and the heat flux are

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (0.0628 \text{ kg/s}) (4000 \text{ J/kg}\cdot^\circ\text{C}) (75 - 25)^\circ\text{C} = 12,560 \text{ W}$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{12,560 \text{ W}}{\pi (0.01 \text{ m}) (10 \text{ m})} = \mathbf{40,000 \text{ W/m}^2}$$

(b) The Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(1000 \text{ kg/m}^3) (0.8 \text{ m/s}) (0.01 \text{ m})}{0.002 \text{ kg/m}\cdot\text{s}} = 4000$$

which is greater than 2300 and smaller than 10,000. Therefore, we have transitional flow. However, we use turbulent flow relation. The entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.01 \text{ m}) = 0.1 \text{ m}$$

which is much shorter than the total length of the tube. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(4000)^{0.8} (10)^{0.4} = 44$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.48 \text{ W/m}\cdot^\circ\text{C}}{0.01 \text{ m}} (44) = 2112 \text{ W/m}^2\cdot^\circ\text{C}$$

The surface temperature at the exit is

$$\dot{q} = h(T_s - T_e) \longrightarrow 40,000 \text{ W} = (2112 \text{ W/m}^2\cdot^\circ\text{C}) (T_s - 75)^\circ\text{C} \longrightarrow T_s = \mathbf{93.9^\circ\text{C}}$$

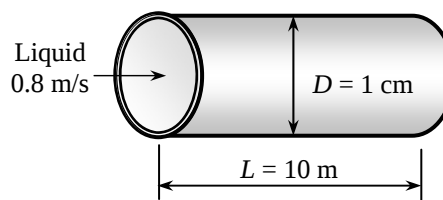
(c) From Moody chart:

$$\text{Re} = 4000, \varepsilon/D = 0.046 \text{ mm} / 10 = 0.0045 \rightarrow f = 0.044$$

Then the pressure drop and the minimum power required to overcome this pressure drop are determined to be

$$\Delta P = f \frac{\rho V^2}{2D} L = (0.044) \frac{(1000 \text{ kg/m}^3) (0.8 \text{ m/s})^2}{2(0.01 \text{ m})} (10 \text{ m}) = \mathbf{14,080 \text{ Pa}}$$

$$\dot{W} = \dot{V} \Delta P = \left(\pi (0.01 \text{ m})^2 / 4 \right) (0.8 \text{ m/s}) (14,080 \text{ Pa}) = \mathbf{0.88 \text{ W}}$$



13-43 The average flow velocity in a pipe is given. The pressure drop and the pumping power are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be $\rho = 999.7 \text{ kg/m}^3$ and $\mu = 1.307 \times 10^{-3} \text{ kg/m} \cdot \text{s}$, respectively.

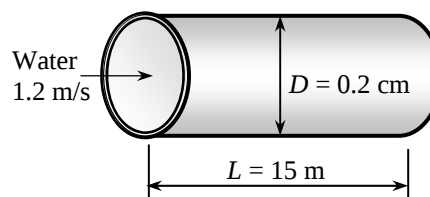
Analysis (a) First we need to determine the flow regime. The Reynolds number of the flow is

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{(999.7 \text{ kg/m}^3)(1.2 \text{ m/s})(2 \times 10^{-3} \text{ m})}{1.307 \times 10^{-3} \text{ kg/m} \cdot \text{s}} = 1836$$

which is less than 2300. Therefore, the flow is laminar. Then the friction factor and the pressure drop become

$$f = \frac{64}{\text{Re}} = \frac{64}{1836} = 0.0349$$

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.0349 \frac{15 \text{ m}}{0.002 \text{ m}} \frac{(999.7 \text{ kg/m}^3)(1.2 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) = \mathbf{188 \text{ kPa}}$$



(b) The volume flow rate and the pumping power requirements are

$$\dot{V} = V_{\text{avg}} A_c = V_{\text{avg}} (\pi D^2 / 4) = (1.2 \text{ m/s}) [\pi (0.002 \text{ m})^2 / 4] = 3.77 \times 10^{-6} \text{ m}^3 / \text{s}$$

$$\dot{W}_{\text{pump}} = \dot{V} \Delta P = (3.77 \times 10^{-6} \text{ m}^3 / \text{s})(188 \text{ kPa}) \left(\frac{1000 \text{ W}}{1 \text{ kPa} \cdot \text{m}^3 / \text{s}} \right) = \mathbf{0.71 \text{ W}}$$

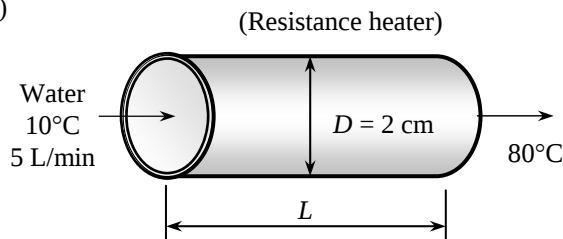
Therefore, power input in the amount of 0.71 W is needed to overcome the frictional losses in the flow due to viscosity.

13-44 Water is to be heated in a tube equipped with an electric resistance heater on its surface. The power rating of the heater and the inner surface temperature are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The surface heat flux is uniform. 3 The inner surfaces of the tube are smooth.

Properties The properties of water at the average temperature of $(80+10)/2 = 45^\circ\text{C}$ are (Table A-15)

$$\begin{aligned}\rho &= 990.1 \text{ kg/m}^3 \\ k &= 0.637 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= \mu / \rho = 0.602 \times 10^{-6} \text{ m}^2/\text{s} \\ c_p &= 4180 \text{ J/kg}\cdot^\circ\text{C} \\ \text{Pr} &= 3.91\end{aligned}$$



Analysis The power rating of the resistance heater is

$$\dot{m} = \rho \dot{V} = (990.1 \text{ kg/m}^3)(0.005 \text{ m}^3/\text{min}) = 4.951 \text{ kg/min} = 0.0825 \text{ kg/s}$$

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (0.0825 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(80 - 10)^\circ\text{C} = \mathbf{24,140 \text{ W}}$$

The velocity of water and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{(5 \times 10^{-3} / 60) \text{ m}^3/\text{s}}{\pi(0.02 \text{ m})^2 / 4} = 0.2653 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.2653 \text{ m/s})(0.02 \text{ m})}{0.602 \times 10^{-6} \text{ m}^2/\text{s}} = 8813$$

which is less than 10,000 but much greater than 2300. We assume the flow to be turbulent. The entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.02 \text{ m}) = 0.20 \text{ m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(8813)^{0.8} (3.91)^{0.4} = 56.85$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.637 \text{ W/m}\cdot^\circ\text{C}}{0.02 \text{ m}} (56.85) = 1811 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the inner surface temperature of the pipe at the exit becomes

$$\begin{aligned}\dot{Q} &= hA_s (T_{s,e} - T_e) \\ 24,140 \text{ W} &= (1811 \text{ W/m}^2\cdot^\circ\text{C})[\pi(0.02 \text{ m})(13 \text{ m})](T_{s,e} - 80)^\circ\text{C} \\ T_{s,e} &= \mathbf{96.3^\circ\text{C}}\end{aligned}$$

13-45 Flow of hot air through uninsulated square ducts of a heating system in the attic is considered. The exit temperature and the rate of heat loss are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The inner surfaces of the duct are smooth. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 80°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at 1 atm and this temperature are (Table A-22)

$$\rho = 0.9994 \text{ kg/m}^3$$

$$k = 0.02953 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 2.097 \times 10^{-5} \text{ m}^2/\text{s}$$

$$c_p = 1008 \text{ J/kg}\cdot^\circ\text{C}$$

$$\text{Pr} = 0.7154$$

Analysis The characteristic length that is the hydraulic diameter, the mean velocity of air, and the Reynolds number are

$$D_h = \frac{4A_c}{P} = \frac{4a^2}{4a} = a = 0.15 \text{ m}$$

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.10 \text{ m}^3/\text{s}}{(0.15 \text{ m})^2} = 4.444 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4.444 \text{ m/s})(0.15 \text{ m})}{2.097 \times 10^{-5} \text{ m}^2/\text{s}} = 31,791$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10(0.15 \text{ m}) = 1.5 \text{ m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(31,791)^{0.8} (0.7154)^{0.3} = 83.16$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02953 \text{ W/m}\cdot^\circ\text{C}}{0.15 \text{ m}} (83.16) = 16.37 \text{ W/m}^2\cdot^\circ\text{C}$$

Next we determine the exit temperature of air,

$$A_s = 4aL = 4(0.15 \text{ m})(10 \text{ m}) = 6 \text{ m}^2$$

$$\dot{m} = \rho \dot{V} = (0.9994 \text{ kg/m}^3)(0.10 \text{ m}^3/\text{s}) = 0.09994 \text{ kg/s}$$

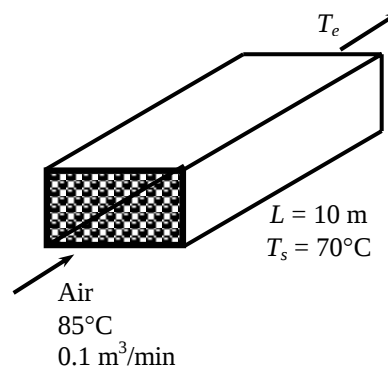
$$T_e = T_s - (T_s - T_i) e^{-hA_s/(\dot{m}c_p)} = 70 - (70 - 85) e^{-\frac{(16.37)(6)}{(0.09994)(1008)}} = 75.7^\circ\text{C}$$

Then the logarithmic mean temperature difference and the rate of heat loss from the air becomes

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{75.7 - 85}{\ln\left(\frac{70 - 75.7}{70 - 85}\right)} = 9.58^\circ\text{C}$$

$$\dot{Q} = hA_s \Delta T_{\text{ln}} = (16.37 \text{ W/m}^2\cdot^\circ\text{C})(6 \text{ m}^2)(9.58^\circ\text{C}) = 941 \text{ W}$$

Note that the temperature of air drops by almost 10°C as it flows in the duct as a result of heat loss.



13-46 EES Prob. 13-45 is reconsidered. The effect of the volume flow rate of air on the exit temperature of air and the rate of heat loss is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

T_i=85 [C]
L=10 [m]
side=0.15 [m]
V_{dot}=0.10 [m³/s]
T_s=70 [C]

"PROPERTIES"

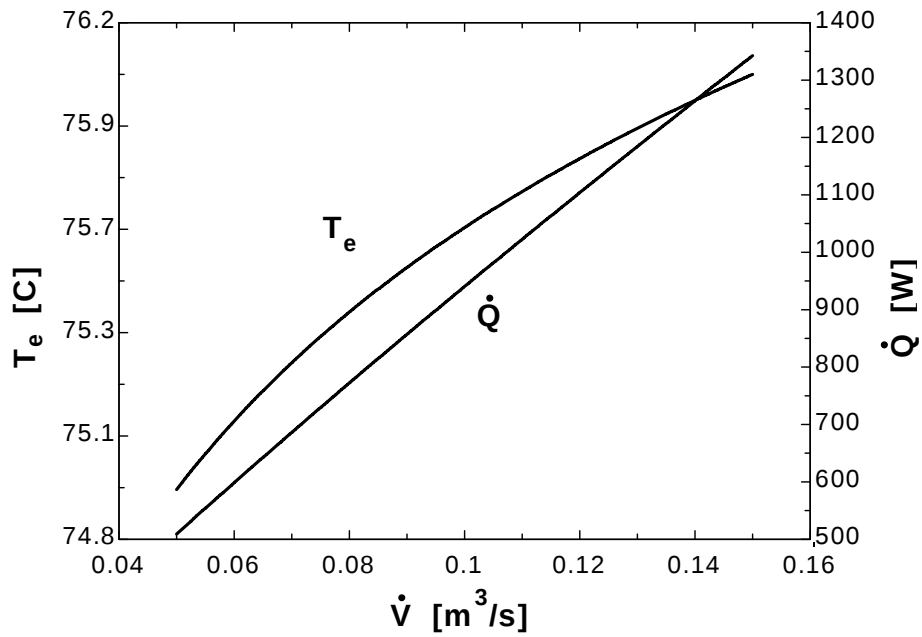
Fluid\$='air'
C_p=CP(Fluid\$, T=T_{ave})*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid\$, T=T_{ave})
Pr=Prandtl(Fluid\$, T=T_{ave})
rho=Density(Fluid\$, T=T_{ave}, P=101.3)
mu=Viscosity(Fluid\$, T=T_{ave})
nu=mu/rho
T_{ave}=1/2*(T_i+T_e)

"ANALYSIS"

D_h=(4*A_c)/p
A_c=side^2
p=4*side
Vel=V_{dot}/A_c
Re=(Vel*D_h)/nu "The flow is turbulent"
L_t=10*D_h "The entry length is much shorter than the total length of the duct."
Nusselt=0.023*Re^0.8*Pr^0.3
h=k/D_h*Nusselt
A=4*side*L
m_{dot}=rho*V_{dot}
T_e=T_s-(T_s-T_i)*exp((-h*A)/(m_{dot}*C_p))
DELTAT_{ln}=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
Q_{dot}=h*A*DELTAT_{ln}

V [m ³ /s]	T _e [C]	Q [W]
0.05	74.89	509
0.055	75	554.1
0.06	75.09	598.6
0.065	75.18	642.7
0.07	75.26	686.3
0.075	75.34	729.5
0.08	75.41	772.4
0.085	75.48	814.8
0.09	75.54	857
0.095	75.6	898.9
0.1	75.66	940.4
0.105	75.71	981.7
0.11	75.76	1023
0.115	75.81	1063
0.12	75.86	1104

0.125	75.9	1144
0.13	75.94	1184
0.135	75.98	1224
0.14	76.02	1264
0.145	76.06	1303
0.15	76.1	1343



13-47 Air enters the constant spacing between the glass cover and the plate of a solar collector. The net rate of heat transfer and the temperature rise of air are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The inner surfaces of the spacing are smooth. 3 Air is an ideal gas with constant properties. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and estimated average temperature of 35°C are (Table A-22)

$$\rho = 1.145 \text{ kg/m}^3, \quad k = 0.02625 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.655 \times 10^{-5} \text{ m}^2/\text{s}, \quad c_p = 1007 \text{ J/kg} \cdot ^\circ\text{C}, \quad \text{Pr} = 0.7268$$

Analysis Mass flow rate, cross sectional area, hydraulic diameter, mean velocity of air and the Reynolds number are

$$\dot{m} = \rho \dot{V} = (1.145 \text{ kg/m}^3)(0.15 \text{ m}^3/\text{s}) = 0.1718 \text{ kg/s}$$

$$A_c = (1 \text{ m})(0.03 \text{ m}) = 0.03 \text{ m}^2$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.03 \text{ m}^2)}{2(1 \text{ m} + 0.03 \text{ m})} = 0.05825 \text{ m}$$

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.15 \text{ m}^3/\text{s}}{0.03 \text{ m}^2} = 5 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(5 \text{ m/s})(0.05825 \text{ m})}{1.655 \times 10^{-5} \text{ m}^2/\text{s}} = 17,600$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10(0.05825 \text{ m}) = 0.5825 \text{ m}$$

which are much shorter than the total length of the collector. Therefore, we can assume fully developed turbulent flow in the entire collector, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(17,600)^{0.8} (0.7268)^{0.4} = 50.43$$

$$\text{and } h = \frac{k}{D_h} \text{Nu} = \frac{0.02625 \text{ W/m} \cdot ^\circ\text{C}}{0.05825 \text{ m}} (50.43) = 22.73 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The exit temperature of air can be calculated using the “average” surface temperature as

$$A_s = 2(5 \text{ m})(1 \text{ m}) = 10 \text{ m}^2, \quad T_{s,\text{avg}} = \frac{60 + 20}{2} = 40^\circ\text{C}$$

$$T_e = T_{s,\text{avg}} - (T_{s,\text{avg}} - T_i) \exp\left(-\frac{hA_s}{\dot{m}c_p}\right) = 40 - (40 - 30) \exp\left(-\frac{22.73 \times 10}{0.1718 \times 1007}\right) = 37.31^\circ\text{C}$$

The temperature rise of air is

$$\Delta T = 37.3^\circ\text{C} - 30^\circ\text{C} = \mathbf{7.3^\circ\text{C}}$$

The logarithmic mean temperature difference and the heat loss from the glass are

$$\Delta T_{\ln, \text{glass}} = \frac{T_e - T_i}{\ln \frac{T_s - T_e}{T_s - T_i}} = \frac{37.31 - 30}{\ln \frac{20 - 37.31}{20 - 30}} = 13.32^\circ\text{C}$$

$$\dot{Q}_{\text{glass}} = hA_s \Delta T_{\ln} = (22.73 \text{ W/m}^2 \cdot ^\circ\text{C})(5 \text{ m}^2)(13.32^\circ\text{C}) = 1514 \text{ W}$$

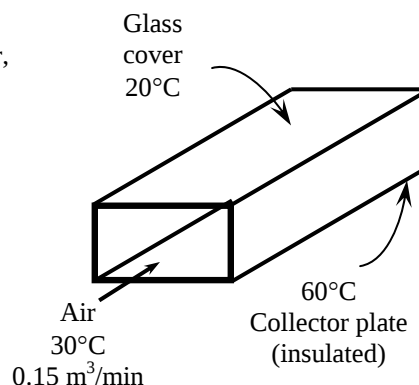
The logarithmic mean temperature difference and the heat gain of the absorber are

$$\Delta T_{\ln, \text{absorber}} = \frac{T_e - T_i}{\ln \frac{T_s - T_e}{T_s - T_i}} = \frac{37.31 - 30}{\ln \frac{60 - 37.31}{60 - 30}} = 26.17^\circ\text{C}$$

$$\dot{Q}_{\text{absorber}} = hA \Delta T_{\ln} = (22.73 \text{ W/m}^2 \cdot ^\circ\text{C})(5 \text{ m}^2)(26.17^\circ\text{C}) = 2975 \text{ W}$$

Then the net rate of heat transfer becomes

$$\dot{Q}_{\text{net}} = 2975 - 1514 = \mathbf{1461 \text{ W}}$$

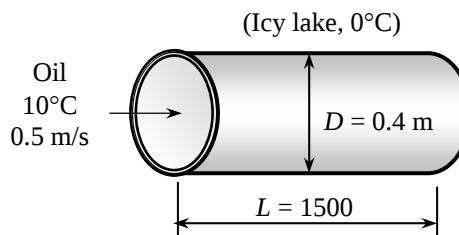


13-48 Oil flows through a pipeline that passes through icy waters of a lake. The exit temperature of the oil and the rate of heat loss are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature of the pipe is very nearly 0°C. 3 The thermal resistance of the pipe is negligible. 4 The inner surfaces of the pipeline are smooth. 5 The flow is hydrodynamically developed when the pipeline reaches the lake.

Properties The properties of oil at 10°C are (Table A-19)

$$\begin{aligned}\rho &= 893.6 \text{ kg/m}^3, & k &= 0.1460 \text{ W/m} \cdot ^\circ\text{C} \\ \mu &= 2.326 \text{ kg/m} \cdot \text{s}, & \nu &= 2.592 \times 10^{-3} \text{ m}^2/\text{s} \\ c_p &= 1839 \text{ J/kg} \cdot ^\circ\text{C}, & \text{Pr} &= 28,750\end{aligned}$$



Analysis (a) The Reynolds number in this case is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.5 \text{ m/s})(0.4 \text{ m})}{2.592 \times 10^{-3} \text{ m}^2/\text{s}} = 77.16$$

which is less than 2300. Therefore, the flow is laminar, and the thermal entry length is roughly

$$L_t = 0.05 \text{ Re Pr } D = 0.05(77.16)(28,750)(0.4 \text{ m}) = 44,367 \text{ m}$$

which is much longer than the total length of the pipe. Therefore, we assume thermally developing flow, and determine the Nusselt number from

$$\text{Nu} = \frac{hD}{k} = 3.66 + \frac{0.065(D/L) \text{ Re Pr}}{1 + 0.04[(D/L) \text{ Re Pr}]^{2/3}} = 3.66 + \frac{0.065\left(\frac{0.4 \text{ m}}{1500 \text{ m}}\right)(77.16)(28,750)}{1 + 0.04\left[\left(\frac{0.4 \text{ m}}{1500 \text{ m}}\right)(77.16)(28,750)\right]^{2/3}} = 13.73$$

$$\text{and } h = \frac{k}{D} \text{Nu} = \frac{0.1460 \text{ W/m} \cdot ^\circ\text{C}}{0.4 \text{ m}} (13.73) = 5.011 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Next we determine the exit temperature of oil

$$A_s = \pi DL = \pi(0.4 \text{ m})(1500 \text{ m}) = 1885 \text{ m}^2$$

$$\dot{m} = \rho \dot{V} = \rho A_c V_{\text{avg}} = \rho \left(\frac{\pi D^2}{4} \right) V_{\text{avg}} = (893.6 \text{ kg/m}^3) \frac{\pi(0.4 \text{ m})^2}{4} (0.5 \text{ m/s}) = 56.15 \text{ kg/s}$$

$$T_e = T_s - (T_s - T_i) e^{-hA_s / (\dot{m} c_p)} = 0 - (0 - 10) e^{-\frac{(5.011)(1885)}{(56.15)(1839)}} = 9.13^\circ\text{C}$$

(b) The logarithmic mean temperature difference and the rate of heat loss from the oil are

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{9.13 - 10}{\ln\left(\frac{0 - 9.13}{0 - 10}\right)} = 9.56^\circ\text{C}$$

$$\dot{Q} = hA_s \Delta T_{\text{ln}} = (5.011 \text{ W/m}^2 \cdot ^\circ\text{C})(1885 \text{ m}^2)(9.56^\circ\text{C}) = 90,300 \text{ W} = 90.3 \text{ kW}$$

The friction factor is

$$f = \frac{64}{\text{Re}} = \frac{64}{77.16} = 0.8294$$

Then the pressure drop in the pipe and the required pumping power become

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.8294 \frac{1500 \text{ m}}{0.4 \text{ m}} \frac{(893.6 \text{ kg/m}^3)(0.5 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) = 347.4 \text{ kPa}$$

$$\dot{W}_{\text{pump,u}} = \dot{V} \Delta P = A_c V_{\text{avg}} \Delta P = \frac{\pi(0.4 \text{ m})^2}{4} (0.5 \text{ m/s})(347.4 \text{ kPa}) \left(\frac{1 \text{ kW}}{1 \text{ kPa} \cdot \text{m}^3/\text{s}} \right) = 21.8 \text{ kW}$$

Discussion The power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be much more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

13-49 Laminar flow of a fluid through an isothermal square channel is considered. The change in the pressure drop and the rate of heat transfer are to be determined when the mean velocity is doubled.

Assumptions 1 The flow is fully developed. 2 The effect of the change in ΔT_{in} on the rate of heat transfer is not considered.

Analysis The pressure drop of the fluid for laminar flow is expressed as

$$\Delta P_1 = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = \frac{64}{\text{Re}} \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = \frac{64\nu}{V_{\text{avg}} D} \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 32 V_{\text{avg}} \frac{\nu L \rho}{D^2}$$

When the free-stream velocity of the fluid is doubled, the pressure drop becomes

$$\Delta P_2 = f \frac{L}{D} \frac{\rho (2V_{\text{avg}})^2}{2} = \frac{64}{\text{Re}} \frac{L}{D} \frac{\rho 4V_{\text{avg}}^2}{2} = \frac{64\nu}{2V_{\text{avg}} D} \frac{L}{D} \frac{\rho 4V_{\text{avg}}^2}{2} = 64 V_{\text{avg}} \frac{\nu L \rho}{D^2}$$

Their ratio is

$$\frac{\Delta P_2}{\Delta P_1} = \frac{64}{32} = \mathbf{2}$$

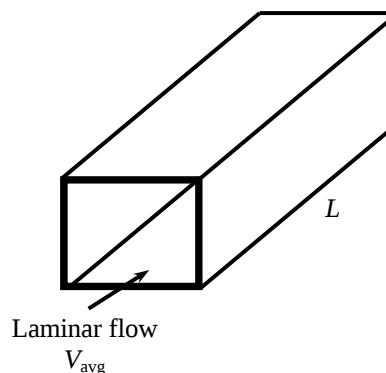
The rate of heat transfer between the fluid and the walls of the channel is expressed as

$$\dot{Q}_1 = h A_s \Delta T_{\text{in}} = \frac{k}{D} \text{Nu} A_s \Delta T_{\text{in}} = \frac{k}{D} 2.98 A_s \Delta T_{\text{in}}$$

When the effect of the change in ΔT_{in} on the rate of heat transfer is disregarded, the rate of heat transfer remains the same. Therefore,

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \mathbf{1}$$

Therefore, doubling the velocity will double the pressure drop but it will not affect the heat transfer rate.



13-50 Turbulent flow of a fluid through an isothermal square channel is considered. The change in the pressure drop and the rate of heat transfer are to be determined when the free-stream velocity is doubled.

Assumptions 1 The flow is fully developed. 2 The effect of the change in ΔT_{ln} on the rate of heat transfer is not considered.

Analysis The pressure drop of the fluid for turbulent flow is expressed as

$$\Delta P_1 = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.184 \text{Re}^{-0.2} \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.184 \frac{V_{\text{avg}}^{-0.2} D^{-0.2}}{\nu^{-0.2}} \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.092 V_{\text{avg}}^{1.8} \left(\frac{D}{\nu} \right)^{-0.2} \frac{L \rho}{D}$$

When the free-stream velocity of the fluid is doubled, the pressure drop becomes

$$\begin{aligned} \Delta P_2 &= f \frac{L}{D} \frac{\rho (2V_{\text{avg}})^2}{2} = 0.184 \text{Re}^{-0.2} \frac{L}{D} \frac{\rho 4V_{\text{avg}}^2}{2} = 0.184 \frac{(2V_{\text{avg}})^{-0.2} D^{-0.2}}{\nu^{-0.2}} \frac{L}{D} \frac{\rho 4V_{\text{avg}}^2}{2} \\ &= 0.368(2)^{-0.2} V_{\text{avg}}^{1.8} \left(\frac{D}{\nu} \right)^{-0.2} \frac{L \rho}{D} \end{aligned}$$

Their ratio is

$$\frac{\Delta P_2}{\Delta P_1} = \frac{0.368(2)^{-0.2} V_{\text{avg}}^{1.8}}{0.092 V_{\text{avg}}^{1.8}} = 4(2)^{-0.2} = \mathbf{3.48}$$

The rate of heat transfer between the fluid and the walls of the channel is expressed as

$$\begin{aligned} \dot{Q}_1 &= hA\Delta T_{\text{ln}} = \frac{k}{D} \text{Nu} A \Delta T_{\text{ln}} = \frac{k}{D} 0.023 \text{Re}^{0.8} \text{Pr}^{1/3} A \Delta T_{\text{ln}} \\ &= 0.023 V_{\text{avg}}^{0.8} \left(\frac{D}{\nu} \right)^{0.8} \frac{k}{D} \text{Pr}^{1/3} A \Delta T_{\text{ln}} \end{aligned}$$

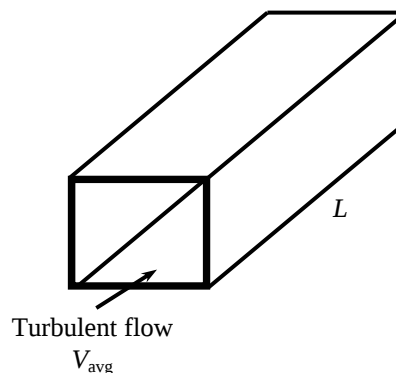
When the free-stream velocity of the fluid is doubled, the heat transfer rate becomes

$$\dot{Q}_2 = 0.023(2V_{\text{avg}})^{0.8} \left(\frac{D}{\nu} \right)^{0.8} \frac{k}{D} \text{Pr}^{1/3} A \Delta T_{\text{ln}}$$

Their ratio is

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2V_{\text{avg}})^{0.8}}{V_{\text{avg}}^{0.8}} = 2^{0.8} = \mathbf{1.74}$$

Therefore, doubling the velocity will increase the pressure drop 3.8 times but it will increase the heat transfer rate by only 74%.



13-51E Water is heated in a parabolic solar collector. The required length of parabolic collector and the surface temperature of the collector tube are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The thermal resistance of the tube is negligible. 3 The inner surfaces of the tube are smooth.

Properties The properties of water at the average temperature of $(55+200)/2 = 127.5^\circ\text{F}$ are (Table A-15E)

$$\begin{aligned}\rho &= 61.59 \text{ lbm/ft}^3 \\ k &= 0.374 \text{ Btu/ft}\cdot^\circ\text{F} \\ \nu &= \mu / \rho = 0.5681 \times 10^{-5} \text{ ft}^2/\text{s} \\ c_p &= 0.999 \text{ Btu/lbm}\cdot^\circ\text{F} \\ \text{Pr} &= 3.368\end{aligned}$$

Analysis The total rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (4 \text{ lbm/s})(0.999 \text{ Btu/lbm}\cdot^\circ\text{F})(200 - 55)^\circ\text{F} = 579.4 \text{ Btu/s} = 2.086 \times 10^6 \text{ Btu/h}$$

The length of the tube required is

$$L = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{2.086 \times 10^6 \text{ Btu/h}}{350 \text{ Btu/h}\cdot\text{ft}} = \mathbf{5960 \text{ ft}}$$

The velocity of water and the Reynolds number are

$$V_{avg} = \frac{\dot{m}}{\rho A_c} = \frac{4 \text{ lbm/s}}{(61.59 \text{ lbm/ft}^3)\pi \frac{(1.25/12 \text{ ft})^2}{4}} = 7.621 \text{ ft/s}$$

$$\text{Re} = \frac{V_{avg} D_h}{\nu} = \frac{(7.621 \text{ m/s})(1.25/12 \text{ ft})}{0.5681 \times 10^{-5} \text{ ft}^2/\text{s}} = 1.397 \times 10^5$$

which is greater than 10,000. Therefore, we can assume fully developed turbulent flow in the entire tube, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} = 0.023(1.397 \times 10^5)^{0.8} (3.368)^{0.4} = 488.5$$

The heat transfer coefficient is

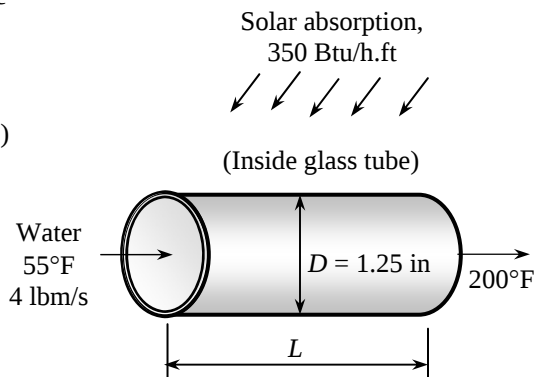
$$h = \frac{k}{D_h} \text{Nu} = \frac{0.374 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{1.25/12 \text{ ft}} (488.5) = 1754 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

The heat flux on the tube is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{2.086 \times 10^6 \text{ Btu/h}}{\pi(1.25/12 \text{ ft})(5960 \text{ ft})} = 1070 \text{ Btu/h}\cdot\text{ft}^2$$

Then the surface temperature of the tube at the exit becomes

$$\dot{q} = h(T_s - T_e) \longrightarrow T_s = T_e + \frac{\dot{q}}{h} = 200^\circ\text{F} + \frac{1070 \text{ Btu/h}\cdot\text{ft}^2}{1754 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}} = \mathbf{200.6^\circ\text{F}}$$



13-52 A circuit board is cooled by passing cool air through a channel drilled into the board. The maximum total power of the electronic components is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat flux at the top surface of the channel is uniform, and heat transfer through other surfaces is negligible. 3 The inner surfaces of the channel are smooth. 4 Air is an ideal gas with constant properties. 5 The pressure of air in the channel is 1 atm.

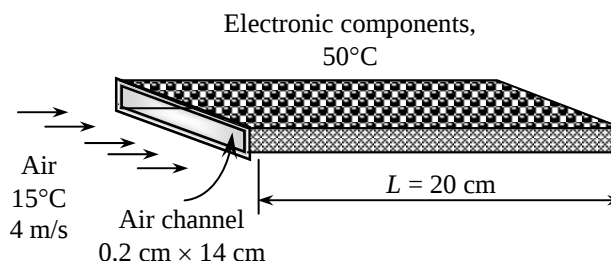
Properties The properties of air at 1 atm and estimated average temperature of 25°C are (Table A-22)

$$\begin{aligned}\rho &= 1.184 \text{ kg/m}^3 \\ k &= 0.02551 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.562 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7296\end{aligned}$$

Analysis The cross-sectional and heat transfer surface areas are

$$A_c = (0.002 \text{ m})(0.14 \text{ m}) = 0.00028 \text{ m}^2$$

$$A_s = (0.14 \text{ m})(0.2 \text{ m}) = 0.028 \text{ m}^2$$



To determine heat transfer coefficient, we first need to find the Reynolds number,

$$D_h = \frac{4A_c}{P} = \frac{4(0.00028 \text{ m}^2)}{2(0.002 \text{ m} + 0.14 \text{ m})} = 0.003944 \text{ m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4 \text{ m/s})(0.003944 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 1010$$

which is less than 2300. Therefore, the flow is laminar and the thermal entry length is

$$L_t = 0.05 \text{ Re Pr } D_h = 0.05(1010)(0.7296)(0.003944 \text{ m}) = 0.1453 \text{ m} < 0.20 \text{ m}$$

Therefore, we have developing flow through most of the channel. However, we take the conservative approach and assume fully developed flow, and from Table 13-1 we read $\text{Nu} = 8.24$. Then the heat transfer coefficient becomes

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02551 \text{ W/m} \cdot ^\circ\text{C}}{0.003944 \text{ m}} (8.24) = 53.30 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Also,

$$\dot{m} = \rho V A_c = (1.184 \text{ kg/m}^3)(4 \text{ m/s})(0.00028 \text{ m}^2) = 0.001326 \text{ kg/s}$$

Heat flux at the exit can be written as $\dot{q} = h(T_s - T_e)$ where $T_s = 50^\circ\text{C}$ at the exit. Then the heat transfer rate can be expressed as $\dot{Q} = \dot{q} A_s = h A_s (T_s - T_e)$, and the exit temperature of the air can be determined from

$$\begin{aligned}h A_s (T_s - T_e) &= \dot{m} c_p (T_e - T_i) \\ (53.30 \text{ W/m}^2 \cdot ^\circ\text{C})(0.028 \text{ m}^2)(50^\circ\text{C} - T_e) &= (0.001326 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})(T_e - 15^\circ\text{C}) \\ T_e &= 33.5^\circ\text{C}\end{aligned}$$

Then the maximum total power of the electronic components that can safely be mounted on this circuit board becomes

$$\dot{Q}_{\text{max}} = \dot{m} c_p (T_e - T_i) = (0.001326 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})(33.5 - 15^\circ\text{C}) = \mathbf{24.7 \text{ W}}$$

13-53 A circuit board is cooled by passing cool helium gas through a channel drilled into the board. The maximum total power of the electronic components is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat flux at the top surface of the channel is uniform, and heat transfer through other surfaces is negligible. 3 The inner surfaces of the channel are smooth. 4 Helium is an ideal gas. 5 The pressure of helium in the channel is 1 atm.

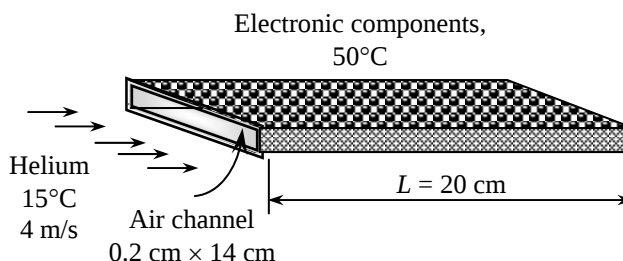
Properties The properties of helium at the estimated average temperature of 25°C are (from EES)

$$\begin{aligned}\rho &= 0.1635 \text{ kg/m}^3 \\ k &= 0.1553 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= 1.214 \times 10^{-4} \text{ m}^2/\text{s} \\ c_p &= 5193 \text{ J/kg}\cdot^\circ\text{C} \\ \text{Pr} &= 0.6636\end{aligned}$$

Analysis The cross-sectional and heat transfer surface areas are

$$A_c = (0.002 \text{ m})(0.14 \text{ m}) = 0.00028 \text{ m}^2$$

$$A_s = (0.14 \text{ m})(0.2 \text{ m}) = 0.028 \text{ m}^2$$



To determine heat transfer coefficient, we need to first find the Reynolds number

$$D_h = \frac{4A_c}{p} = \frac{4(0.00028 \text{ m}^2)}{2(0.002 \text{ m} + 0.14 \text{ m})} = 0.003944 \text{ m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4 \text{ m/s})(0.003944 \text{ m})}{1.214 \times 10^{-4} \text{ m}^2/\text{s}} = 130.0$$

which is less than 2300. Therefore, the flow is laminar and the thermal entry length is

$$L_t = 0.05 \text{ Re Pr } D_h = 0.05(130.0)(0.6636)(0.003944 \text{ m}) = 0.0170 \text{ m} \ll 0.20 \text{ m}$$

Therefore, the flow is fully developed flow, and from Table 13-3 we read $\text{Nu} = 8.24$. Then the heat transfer coefficient becomes

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.1553 \text{ W/m}\cdot^\circ\text{C}}{0.003944 \text{ m}} (8.24) = 324.5 \text{ W/m}^2\cdot^\circ\text{C}$$

Also,

$$\dot{m} = \rho V A_c = (0.1635 \text{ kg/m}^3)(4 \text{ m/s})(0.00028 \text{ m}^2) = 0.0001831 \text{ kg/s}$$

Heat flux at the exit can be written as $\dot{q} = h(T_s - T_e)$ where $T_s = 50^\circ\text{C}$ at the exit. Then the heat transfer rate can be expressed as $\dot{Q} = \dot{q} A_s = h A_s (T_s - T_e)$, and the exit temperature of the air can be determined from

$$\dot{m} c_p (T_e - T_i) = h A_s (T_s - T_e)$$

$$(0.0001831 \text{ kg/s})(5193 \text{ J/kg}\cdot^\circ\text{C})(T_e - 15^\circ\text{C}) = (324.5 \text{ W/m}^2\cdot^\circ\text{C})(0.028 \text{ m}^2)(50^\circ\text{C} - T_e)$$

$$T_e = 46.68^\circ\text{C}$$

Then the maximum total power of the electronic components that can safely be mounted on this circuit board becomes

$$\dot{Q}_{\text{max}} = \dot{m} c_p (T_e - T_i) = (0.0001831 \text{ kg/s})(5193 \text{ J/kg}\cdot^\circ\text{C})(46.68 - 15^\circ\text{C}) = \mathbf{30.1 \text{ W}}$$

13-54 EES Prob. 13-52 is reconsidered. The effects of air velocity at the inlet of the channel and the maximum surface temperature on the maximum total power dissipation of electronic components are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

L=0.20 [m]
width=0.14 [m]
height=0.002 [m]
T_i=15 [C]
Vel=4 [m/s]
T_s=50 [C]

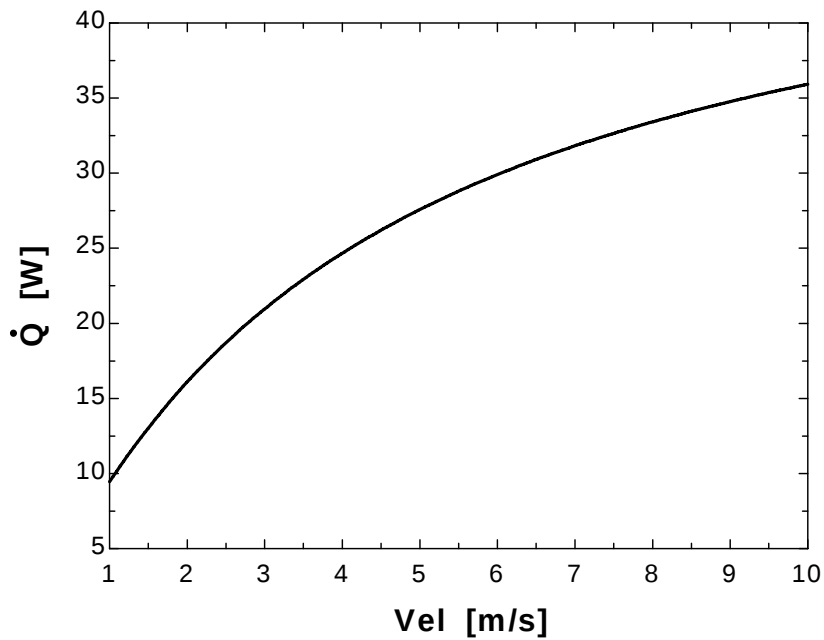
"PROPERTIES"

Fluid\$='air'
c_p=CP(Fluid\$, T=T_{ave})*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid\$, T=T_{ave})
Pr=Prandtl(Fluid\$, T=T_{ave})
rho=Density(Fluid\$, T=T_{ave}, P=101.3)
mu=Viscosity(Fluid\$, T=T_{ave})
nu=mu/rho
T_{ave}=1/2*(T_i+T_e)

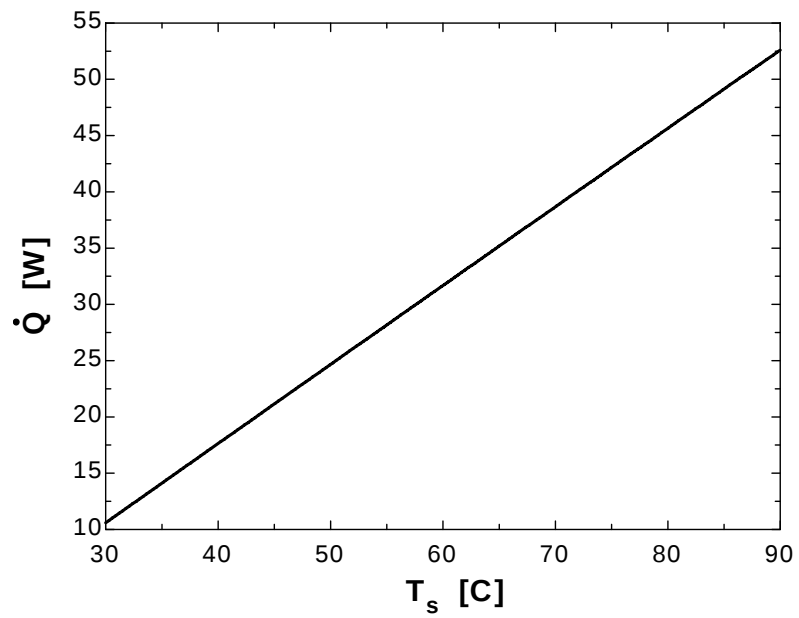
"ANALYSIS"

A_c=width*height
A=width*L
p=2*(width+height)
D_h=(4*A_c)/p
Re=(Vel*D_h)/nu "The flow is laminar"
L_t=0.05*Re*Pr*D_h
"Taking conservative approach and assuming fully developed laminar flow, from Table 13-1 we read"
Nusselt=8.24
h=k/D_h*Nusselt
m_{dot}=rho*Vel*A_c
Q_{dot}=h*A*(T_s-T_e)
Q_{dot}=m_{dot}*c_p*(T_e-T_i)

Vel [m/s]	Q [W]
1	9.453
2	16.09
3	20.96
4	24.67
5	27.57
6	29.91
7	31.82
8	33.41
9	34.76
10	35.92



T_s [C]	Q [W]
30	10.59
35	14.12
40	17.64
45	21.15
50	24.67
55	28.18
60	31.68
65	35.18
70	38.68
75	42.17
80	45.65
85	49.13
90	52.6



13-55 Air enters a rectangular duct. The exit temperature of the air, the rate of heat transfer, and the fan power are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The inner surfaces of the duct are smooth. 3 Air is an ideal gas with constant properties. 4 The pressure of air in the duct is 1 atm.

Properties We assume the bulk mean temperature for air to be 40°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at this temperature and 1 atm are (Table A-22)

$$\begin{aligned}\rho &= 1.127 \text{ kg/m}^3 & c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ k &= 0.02662 \text{ W/m} \cdot ^\circ\text{C} & \text{Pr} &= 0.7255 \\ \nu &= 1.702 \times 10^{-5} \text{ m}^2/\text{s}\end{aligned}$$

Analysis (a) The hydraulic diameter, the mean velocity of air, and the Reynolds number are

$$D_h = \frac{4A_c}{p} = \frac{4(0.15 \text{ m})(0.20 \text{ m})}{2[(0.15 \text{ m}) + (0.20 \text{ m})]} = 0.1714 \text{ m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(7 \text{ m/s})(0.1714 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 70,500$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10(0.1714 \text{ m}) = 1.714 \text{ m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(70,500)^{0.8} (0.7255)^{0.3} = 157.9$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02662 \text{ W/m} \cdot ^\circ\text{C}}{0.1714 \text{ m}} (157.9) = 24.52 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Next we determine the exit temperature of air

$$A_s = 2 \times 7[(0.15 \text{ m}) + (0.20 \text{ m})] = 4.9 \text{ m}^2$$

$$A_c = (0.15 \text{ m})(0.20 \text{ m}) = 0.03 \text{ m}^2$$

$$\dot{m} = \rho V A_c = (1.127 \text{ kg/m}^3)(7 \text{ m/s})(0.03 \text{ m}^2) = 0.2367 \text{ kg/s}$$

$$T_e = T_s - (T_s - T_i) e^{-hA_s / (\dot{m} c_p)} = 10 - (10 - 50) e^{-\frac{(24.52)(4.9)}{(0.2367)(1007)}} = \mathbf{34.16^\circ\text{C}}$$

(b) The logarithmic mean temperature difference and the rate of heat loss from the air are

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{34.16 - 50}{\ln\left(\frac{10 - 34.16}{10 - 50}\right)} = 31.42^\circ\text{C}$$

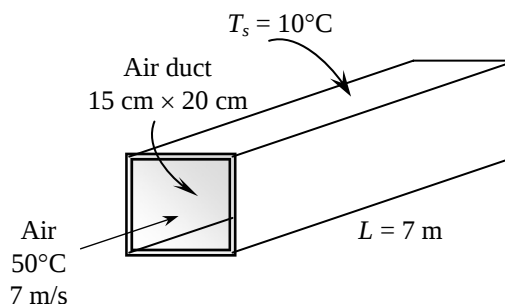
$$\dot{Q} = h A_s \Delta T_{\text{ln}} = (24.52 \text{ W/m}^2 \cdot ^\circ\text{C})(4.9 \text{ m}^2)(31.42^\circ\text{C}) = \mathbf{3775 \text{ W}}$$

(c) The friction factor, the pressure drop, and then the fan power can be determined for the case of fully developed turbulent flow to be

$$f = 0.184 \text{Re}^{-0.2} = 0.184(70,500)^{-0.2} = 0.01973$$

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.01973 \frac{(7 \text{ m})}{(0.1714 \text{ m})} \frac{(1.127 \text{ kg/m}^3)(7 \text{ m/s})^2}{2} = 22.25 \text{ N/m}^2$$

$$\dot{W}_{\text{pump}} = \frac{\dot{m} \Delta P}{\rho} = \frac{(0.2367 \text{ kg/s})(22.25 \text{ N/m}^2)}{1.127 \text{ kg/m}^3} = \mathbf{4.67 \text{ W}}$$



13-56 EES Prob. 13-55 is reconsidered. The effect of air velocity on the exit temperature of air, the rate of heat transfer, and the fan power is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

L=7 [m]
height=0.15 [m]
width=0.20 [m]
T_i=50 [C]
Vel=7 [m/s]
T_s=10 [C]

"PROPERTIES"

Fluid\$='air'
c_p=CP(Fluid\$, T=T_{ave})*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid\$, T=T_{ave})
Pr=Prandtl(Fluid\$, T=T_{ave})
rho=Density(Fluid\$, T=T_{ave}, P=101.3)
mu=Viscosity(Fluid\$, T=T_{ave})
nu=mu/rho
T_{ave}=1/2*(T_i+T_e)

"ANALYSIS"

"(a)"

A_c=width*height
p=2*(width+height)
D_h=(4*A_c)/p
Re=(Vel*D_h)/nu **"The flow is turbulent"**
L_t=10*D_h **"The entry length is much shorter than the total length of the duct."**
Nusselt=0.023*Re^{0.8}*Pr^{0.3}
h=k/D_h*Nusselt
A=2*L*(width+height)
m_{dot}=rho*Vel*A_c
T_e=T_s-(T_s-T_i)*exp((-h*A)/(m_{dot}*c_p))

"(b)"

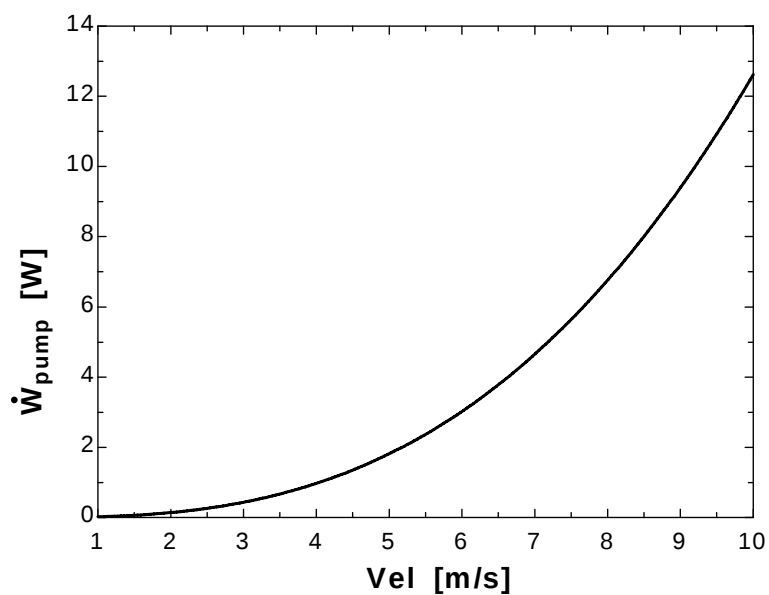
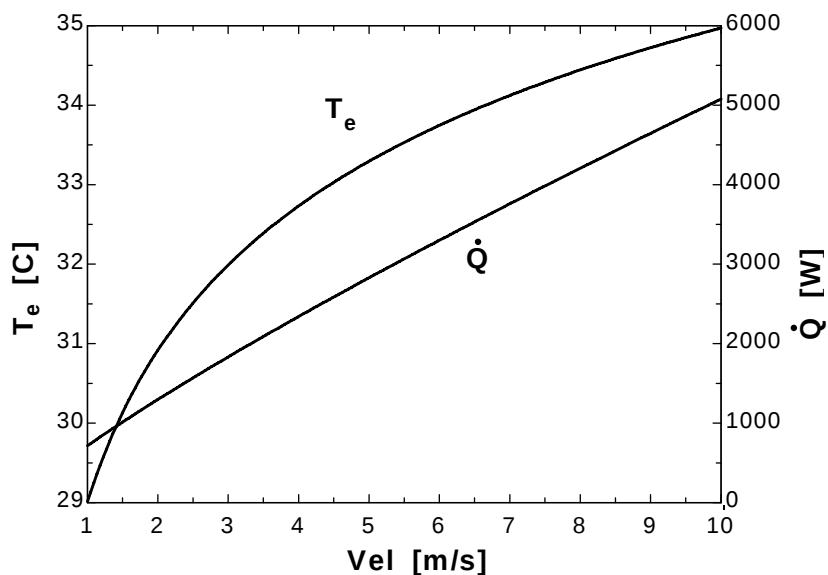
DELTA T_{ln}=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
Q_{dot}=h*A*DELTA T_{ln}

"(c)"

f=0.184*Re^(-0.2)
DELTA P=f*L/D_h*(rho*Vel²)/2
W_{dot pump}=(m_{dot}*DELTA P)/rho

Vel [m/s]	T _e [C]	Q [W]	W _{pump} [W]
1	29.01	715.6	0.02012
1.5	30.14	1014	0.06255
2	30.92	1297	0.1399
2.5	31.51	1570	0.2611
3	31.99	1833	0.4348
3.5	32.39	2090	0.6692
4	32.73	2341	0.9722
4.5	33.03	2587	1.352
5	33.29	2829	1.815
5.5	33.53	3066	2.369

6	33.75	3300	3.022
6.5	33.94	3531	3.781
7	34.12	3759	4.652
7.5	34.29	3984	5.642
8	34.44	4207	6.759
8.5	34.59	4427	8.008
9	34.72	4646	9.397
9.5	34.85	4862	10.93
10	34.97	5076	12.62



13-57 Hot air enters a sheet metal duct located in a basement. The exit temperature of hot air and the rate of heat loss are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the duct are smooth. 3 The thermal resistance of the duct is negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties We expect the air temperature to drop somewhat, and evaluate the air properties at 1 atm and the estimated bulk mean temperature of 50°C (Table A-22),

$$\rho = 1.092 \text{ kg/m}^3; \quad k = 0.02735 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}; \quad c_p = 1007 \text{ J/kg} \cdot ^\circ\text{C}$$

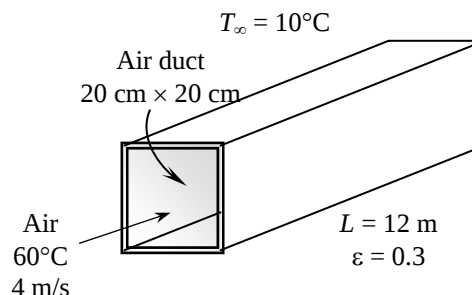
$$\text{Pr} = 0.7228$$

Analysis The surface area and the Reynolds number are

$$A_s = 4aL = 4 \times (0.2 \text{ m})(12 \text{ m}) = 9.6 \text{ m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{4a^2}{4a} = a = 0.2 \text{ m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4 \text{ m/s})(0.20 \text{ m})}{1.798 \times 10^{-5} \text{ m}^2/\text{s}} = 44,494$$



which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10(0.2 \text{ m}) = 2.0 \text{ m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow for the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(44,494)^{0.8} (0.7228)^{0.3} = 109.2$$

and

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02735 \text{ W/m} \cdot ^\circ\text{C}}{0.2 \text{ m}} (109.2) = 14.93 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The mass flow rate of air is

$$\dot{m} = \rho A_c V = (1.092 \text{ kg/m}^3)(0.2 \times 0.2 \text{ m}^2)(4 \text{ m/s}) = 0.1747 \text{ kg/s}$$

In steady operation, heat transfer from hot air to the duct must be equal to the heat transfer from the duct to the surrounding (by convection and radiation), which must be equal to the energy loss of the hot air in the duct. That is,

$$\dot{Q} = \dot{Q}_{\text{conv}, \text{in}} = \dot{Q}_{\text{conv} + \text{rad}, \text{out}} = \Delta \dot{E}_{\text{hot air}}$$

Assuming the duct to be at an average temperature of T_s , the quantities above can be expressed as

$$\dot{Q}_{\text{conv}, \text{in}}: \quad \dot{Q} = h_i A_s \Delta T_{\text{ln}} = h_i A_s \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \rightarrow \dot{Q} = (14.93 \text{ W/m}^2 \cdot ^\circ\text{C})(9.6 \text{ m}^2) \frac{T_e - 60}{\ln\left(\frac{T_s - T_e}{T_s - 60}\right)}$$

$$\dot{Q}_{\text{conv} + \text{rad}, \text{out}}: \quad \dot{Q} = h_o A_s (T_s - T_o) + \varepsilon A_s \sigma (T_s^4 - T_o^4) \rightarrow \dot{Q} = (10 \text{ W/m}^2 \cdot ^\circ\text{C})(9.6 \text{ m}^2)(T_s - 10)^\circ\text{C} + 0.3(9.6 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(T_s + 273)^4 - (10 + 273)^4]$$

$$\Delta \dot{E}_{\text{hot air}}: \quad \dot{Q} = \dot{m} c_p (T_e - T_i) \rightarrow \dot{Q} = (0.1747 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})(60 - T_e)^\circ\text{C}$$

This is a system of three equations with three unknowns whose solution is

$$\dot{Q} = 2622 \text{ W}, T_e = 45.1^\circ\text{C}, \text{ and } T_s = 33.3^\circ\text{C}$$

Therefore, the hot air will lose heat at a rate of 2622 W and exit the duct at 45.1°C.

13-58 EES Prob. 13-57 is reconsidered. The effects of air velocity and the surface emissivity on the exit temperature of air and the rate of heat loss are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

T_i=60 [C]
 L=12 [m]
 side=0.20 [m]
 Vel=4 [m/s]
 epsilon=0.3
 T_o=10 [C]
 h_o=10 [W/m²-C]
 T_{surr}=10 [C]

"PROPERTIES"

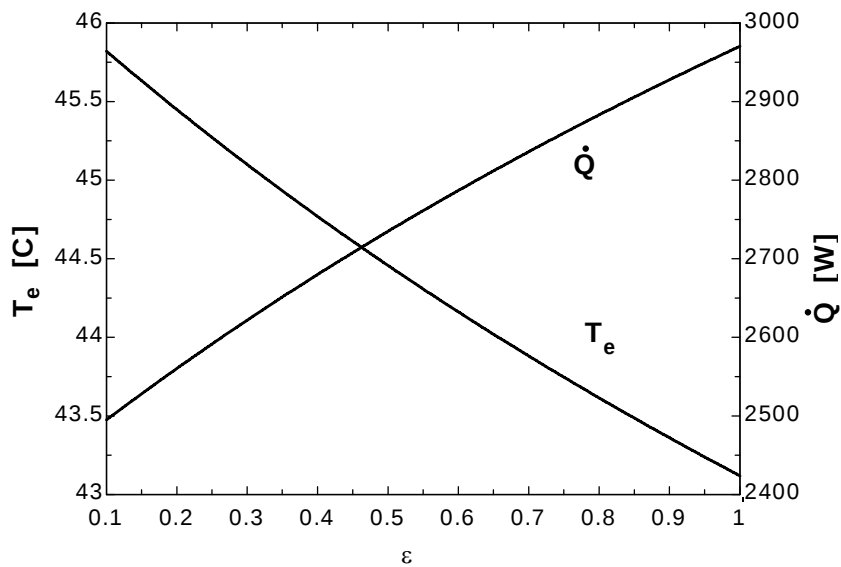
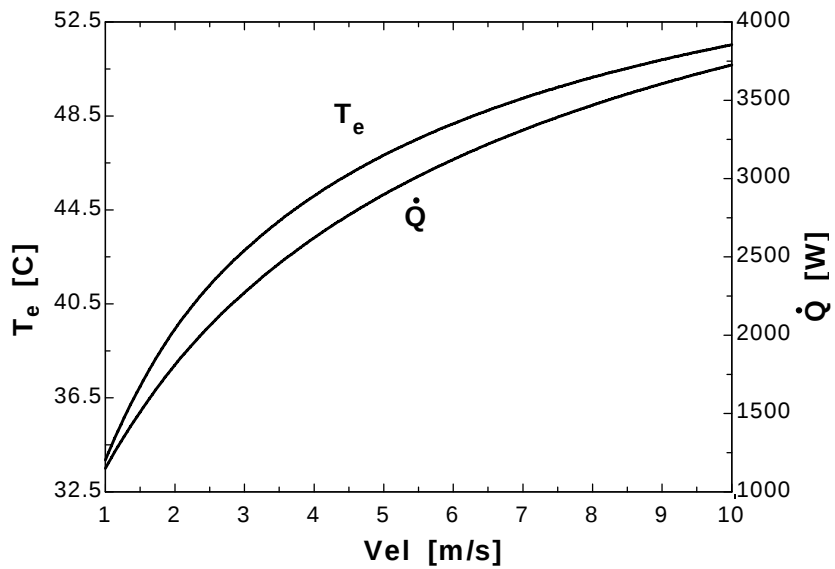
Fluid\$='air'
 c_p=CP(Fluid\$, T=T_{ave})*Convert(kJ/kg-C, J/kg-C)
 k=Conductivity(Fluid\$, T=T_{ave})
 Pr=Prandtl(Fluid\$, T=T_{ave})
 rho=Density(Fluid\$, T=T_{ave}, P=101.3)
 mu=Viscosity(Fluid\$, T=T_{ave})
 nu=mu/rho
 T_{ave}=T_i-10 "assumed average bulk mean temperature"

"ANALYSIS"

A=4*side*L
 A_c=side^2
 p=4*side
 D_h=(4*A_c)/p
 Re=(Vel*D_h)/nu "The flow is turbulent"
 L_t=10*D_h "The entry length is much shorter than the total length of the duct."
 Nusselt=0.023*Re^{0.8}*Pr^{0.3}
 h_i=k/D_h*Nusselt
 m_{dot}=rho*Vel*A_c
 Q_{dot}=Q_{dot_conv_in}
 Q_{dot_conv_in}=Q_{dot_conv_out}+Q_{dot_rad_out}
 Q_{dot_conv_in}=h_i*A*DELTA T_{ln}
 DELTA T_{ln}=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
 Q_{dot_conv_out}=h_o*A*(T_s-T_o)
 Q_{dot_rad_out}=epsilon*A*sigma*((T_s+273)⁴-(T_{surr}+273)⁴)
 sigma=5.67E-8 "[W/m²-K⁴], Stefan-Boltzmann constant"
 Q_{dot}=m_{dot}*c_p*(T_i-T_e)

Vel [m/s]	T _e [C]	Q [W]
1	33.85	1150
2	39.43	1810
3	42.78	2273
4	45.1	2622
5	46.83	2898
6	48.17	3122
7	49.25	3310
8	50.14	3469
9	50.89	3606
10	51.53	3726

ε	T_e [C]	Q [W]
0.1	45.82	2495
0.2	45.45	2560
0.3	45.1	2622
0.4	44.77	2680
0.5	44.46	2735
0.6	44.16	2787
0.7	43.88	2836
0.8	43.61	2883
0.9	43.36	2928
1	43.12	2970



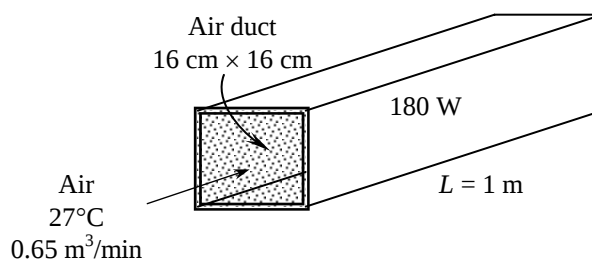
13-59 The components of an electronic system located in a rectangular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the duct are smooth. 3 The thermal resistance of the duct is negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 35°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.145 \text{ kg/m}^3 \\ k &= 0.02625 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.655 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7268\end{aligned}$$

Analysis (a) The mass flow rate of air and the exit temperature are determined from



$$\dot{m} = \rho \dot{V} = (1.145 \text{ kg/m}^3)(0.65 \text{ m}^3/\text{min}) = 0.7443 \text{ kg/min} = 0.0124 \text{ kg/s}$$

$$\dot{Q} = \dot{m} c_p (T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m} c_p} = 27^\circ\text{C} + \frac{(0.85)(180 \text{ W})}{(0.0124 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})} = 39.3^\circ\text{C}$$

(b) The mean fluid velocity and hydraulic diameter are

$$\begin{aligned}V_m &= \frac{\dot{V}}{A_c} = \frac{0.65 \text{ m}^3/\text{min}}{(0.16 \text{ m})(0.16 \text{ m})} = 25.4 \text{ m/min} = 0.4232 \text{ m/s} \\ D_h &= \frac{4A_c}{p} = \frac{4(0.16 \text{ m})(0.16 \text{ m})}{4(0.16 \text{ m})} = 0.16 \text{ m}\end{aligned}$$

Then

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.4232 \text{ m/s})(0.16 \text{ m})}{1.655 \times 10^{-5} \text{ m}^2/\text{s}} = 4091$$

which is not greater than 10,000 but the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(4091)^{0.8} (0.7268)^{0.4} = 15.69$$

and

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02625 \text{ W/m} \cdot ^\circ\text{C}}{0.16 \text{ m}} (15.69) = 2.574 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The highest component surface temperature will occur at the exit of the duct. Assuming uniform surface heat flux, its value is determined from

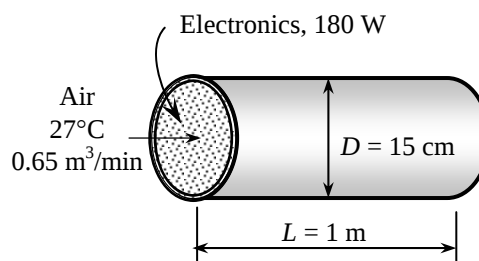
$$\begin{aligned}\dot{Q} / A_s &= h(T_{s, \text{highest}} - T_e) \\ T_{s, \text{highest}} &= T_e + \frac{\dot{Q} / A_s}{h} = 39.2^\circ\text{C} + \frac{(0.85)(180 \text{ W}) / [4(0.16 \text{ m})(1 \text{ m})]}{2.574 \text{ W/m}^2 \cdot ^\circ\text{C}} = 132^\circ\text{C}\end{aligned}$$

13-60 The components of an electronic system located in a circular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the duct are smooth. 3 The thermal resistance of the duct is negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 35°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.145 \text{ kg/m}^3 \\ k &= 0.02625 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.655 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7268\end{aligned}$$



Analysis (a) The mass flow rate of air and the exit temperature are determined from

$$\begin{aligned}\dot{m} &= \rho \dot{V} = (1.145 \text{ kg/m}^3)(0.65 \text{ m}^3/\text{min}) = 0.7443 \text{ kg/min} = 0.0124 \text{ kg/s} \\ \dot{Q} &= \dot{m} c_p (T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m} c_p} = 27^\circ\text{C} + \frac{(0.85)(180 \text{ W})}{(0.0124 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})} = \mathbf{39.3^\circ\text{C}}\end{aligned}$$

(b) The mean fluid velocity is

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.65 \text{ m}^3/\text{min}}{\pi(0.15 \text{ m})^2/4} = 36.8 \text{ m/min} = 0.613 \text{ m/s}$$

Then,

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.613 \text{ m/s})(0.15 \text{ m})}{1.655 \times 10^{-5} \text{ m}^2/\text{s}} = 5556$$

which is not greater than 10,000 but the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(5556)^{0.8} (0.7268)^{0.4} = 20.05$$

and

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02625 \text{ W/m} \cdot ^\circ\text{C}}{0.15 \text{ m}} (20.05) = 3.51 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The highest component surface temperature will occur at the exit of the duct. Assuming uniform heat flux, its value is determined from

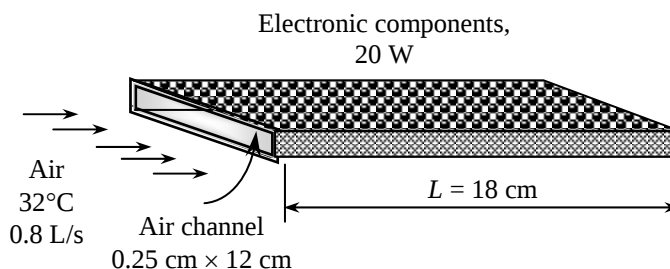
$$\dot{q} = h(T_{s, \text{highest}} - T_e) \rightarrow T_{s, \text{highest}} = T_e + \frac{\dot{q}}{h} = 39.2^\circ\text{C} + \frac{(0.85)(180 \text{ W})/[\pi(0.15 \text{ m})(1 \text{ m})]}{3.51 \text{ W/m}^2 \cdot ^\circ\text{C}} = \mathbf{131.7^\circ\text{C}}$$

13-61 Air enters a hollow-core printed circuit board. The exit temperature of the air and the highest temperature on the inner surface are to be determined.

Assumptions 1 Steady flow conditions exist. 2 Heat generated is uniformly distributed over the two surfaces of the PCB. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 40°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the hollow core whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.127 \text{ kg/m}^3 \\ k &= 0.02662 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.702 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7255 \\ \mu_b &= 1.918 \times 10^{-5} \text{ kg/m} \cdot \text{s} \\ \mu_{s, @ 60^\circ\text{C}} &= 2.008 \times 10^{-5} \text{ kg/m} \cdot \text{s}\end{aligned}$$



Analysis (a) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho \dot{V} = (1.127 \text{ kg/m}^3)(0.8 \times 10^{-3} \text{ m}^3/\text{s}) = 9.02 \times 10^{-4} \text{ kg/s}$$

$$\dot{Q} = \dot{m} c_p (T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m} c_p} = 32^\circ\text{C} + \frac{20 \text{ W}}{(9.02 \times 10^{-4} \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})} = 54.0^\circ\text{C}$$

(b) The mean fluid velocity and hydraulic diameter are

$$\begin{aligned}V_{\text{avg}} &= \frac{\dot{V}}{A_c} = \frac{0.8 \times 10^{-3} \text{ m}^3/\text{s}}{(0.12 \text{ m})(0.0025 \text{ m})} = 2.67 \text{ m/s} \\ D_h &= \frac{4A_c}{P} = \frac{4(0.12 \text{ m})(0.0025 \text{ m})}{2[(0.12 \text{ m}) + (0.0025 \text{ m})]} = 0.0049 \text{ m}\end{aligned}$$

Then,

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(2.67 \text{ m/s})(0.0049 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 769$$

which is less than 2300. Therefore, the flow is laminar and the thermal entry length in this case is

$$L_t = 0.05 \text{ Re Pr } D_h = 0.05(769)(0.7255)(0.0049 \text{ m}) = 0.14 \text{ m}$$

which is shorter than the total length of the duct. Therefore, we assume thermally developing flow, and determine the Nusselt number from

$$\text{Nu} = \frac{h D_h}{k} = 1.86 \left(\frac{\text{Re Pr } D}{L} \right)^{1/3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} = 1.86 \left[\frac{(769)(0.7255)(0.0049)}{0.18} \right]^{1/3} \left(\frac{1.918 \times 10^{-5}}{2.008 \times 10^{-5}} \right)^{0.14} = 4.58$$

and,

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02662 \text{ W/m} \cdot ^\circ\text{C}}{0.0049 \text{ m}} (4.58) = 24.9 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The highest component surface temperature will occur at the exit of the duct. Its value is determined from

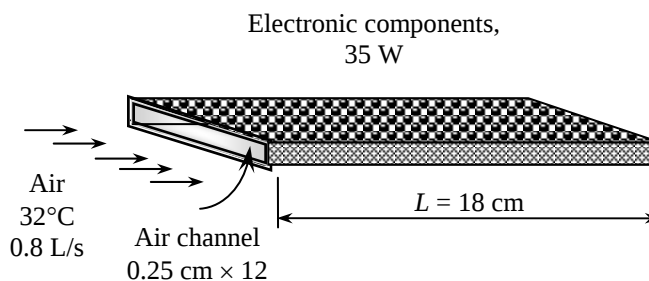
$$\begin{aligned}\dot{Q} &= h A_s (T_{s, \text{highest}} - T_e) \rightarrow T_{s, \text{highest}} = T_e + \frac{\dot{Q}}{h A_s} \\ &= 54.0^\circ\text{C} + \frac{20 \text{ W}}{(24.9 \text{ W/m}^2 \cdot ^\circ\text{C}) [2(0.12 \times 0.18 + 0.0025 \times 0.18) \text{ m}^2]} = 72.2^\circ\text{C}\end{aligned}$$

13-62 Air enters a hollow-core printed circuit board. The exit temperature of the air and the highest temperature on the inner surface are to be determined.

Assumptions 1 Steady flow conditions exist. 2 Heat generated is uniformly distributed over the two surfaces of the PCB. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 40°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the hollow core whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.127 \text{ kg/m}^3 \\ k &= 0.02662 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.702 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7255 \\ \mu_b &= 1.918 \times 10^{-5} \text{ kg/m} \cdot \text{s} \\ \mu_{s, @ 80^\circ\text{C}} &= 2.096 \times 10^{-5} \text{ kg/m} \cdot \text{s}\end{aligned}$$



Analysis (a) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho \dot{V} = (1.127 \text{ kg/m}^3)(0.8 \times 10^{-3} \text{ m}^3/\text{s}) = 9.02 \times 10^{-4} \text{ kg/s}$$

$$\dot{Q} = \dot{m} c_p (T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m} c_p} = 32^\circ\text{C} + \frac{35 \text{ W}}{(9.02 \times 10^{-4} \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C})} = 70.5^\circ\text{C}$$

(b) The mean fluid velocity and hydraulic diameter are

$$\begin{aligned}V_{\text{avg}} &= \frac{\dot{V}}{A_c} = \frac{0.8 \times 10^{-3} \text{ m}^3/\text{s}}{(0.12 \text{ m})(0.0025 \text{ m})} = 2.67 \text{ m/s} \\ D_h &= \frac{4A_c}{P} = \frac{4(0.12 \text{ m})(0.0025 \text{ m})}{2[(0.12 \text{ m}) + (0.0025 \text{ m})]} = 0.0049 \text{ m}\end{aligned}$$

Then,

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(2.67 \text{ m/s})(0.0049 \text{ m})}{1.702 \times 10^{-5} \text{ m}^2/\text{s}} = 769$$

which is less than 2300. Therefore, the flow is laminar and the thermal entry length in this case is

$$L_t = 0.05 \text{ Re Pr } D_h = 0.05(769)(0.71)(0.0049 \text{ m}) = 0.14 \text{ m}$$

which is shorter than the total length of the duct. Therefore, we assume thermally developing flow, and determine the Nusselt number from

$$\text{Nu} = \frac{h D_h}{k} = 1.86 \left(\frac{\text{Re Pr } D}{L} \right)^{1/3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} = 1.86 \left[\frac{(769)(0.7255)(0.0049)}{0.18} \right]^{1/3} \left(\frac{1.918 \times 10^{-5}}{2.096 \times 10^{-5}} \right)^{0.14} = 4.55$$

and,

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02662 \text{ W/m} \cdot ^\circ\text{C}}{0.0049 \text{ m}} (4.55) = 24.7 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The highest component surface temperature will occur at the exit of the duct. Its value is determined from

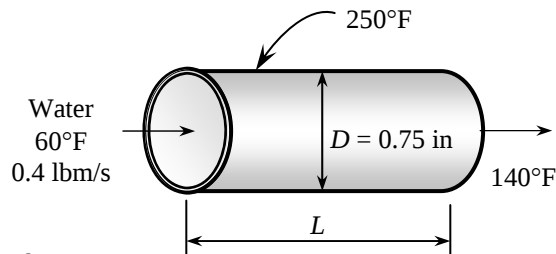
$$\begin{aligned}\dot{Q} &= h A_s (T_{s, \text{highest}} - T_e) \rightarrow T_{s, \text{highest}} = T_e + \frac{\dot{Q}}{h A_s} \\ &= 70.5^\circ\text{C} + \frac{35 \text{ W}}{(24.7 \text{ W/m}^2 \cdot ^\circ\text{C})[2(0.12 \times 0.18 + 0.0025 \times 0.18) \text{ m}^2]} = 102.6^\circ\text{C}\end{aligned}$$

13-63E Water is heated by passing it through thin-walled copper tubes. The length of the copper tube that needs to be used is to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the tube are smooth. 3 The thermal resistance of the tube is negligible. 4 The temperature at the tube surface is constant.

Properties The properties of water at the bulk mean fluid temperature of $T_{b,avg} = (60 + 140) / 2 = 100^\circ\text{F}$ are (Table A-15E)

$$\begin{aligned}\rho &= 62.0 \text{ lbm/ft}^3 \\ k &= 0.363 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F} \\ \nu &= \mu / \rho = 0.738 \times 10^{-5} \text{ ft}^2/\text{s} \\ c_p &= 0.999 \text{ Btu/lbm}\cdot^\circ\text{F} \\ \text{Pr} &= 4.54\end{aligned}$$



Analysis (a) The mass flow rate and the Reynolds number are

$$\dot{m} = \rho A_c V_{avg} \rightarrow V_{avg} = \frac{\dot{m}}{\rho A_c} = \frac{0.4 \text{ lbm/s}}{(62 \text{ lbm/ft}^3)[\pi(0.75/12 \text{ ft})^2/4]} = 2.10 \text{ ft/s}$$

$$\text{Re} = \frac{V_{avg} D_h}{\nu} = \frac{(2.10 \text{ ft/s})(0.75/12 \text{ ft})}{0.738 \times 10^{-5} \text{ ft}^2/\text{s}} = 17,810$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.75 \text{ in}) = 7.5 \text{ in}$$

which is probably shorter than the total length of the pipe we will determine. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(17,810)^{0.8} (4.54)^{0.4} = 105.9$$

and

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.363 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{(0.75/12 \text{ ft})} (105.9) = 615 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

The logarithmic mean temperature difference and then the rate of heat transfer per ft length of the tube are

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{140 - 60}{\ln\left(\frac{250 - 140}{250 - 60}\right)} = 146.4^\circ\text{F}$$

$$\dot{Q} = hA_s \Delta T_{\ln} = (615 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})[\pi(0.75/12 \text{ ft})(1 \text{ ft})](146.4^\circ\text{F}) = 17,675 \text{ Btu/h}$$

The rate of heat transfer needed to raise the temperature of water from 60°F to 140°F is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.4 \times 3600 \text{ lbm/h})(0.999 \text{ Btu/lbm}\cdot^\circ\text{F})(140 - 60)^\circ\text{F} = 115,085 \text{ Btu/h}$$

Then the length of the copper tube that needs to be used becomes

$$\text{Length} = \frac{115,085 \text{ Btu/h}}{17,675 \text{ Btu/h}} = \mathbf{6.51 \text{ ft}}$$

(b) The friction factor, the pressure drop, and then the pumping power required to overcome this pressure drop can be determined for the case of fully developed turbulent flow to be

$$f = 0.184 \text{Re}^{-0.2} = 0.184(17,810)^{-0.2} = 0.02598$$

$$\Delta P = f \frac{L}{D} \frac{\rho V_{avg}^2}{2} = 0.02598 \frac{(7.69 \text{ ft})}{(0.75/12 \text{ ft})} \frac{(62 \text{ lbm/ft}^3)(2.10 \text{ ft/s})^2}{2} \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm}\cdot\text{ft/s}^2} \right) = 13.58 \text{ lbf/ft}^2$$

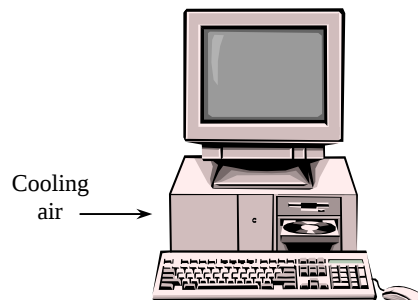
$$\dot{W}_{\text{pump}} = \frac{\dot{m} \Delta P}{\rho} = \frac{(0.4 \text{ lbm/s})(13.58 \text{ lbf/ft}^2)}{62 \text{ lbm/ft}^3} \left(\frac{1 \text{ hp}}{550 \text{ lbf}\cdot\text{ft/s}} \right) = \mathbf{0.00016 \text{ hp}}$$

13-64 A computer is cooled by a fan blowing air through its case. The flow rate of the air, the fraction of the temperature rise of air that is due to heat generated by the fan, and the highest allowable inlet air temperature are to be determined.

Assumptions 1 Steady flow conditions exist. 2 Heat flux is uniformly distributed. 3 Air is an ideal gas with constant properties. 4 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 25°C. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.184 \text{ kg/m}^3 \\ k &= 0.02551 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= 1.562 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg}\cdot^\circ\text{C} \\ \text{Pr} &= 0.7296\end{aligned}$$



Analysis (a) Noting that the electric energy consumed by the fan is converted to thermal energy, the mass flow rate of air is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q} + \dot{W}_{\text{elect, fan}}}{c_p(T_e - T_i)} = \frac{(8 \times 10 + 25) \text{ W}}{(1007 \text{ J/kg}\cdot^\circ\text{C})(10^\circ\text{C})} = \mathbf{0.01043 \text{ kg/s}}$$

(b) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor is

$$\begin{aligned}\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T &= \frac{\dot{Q}}{\dot{m}c_p} = \frac{25 \text{ W}}{(0.01043 \text{ kg/s})(1007 \text{ J/kg}\cdot^\circ\text{C})} = \mathbf{2.38^\circ\text{C}} \\ f &= \frac{2.38^\circ\text{C}}{10^\circ\text{C}} = 0.238 = \mathbf{23.8\%}\end{aligned}$$

(c) The mean velocity of air is

$$\dot{m} = \rho A_c V_{\text{avg}} \rightarrow V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{(0.01043 / 8) \text{ kg/s}}{(1.184 \text{ kg/m}^3)[(0.003 \text{ m})(0.12 \text{ m})]} = 3.06 \text{ m/s}$$

$$\text{and } D_h = \frac{4A_c}{P} = \frac{4(0.003 \text{ m})(0.12 \text{ m})}{2(0.003 \text{ m} + 0.12 \text{ m})} = 0.00585 \text{ m}$$

Therefore,

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.06 \text{ m/s})(0.00585 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 1146$$

which is less than 2300. Therefore, the flow is laminar. Assuming fully developed flow, the Nusselt number is determined from Table 13-4 corresponding to $a/b = 12/0.3 = 40$ to be $\text{Nu} = 8.24$. Then,

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02551 \text{ W/m}\cdot^\circ\text{C}}{0.00585 \text{ m}} (8.24) = 35.9 \text{ W/m}^2\cdot^\circ\text{C}$$

The highest component surface temperature will occur at the exit of the duct. Assuming uniform heat flux, the air temperature at the exit is determined from

$$\dot{q} = h(T_{s,\text{max}} - T_e) \rightarrow T_e = T_{s,\text{max}} - \frac{\dot{q}}{h} = 70^\circ\text{C} - \frac{[(80 + 25) \text{ W}]/[8 \times 2(0.12 \times 0.18 + 0.003 \times 0.18) \text{ m}^2]}{35.9 \text{ W/m}^2\cdot^\circ\text{C}} = 61.7^\circ\text{C}$$

The highest allowable inlet temperature then becomes

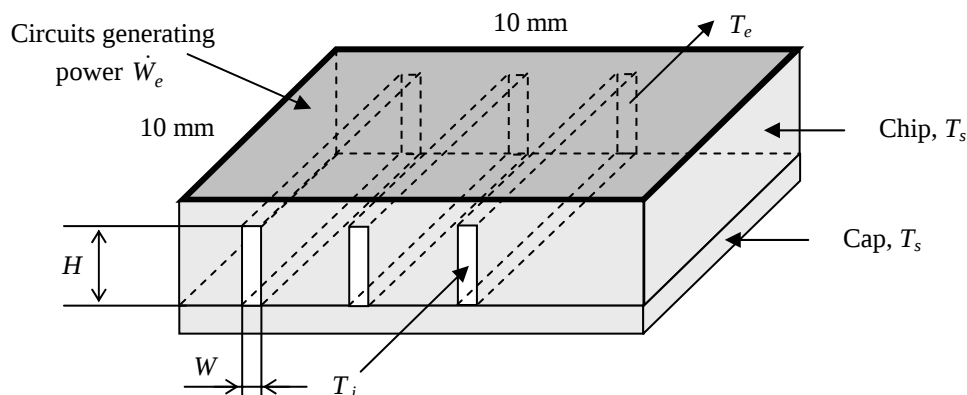
$$T_e - T_i = 10^\circ\text{C} \rightarrow T_i = T_e - 10^\circ\text{C} = 61.7^\circ\text{C} - 10^\circ\text{C} = \mathbf{51.7^\circ\text{C}}$$

Discussion Although the Reynolds number is less than 2300, the flow in this case will most likely be turbulent because of the electronic components that protrude into flow. Therefore, the heat transfer coefficient determined above is probably conservative.

Review Problems

13-65 A silicon chip is cooled by passing water through microchannels etched in the back of the chip. The outlet temperature of water and the chip power dissipation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The flow of water is fully developed. 3 All the heat generated by the circuits on the top surface of the chip is transferred to the water.



Properties Assuming a bulk mean fluid temperature of 25°C, the properties of water are (Table A-15)

$$\begin{aligned}\rho &= 997 \text{ kg/m}^3 & c_p &= 4180 \text{ J/kg}\cdot^\circ\text{C} \\ k &= 0.607 \text{ W/m}\cdot^\circ\text{C} & \text{Pr} &= 6.14 \\ \mu &= 0.891 \times 10^{-3} \text{ m}^2/\text{s}\end{aligned}$$

Analysis (a) The mass flow rate for one channel, the hydraulic diameter, and the Reynolds number are

$$\dot{m} = \frac{\dot{m}_{\text{total}}}{n_{\text{channel}}} = \frac{0.005 \text{ kg/s}}{50} = 0.0001 \text{ kg/s}$$

$$D_h = \frac{4A}{p} = \frac{4(H \times W)}{2(H + W)} = \frac{4(50 \times 200)}{2(50 + 200)} = 80 \text{ }\mu\text{m} = 8 \times 10^{-5} \text{ m}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{\rho \dot{m} V D_h}{\rho A_c \mu} = \frac{\dot{m} D_h}{A_c \mu} = \frac{(0.0001 \text{ kg/s})(8 \times 10^{-5} \text{ m})}{(50 \times 200 \times 10^{-12} \text{ m}^2)(0.891 \times 10^{-3} \text{ kg/m}\cdot\text{s})} = 898$$

which is smaller than 2300. Therefore, the flow is laminar. We take fully developed laminar flow in the entire duct. The Nusselt number in this case is

$$Nu = 3.66$$

Heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.607 \text{ W/m}\cdot^\circ\text{C}}{8 \times 10^{-5} \text{ m}} (3.66) = 27,770 \text{ W/m}^2\cdot^\circ\text{C}$$

Next we determine the exit temperature of water

$$A = 2WL + 2HL = 2(0.05 \times 0.01) + 2(0.05 \times 0.2) = 0.021 \text{ mm}^2 = 2.1 \times 10^{-6} \text{ m}^2$$

$$T_e = T_s - (T_s - T_i) e^{-hA/(\dot{m}c_p)} = 350 - (350 - 290) \exp\left[-\frac{(27,770)(2.1 \times 10^{-6})}{(0.0001)(4180)}\right] = \mathbf{297.8 \text{ K}}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.0001 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(350 - 297.8)^\circ\text{C} = 21.82 \text{ W}$$

(b) Noting that there are 50 such channels, the chip power dissipation becomes

$$\dot{W}_e = n_{\text{channel}} \dot{Q}_{\text{one channel}} = 50(21.82 \text{ W}) = \mathbf{1091 \text{ W}}$$

13-66 Water is heated by passing it through five identical tubes that are maintained at a specified temperature. The rate of heat transfer and the length of the tubes necessary are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The surface temperature is constant and uniform. 3 The inner surfaces of the tubes are smooth. 4 Heat transfer to the surroundings is negligible.

Properties The properties of water at the bulk mean fluid temperature of $(15+35)/2=25^\circ\text{C}$ are (Table A-15)

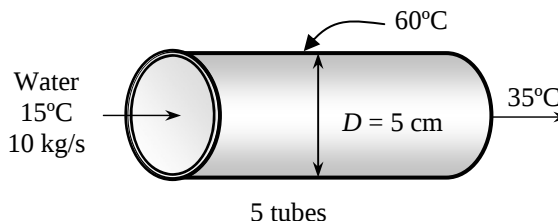
$$\rho = 997 \text{ kg/m}^3$$

$$k = 0.607 \text{ W/m}\cdot^\circ\text{C}$$

$$\mu = 0.891 \times 10^{-3} \text{ m}^2/\text{s}$$

$$c_p = 4180 \text{ J/kg}\cdot^\circ\text{C}$$

$$\text{Pr} = 6.14$$



Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (10 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(35 - 15)^\circ\text{C} = \mathbf{836,000 \text{ W}}$$

(b) The water velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{(10/5) \text{ kg/s}}{(997 \text{ kg/m}^3)\pi(0.05 \text{ m})^2/4} = 1.02 \text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(997 \text{ kg/m}^3)(1.02 \text{ m/s})(0.05 \text{ m})}{0.891 \times 10^{-3} \text{ kg/m}\cdot\text{s}} = 57,067$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(57,067)^{0.8}(6.14)^{0.4} = 303.5$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m}\cdot^\circ\text{C}}{0.05 \text{ m}}(303.5) = 3684 \text{ W/m}^2\cdot^\circ\text{C}$$

Using the average fluid temperature and considering that there are 5 tubes, the length of the tubes is determined as follows:

$$\dot{Q} = hA(T_s - T_{b,\text{avg}}) \longrightarrow 836,000 \text{ W} = (3684 \text{ W/m}^2\cdot^\circ\text{C})A(60 - 25)^\circ\text{C} \longrightarrow A_s = 6.484 \text{ m}^2$$

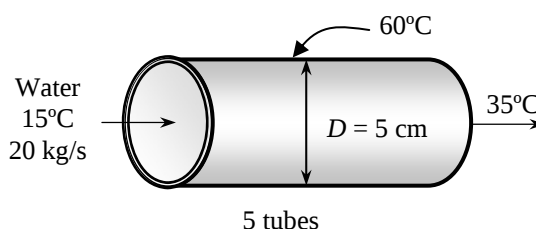
$$A = 5\pi DL \longrightarrow L = \frac{A}{5\pi D} = \frac{6.484 \text{ m}^2}{5\pi(0.05 \text{ m})} = \mathbf{8.26 \text{ m}}$$

13-67 Water is heated by passing it through five identical tubes that are maintained at a specified temperature. The rate of heat transfer and the length of the tubes necessary are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The surface temperature is constant and uniform. 3 The inner surfaces of the tubes are smooth. 4 Heat transfer to the surroundings is negligible.

Properties The properties of water at the bulk mean fluid temperature of $(15+35)/2=25^\circ\text{C}$ are (Table A-15)

$$\begin{aligned}\rho &= 997 \text{ kg/m}^3 \\ k &= 0.607 \text{ W/m}\cdot^\circ\text{C} \\ \mu &= 0.891 \times 10^{-3} \text{ m}^2/\text{s} \\ c_p &= 4180 \text{ J/kg}\cdot^\circ\text{C} \\ \text{Pr} &= 6.14\end{aligned}$$



Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (20 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(35 - 15)^\circ\text{C} = \mathbf{1,672,000 \text{ W}}$$

(b) The water velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{(20/5) \text{ kg/s}}{(997 \text{ kg/m}^3) [\pi (0.05 \text{ m})^2 / 4]} = 2.04 \text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(997 \text{ kg/m}^3)(2.04 \text{ m/s})(0.05 \text{ m})}{0.891 \times 10^{-3} \text{ kg/m}\cdot\text{s}} = 114,320$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(114,320)^{0.8} (6.14)^{0.4} = 529.0$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m}\cdot^\circ\text{C}}{0.05 \text{ m}} (529.0) = 6423 \text{ W/m}^2\cdot^\circ\text{C}$$

Using the average fluid temperature and considering that there are 5 tubes, the length of the tubes is determined as follows:

$$\begin{aligned}\dot{Q} &= hA(T_s - T_{b,\text{avg}}) \longrightarrow 1,672,000 \text{ W} = (6423 \text{ W/m}^2\cdot^\circ\text{C})A(60 - 25)^\circ\text{C} \longrightarrow A_s = 7.438 \text{ m}^2 \\ A &= 5\pi DL \longrightarrow L = \frac{A}{5\pi D} = \frac{7.438 \text{ m}^2}{5\pi(0.05 \text{ m})} = \mathbf{9.47 \text{ m}}\end{aligned}$$

13-68 Water is heated as it flows in a smooth tube that is maintained at a specified temperature. The necessary tube length and the water outlet temperature if the tube length is doubled are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The surface temperature is constant and uniform. 3 The inner surfaces of the tube are smooth. 4 Heat transfer to the surroundings is negligible.

Properties The properties of water at the bulk mean fluid temperature of $(10+40)/2=25^\circ\text{C}$ are (Table A-15)

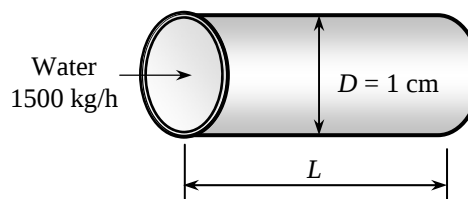
$$\rho = 997 \text{ kg/m}^3$$

$$k = 0.607 \text{ W/m}\cdot^\circ\text{C}$$

$$\mu = 0.891 \times 10^{-3} \text{ m}^2/\text{s}$$

$$c_p = 4180 \text{ J/kg}\cdot^\circ\text{C}$$

$$\text{Pr} = 6.14$$



Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (1500 / 3600 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(40 - 10)^\circ\text{C} = 52,250 \text{ W}$$

The water velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{(1500 / 3600) \text{ kg/s}}{(997 \text{ kg/m}^3)[\pi(0.01 \text{ m})^2 / 4]} = 5.32 \text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(997 \text{ kg/m}^3)(5.32 \text{ m/s})(0.01 \text{ m})}{0.891 \times 10^{-3} \text{ kg/m}\cdot\text{s}} = 59,542$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(59,542)^{0.8} (6.14)^{0.4} = 313.9$$

Heat transfer coefficient is

$$h = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m}\cdot^\circ\text{C}}{0.01 \text{ m}} (313.9) = 19,056 \text{ W/m}^2\cdot^\circ\text{C}$$

Using the average fluid temperature, the length of the tubes is determined as follows:

$$\dot{Q} = hA(T_s - T_{b,\text{avg}}) \longrightarrow 52,250 \text{ W} = (19,056 \text{ W/m}^2\cdot^\circ\text{C})A(49 - 25)^\circ\text{C} \longrightarrow A_s = 0.1142 \text{ m}^2$$

$$A = \pi DL \longrightarrow 0.1142 \text{ m}^2 = \pi(0.01 \text{ m})L \longrightarrow L = \mathbf{3.6 \text{ m}}$$

(b) If the tube length is doubled, the surface area doubles, and the outlet water temperature may be obtained from an energy balance to be

$$\dot{m}c_p(T_e - T_i) = hA_s(T_s - T_{b,\text{avg}})$$

$$(1500 / 3600)(4180)(T_e - 10) = (19,056)(2 \times 0.1142) \left(49 - \frac{10 + T_e}{2} \right)$$

$$T_e = 53.3^\circ\text{C}$$

which is greater than the surface temperature of the wall. This is impossible. It shows that the water reaches the surface temperature before the entire length of tube is covered and in reality the water will leave the tube at the surface temperature of 49°C . This example demonstrates that the use of unnecessarily long tubes should be avoided.

13-69 Geothermal water is supplied to a city through stainless steel pipes at a specified rate. The electric power consumption and its daily cost are to be determined, and it is to be assessed if the frictional heating during flow can make up for the temperature drop caused by heat loss.

Assumptions 1 The flow is steady and incompressible. 2 The entrance effects are negligible, and thus the flow is fully developed. 3 The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. 4 The geothermal well and the city are at about the same elevation. 5 The properties of geothermal water are the same as fresh water. 6 The fluid pressures at the wellhead and the arrival point in the city are the same.

Properties The properties of water at 110°C are $\rho = 950.6 \text{ kg/m}^3$, $\mu = 0.255 \times 10^{-3} \text{ kg/m}\cdot\text{s}$, and $c_p = 4.229 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-15). The roughness of stainless steel pipes is $2 \times 10^{-6} \text{ m}$ (Table 13-3).

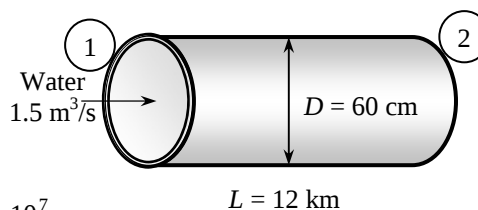
Analysis (a) We take point 1 at the well-head of geothermal resource and point 2 at the final point of delivery at the city, and the entire piping system as the control volume. Both points are at the same elevation ($z_1 = z_2$) and the same velocity ($V_1 = V_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_{\text{turbine}} + h_L \rightarrow h_{\text{pump,u}} = h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. The mean velocity and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2 / 4} = \frac{1.5 \text{ m}^3/\text{s}}{\pi (0.60 \text{ m})^2 / 4} = 5.305 \text{ m/s}$$

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{(950.6 \text{ kg/m}^3)(5.305 \text{ m/s})(0.60 \text{ m})}{0.255 \times 10^{-3} \text{ kg/m}\cdot\text{s}} = 1.186 \times 10^7$$



which is greater than 10,000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D = \frac{2 \times 10^{-6} \text{ m}}{0.60 \text{ m}} = 3.33 \times 10^{-6}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0 \log \left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re} \sqrt{f}} \right) \rightarrow \frac{1}{\sqrt{f}} = -2.0 \log \left(\frac{3.33 \times 10^{-6}}{3.7} + \frac{2.51}{1.187 \times 10^7 \sqrt{f}} \right)$$

It gives $f = 0.00829$. Then the pressure drop, the head loss, and the required power input become

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.00829 \frac{12,000 \text{ m}}{0.60 \text{ m}} \frac{(950.6 \text{ kg/m}^3)(5.305 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) = 2218 \text{ kPa}$$

$$\dot{W}_{\text{elect}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V} \Delta P}{\eta_{\text{pump-motor}}} = \frac{(1.5 \text{ m}^3/\text{s})(2218 \text{ kPa})}{0.65} \left(\frac{1 \text{ kW}}{1 \text{ kPa}\cdot\text{m}^3/\text{s}} \right) = 5118 \text{ kW}$$

Therefore, the pumps will consume 5118 kW of electric power to overcome friction and maintain flow.

(b) The daily cost of electric power consumption is determined by multiplying the amount of power used per day by the unit cost of electricity,

$$\text{Amount} = \dot{W}_{\text{elect,in}} \Delta t = (5118 \text{ kW})(24 \text{ h/day}) = 122,832 \text{ kWh/day}$$

$$\text{Cost} = \text{Amount} \times \text{Unit cost} = (122,832 \text{ kWh/day})(\$0.06/\text{kWh}) = \mathbf{\$7370/\text{day}}$$

(c) The energy consumed by the pump (except the heat dissipated by the motor to the air) is eventually dissipated as heat due to the frictional effects. Therefore, this problem is equivalent to heating the water by a 5118 kW of resistance heater (again except the heat dissipated by the motor). To be conservative, we consider only the useful mechanical energy supplied to the water by the pump. The temperature rise of water due to this addition of energy is

$$\dot{W}_{\text{elect}} = \rho \dot{V}_p \Delta T \rightarrow \Delta T = \frac{\eta_{\text{pump-motor}} \dot{W}_{\text{elect,in}}}{\rho \dot{V}_p} = \frac{0.65 \times (5118 \text{ kJ/s})}{(950.6 \text{ kg/m}^3)(1.5 \text{ m}^3/\text{s})(4.229 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{0.55^\circ\text{C}}$$

Therefore, the temperature of water will rise at least 0.55°C, which is more than the 0.5°C drop in temperature (in reality, the temperature rise will be more since the energy dissipation due to pump inefficiency will also appear as temperature rise of water). Thus we conclude that the frictional heating during flow can more than make up for the temperature drop caused by heat loss.

Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

13-70 Geothermal water is supplied to a city through cast iron pipes at a specified rate. The electric power consumption and its daily cost are to be determined, and it is to be assessed if the frictional heating during flow can make up for the temperature drop caused by heat loss.

Assumptions 1 The flow is steady and incompressible. 2 The entrance effects are negligible, and thus the flow is fully developed. 3 The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. 4 The geothermal well and the city are at about the same elevation. 5 The properties of geothermal water are the same as fresh water. 6 The fluid pressures at the wellhead and the arrival point in the city are the same.

Properties The properties of water at 110°C are $\rho = 950.6 \text{ kg/m}^3$, $\mu = 0.255 \times 10^{-3} \text{ kg/m}\cdot\text{s}$, and $c_p = 4.229 \text{ kJ/kg}\cdot^\circ\text{C}$ (Table A-15). The roughness of cast iron pipes is 0.00026 m (Table 13-3).

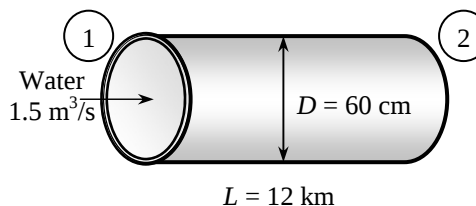
Analysis (a) We take point 1 at the well-head of geothermal resource and point 2 at the final point of delivery at the city, and the entire piping system as the control volume. Both points are at the same elevation ($z_1 = z_2$) and the same velocity ($V_1 = V_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_{\text{turbine}} + h_L \rightarrow h_{\text{pump,u}} = h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. The mean velocity and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2 / 4} = \frac{1.5 \text{ m}^3/\text{s}}{\pi (0.60 \text{ m})^2 / 4} = 5.305 \text{ m/s}$$

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{(950.6 \text{ kg/m}^3)(5.305 \text{ m/s})(0.60 \text{ m})}{0.255 \times 10^{-3} \text{ kg/m}\cdot\text{s}} = 1.187 \times 10^7$$



which is greater than 10,000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D = \frac{0.00026 \text{ m}}{0.60 \text{ m}} = 4.33 \times 10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0 \log \left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re} \sqrt{f}} \right) \rightarrow \frac{1}{\sqrt{f}} = -2.0 \log \left(\frac{4.33 \times 10^{-4}}{3.7} + \frac{2.51}{1.187 \times 10^7 \sqrt{f}} \right)$$

It gives $f = 0.01623$. Then the pressure drop, the head loss, and the required power input become

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.01623 \frac{12,000 \text{ m}}{0.60 \text{ m}} \frac{(950.6 \text{ kg/m}^3)(5.305 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2} \right) \left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2} \right) = 4342 \text{ kPa}$$

$$\dot{W}_{\text{elect}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V} \Delta P}{\eta_{\text{pump-motor}}} = \frac{(1.5 \text{ m}^3/\text{s})(4342 \text{ kPa})}{0.65} \left(\frac{1 \text{ kW}}{1 \text{ kPa}\cdot\text{m}^3/\text{s}} \right) = 10,020 \text{ kW}$$

Therefore, the pumps will consume 10,017 W of electric power to overcome friction and maintain flow.

(b) The daily cost of electric power consumption is determined by multiplying the amount of power used per day by the unit cost of electricity,

$$\text{Amount} = \dot{W}_{\text{elect},\text{in}} \Delta t = (10,020 \text{ kW})(24 \text{ h/day}) = 240,480 \text{ kWh/day}$$

$$\text{Cost} = \text{Amount} \times \text{Unit cost} = (240,480 \text{ kWh/day})(\$0.06/\text{kWh}) = \mathbf{\$14,430/\text{day}}$$

(c) The energy consumed by the pump (except the heat dissipated by the motor to the air) is eventually dissipated as heat due to the frictional effects. Therefore, this problem is equivalent to heating the water by a 10,020 kW of resistance heater (again except the heat dissipated by the motor). To be conservative, we consider only the useful mechanical energy supplied to the water by the pump. The temperature rise of water due to this addition of energy is

$$\dot{W}_{\text{elect}} = \rho \dot{V} c_p \Delta T \rightarrow \Delta T = \frac{\eta_{\text{pump-motor}} \dot{W}_{\text{elect},\text{in}}}{\rho \dot{V} c_p} = \frac{0.65 \times (10,020 \text{ kJ/s})}{(950.6 \text{ kg/m}^3)(1.5 \text{ m}^3/\text{s})(4.229 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{1.08^\circ\text{C}}$$

Therefore, the temperature of water will rise at least 1.08°C, which is more than the 0.5°C drop in temperature (in reality, the temperature rise will be more since the energy dissipation due to pump inefficiency will also appear as temperature rise of water). Thus we conclude that the frictional heating during flow can more than make up for the temperature drop caused by heat loss.

Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

13-71 The velocity profile in fully developed laminar flow in a circular pipe is given. The radius of the pipe, the mean velocity, and the maximum velocity are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is

$$u(r) = V_{\max} \left(1 - \frac{r^2}{R^2} \right)$$

The velocity profile in this case is given by

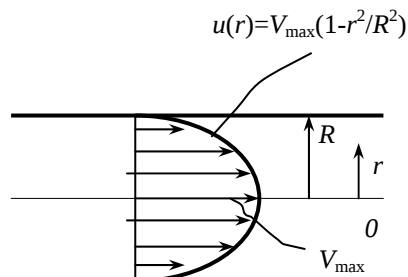
$$u(r) = 6(1 - 100r^2)$$

Comparing the two relations above gives the pipe radius, the maximum velocity, and the mean velocity to be

$$R^2 = \frac{1}{100} \quad \rightarrow \quad R = \mathbf{0.10 \text{ m}}$$

$$V_{\max} = \mathbf{6 \text{ m/s}}$$

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{6 \text{ m/s}}{2} = \mathbf{3 \text{ m/s}}$$



13-72E The velocity profile in fully developed laminar flow in a circular pipe is given. The volume flow rate, the pressure drop, and the useful pumping power required to overcome this pressure drop are to be determined.

Assumptions 1 The flow is steady, laminar, and fully developed. 2 The pipe is horizontal.

Properties The density and dynamic viscosity of water at 40°F are $\rho = 62.42 \text{ lbm/ft}^3$ and $\mu = 1.308 \times 10^{-3} \text{ lbm/ft} \cdot \text{s}$, respectively (Table A-15E).

Analysis The velocity profile in fully developed laminar flow in a circular pipe is

$$u(r) = V_{\max} \left(1 - \frac{r^2}{R^2} \right)$$

The velocity profile in this case is given by

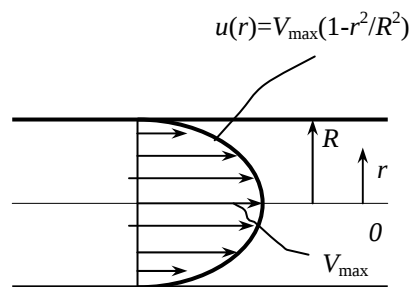
$$u(r) = 0.8(1 - 625r^2)$$

Comparing the two relations above gives the pipe radius, the maximum velocity, and the mean velocity to be

$$R^2 = \frac{1}{625} \rightarrow R = 0.04 \text{ ft}$$

$$V_{\max} = 0.8 \text{ ft/s}$$

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{0.8 \text{ ft/s}}{2} = 0.4 \text{ ft/s}$$



Then the volume flow rate and the pressure drop become

$$\dot{V} = V_{\text{avg}} A_c = V_{\text{avg}} (\pi R^2) = (0.4 \text{ ft/s}) [\pi (0.04 \text{ ft})^2] = \mathbf{0.00201 \text{ ft}^3/\text{s}}$$

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128 \mu L} \rightarrow 0.00201 \text{ ft}^3/\text{s} = \frac{(\Delta P) \pi (0.08 \text{ ft})^4}{128 (1.308 \times 10^{-3} \text{ lbm/ft} \cdot \text{s}) (140 \text{ ft})} \left(\frac{32.2 \text{ lbm} \cdot \text{ft/s}^2}{1 \text{ lbf}} \right)$$

It gives

$$\Delta P = 11.37 \text{ lbf/ft}^2 = \mathbf{0.0790 \text{ psi}}$$

Then the useful pumping power requirement becomes

$$\dot{W}_{\text{pump,u}} = \dot{V} \Delta P = (0.00201 \text{ ft}^3/\text{s}) (11.37 \text{ lbf/ft}^2) \left(\frac{1 \text{ W}}{0.737 \text{ lbf} \cdot \text{ft/s}} \right) = \mathbf{0.031 \text{ W}}$$

Checking The flow was assumed to be laminar. To verify this assumption, we determine the Reynolds number:

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{(62.42 \text{ lbm/ft}^3) (0.4 \text{ ft/s}) (0.08 \text{ ft})}{1.308 \times 10^{-3} \text{ lbm/ft} \cdot \text{s}} = 1527$$

which is less than 2300. Therefore, the flow is laminar.

Discussion Note that the pressure drop across the water pipe and the required power input to maintain flow is negligible. This is due to the very low flow velocity. Such water flows are the exception in practice rather than the rule.

13-73 A compressor is connected to the outside through a circular duct. The power used by compressor to overcome the pressure drop, the rate of heat transfer, and the temperature rise of air are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the duct are smooth. 3 The thermal resistance of the duct is negligible. 4 Air is an ideal gas with constant properties.

Properties We take the bulk mean temperature for air to be 15°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a higher temperature. The properties of air at this temperature and 1 atm pressure are (Table A-22)

$$\rho = 1.225 \text{ kg/m}^3, \quad k = 0.02476 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.470 \times 10^{-5} \text{ m}^2/\text{s}, \quad c_p = 1007 \text{ J/kg} \cdot ^\circ\text{C}, \quad \text{Pr} = 0.7323$$

The density and kinematic viscosity at 95 kPa are

$$P = \frac{95 \text{ kPa}}{101.325 \text{ kPa}} = 0.938 \text{ atm}$$

$$\rho = (1.225 \text{ kg/m}^3)(0.938) = 1.149 \text{ kg/m}^3$$

$$\nu = (1.470 \times 10^{-5} \text{ m}^2/\text{s})/(0.938) = 1.378 \times 10^{-5} \text{ m}^2/\text{s}$$

Analysis The mean velocity of air is

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.27 \text{ m}^3/\text{s}}{\pi(0.2 \text{ m})^2/4} = 8.594 \text{ m/s}$$

$$\text{Then } \text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(8.594 \text{ m/s})(0.2 \text{ m})}{1.378 \times 10^{-5} \text{ m}^2/\text{s}} = 1.247 \times 10^5$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.2 \text{ m}) = 2 \text{ m}$$

which is shorter than the total length of the duct. Therefore, we assume fully developed flow in a smooth pipe, and determine friction factor from

$$f = (0.790 \ln \text{Re} - 1.64)^{-2} = [0.790 \ln(1.247 \times 10^5) - 1.64]^{-2} = 0.01718$$

The pressure drop and the compressor power required to overcome this pressure drop are

$$\dot{m} = \rho \dot{V} = (1.149 \text{ kg/m}^3)(0.27 \text{ m}^3/\text{s}) = 0.3102 \text{ kg/s}$$

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = (0.01718) \frac{(11 \text{ m})}{(0.2 \text{ m})} \frac{(1.149 \text{ kg/m}^3)(8.594 \text{ m/s})^2}{2} = 40.09 \text{ N/m}^2$$

$$\dot{W}_{\text{pump}} = \frac{\dot{m} \Delta P}{\rho} = \frac{(0.3102 \text{ kg/s})(40.09 \text{ N/m}^2)}{1.149 \text{ kg/m}^3} = \mathbf{10.8 \text{ W}}$$

(b) For the fully developed turbulent flow, the Nusselt number is

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(1.247 \times 10^5)^{0.8} (0.7323)^{0.4} = 242.3$$

$$\text{and } h = \frac{k}{D_h} \text{Nu} = \frac{0.02476 \text{ W/m} \cdot ^\circ\text{C}}{0.2 \text{ m}} (242.3) = 30.00 \text{ W/m}^2 \cdot ^\circ\text{C}$$

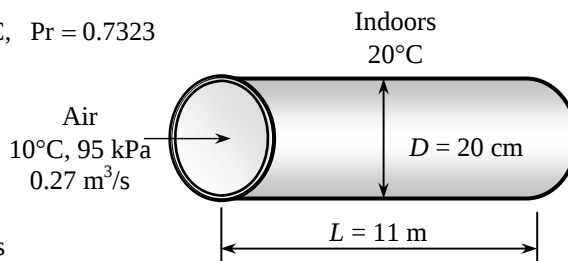
Disregarding the thermal resistance of the duct, the rate of heat transfer to the air in the duct becomes

$$A_s = \pi DL = \pi(0.2 \text{ m})(11 \text{ m}) = 6.912 \text{ m}^2$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{\frac{1}{h_1 A_s} + \frac{1}{h_2 A_s}} = \frac{20 - 10}{\frac{1}{(30.00)(6.912)} + \frac{1}{(10)(6.912)}} = \mathbf{518.4 \text{ W}}$$

(c) The temperature rise of air in the duct is

$$\dot{Q} = \dot{m} c_p \Delta T \rightarrow 518.4 \text{ W} = (0.3102 \text{ kg/s})(1007 \text{ J/kg} \cdot ^\circ\text{C}) \Delta T \rightarrow \Delta T = \mathbf{1.7^\circ\text{C}}$$



13-74 Air enters the underwater section of a duct. The outlet temperature of the air and the fan power needed to overcome the flow resistance are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the duct are smooth. 3 The thermal resistance of the duct is negligible. 4 The surface of the duct is at the temperature of the water. 5 Air is an ideal gas with constant properties. 6 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 20°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.204 \text{ kg/m}^3 \\ k &= 0.02514 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.516 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7309\end{aligned}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 3.958 \times 10^4$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.2 \text{ m}) = 2 \text{ m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(3.958 \times 10^4)^{0.8} (0.7309)^{0.3} = 99.76$$

and

$$h = \frac{k}{D_h} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{0.2 \text{ m}} (99.76) = 12.54 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Next we determine the exit temperature of air,

$$A_s = \pi DL = \pi(0.2 \text{ m})(15 \text{ m}) = 9.425 \text{ m}^2$$

$$\dot{m} = \rho V_{\text{avg}} A_c = (1.204 \text{ kg/m}^3)(3 \text{ m/s}) \left(\frac{\pi(0.2 \text{ m})^2}{4} \right) = (1.204 \text{ kg/m}^3)(0.09425 \text{ m}^3/\text{s}) = 0.1135 \text{ kg/s}$$

and

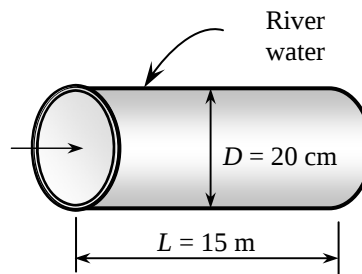
$$T_e = T_s - (T_s - T_i) e^{-hA_s / (\dot{m}c_p)} = 15 - (15 - 25) e^{-\frac{(12.54)(9.425)}{(0.1135)(1007)}} = 18.6^\circ\text{C}$$

The friction factor, pressure drop, and the fan power required to overcome this pressure drop can be determined for the case of fully developed turbulent flow in smooth pipes to be

$$f = (0.790 \ln \text{Re} - 1.64)^{-2} = [0.790 \ln(3.958 \times 10^4) - 1.64]^{-2} = 0.02212$$

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.02212 \frac{15 \text{ m}}{0.2 \text{ m}} \frac{(1.204 \text{ kg/m}^3)(3 \text{ m/s})^2}{2} \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ Pa}}{1 \text{ N/m}^2} \right) = 8.988 \text{ Pa}$$

$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V} \Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.09425 \text{ m}^3/\text{s})(8.988 \text{ Pa})}{0.55} = \left(\frac{1 \text{ W}}{1 \text{ Pa} \cdot \text{m}^3/\text{s}} \right) = 1.54 \text{ W}$$



13-75 Air enters the underwater section of a duct. The outlet temperature of the air and the fan power needed to overcome the flow resistance are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the duct are smooth. 3 The thermal resistance of the duct is negligible. 4 Air is an ideal gas with constant properties. 5 The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 20°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at 1 atm and this temperature are (Table A-22)

$$\begin{aligned}\rho &= 1.204 \text{ kg/m}^3 \\ k &= 0.02514 \text{ W/m} \cdot ^\circ\text{C} \\ \nu &= 1.516 \times 10^{-5} \text{ m}^2/\text{s} \\ c_p &= 1007 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr} &= 0.7309\end{aligned}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 3.958 \times 10^4$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.2 \text{ m}) = 2 \text{ m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number and h from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(3.958 \times 10^4)^{0.8} (0.7309)^{0.3} = 99.76$$

$$\text{and } h = \frac{k}{D_h} \text{Nu} = \frac{0.02514 \text{ W/m} \cdot ^\circ\text{C}}{0.2 \text{ m}} (99.76) = 12.54 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Next we determine the exit temperature of air,

$$A_s = \pi DL = \pi(0.2 \text{ m})(15 \text{ m}) = 9.425 \text{ m}^2$$

$$\dot{m} = \rho V_{\text{avg}} A_c = (1.204 \text{ kg/m}^3)(3 \text{ m/s}) \left(\frac{\pi(0.2 \text{ m})^2}{4} \right) = (1.204 \text{ kg/m}^3)(0.09425 \text{ m}^2/\text{s}) = 0.1135 \text{ kg/s}$$

The unit thermal resistance of the mineral deposit is

$$R_{\text{mineral}} = \frac{L}{k} = \frac{0.0025 \text{ m}}{3 \text{ W/m} \cdot ^\circ\text{C}} = 0.00083 \text{ m}^2 \cdot ^\circ\text{C/W}$$

which is much less than (about 1%) the unit convection resistance,

$$R_{\text{conv}} = \frac{1}{h} = \frac{1}{12.54 \text{ W/m}^2 \cdot ^\circ\text{C}} = 0.0797 \text{ m}^2 \cdot ^\circ\text{C/W}$$

Therefore, the effect of 0.25 mm thick mineral deposit on heat transfer is negligible.

Next we determine the exit temperature of air

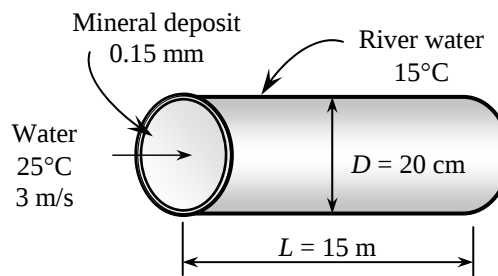
$$T_e = T_s - (T_s - T_i) e^{-\frac{hA}{\dot{m}c_p}} = 15 - (15 - 25) e^{-\frac{(12.54)(9.425)}{(0.1135)(1007)}} = 18.6^\circ\text{C}$$

The friction factor, pressure drop, and the fan power required to overcome this pressure drop can be determined for the case of fully developed turbulent flow in smooth pipes to be

$$f = (0.790 \ln \text{Re} - 1.64)^{-2} = [0.790 \ln(3.958 \times 10^4) - 1.64]^{-2} = 0.02212$$

$$\Delta P = f \frac{L}{D} \frac{\rho V_{\text{avg}}^2}{2} = 0.02212 \frac{15 \text{ m}}{0.2 \text{ m}} \frac{(1.204 \text{ kg/m}^3)(3 \text{ m/s})^2}{2} \left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2} \right) \left(\frac{1 \text{ Pa}}{1 \text{ N/m}^2} \right) = 8.988 \text{ Pa}$$

$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V} \Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.09425 \text{ m}^3/\text{s})(8.988 \text{ Pa})}{0.55} = \left(\frac{1 \text{ W}}{1 \text{ Pa} \cdot \text{m}^3/\text{s}} \right) = 1.54 \text{ W}$$



13-76E The exhaust gases of an automotive engine enter a steel exhaust pipe. The velocity of exhaust gases at the inlet and the temperature of exhaust gases at the exit are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the pipe are smooth. 3 The thermal resistance of the pipe is negligible. 4 Exhaust gases have the properties of air, which is an ideal gas with constant properties.

Properties We take the bulk mean temperature for exhaust gases to be 700°F since the mean temperature of gases at the inlet will drop somewhat as a result of heat loss through the exhaust pipe whose surface is at a lower temperature. The properties of air at this temperature and 1 atm pressure are (Table A-22E)

$$\rho = 0.03421 \text{ lbm/ft}^3 \quad c_p = 0.2535 \text{ Btu/lbm} \cdot ^\circ\text{F}$$

$$k = 0.0280 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} \quad \text{Pr} = 0.694$$

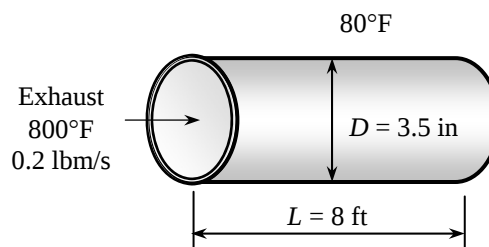
$$\nu = 6.225 \times 10^{-4} \text{ ft}^2/\text{s}$$

Noting that 1 atm = 14.7 psia, the pressure in atm is

$$P = (15.5 \text{ psia})/(14.7 \text{ psia}) = 1.054 \text{ atm. Then,}$$

$$\rho = (0.03421 \text{ lbm/ft}^3)(1.054) = 0.03606 \text{ lbm/ft}^3$$

$$\nu = (6.225 \times 10^{-4} \text{ ft}^2/\text{s})/(1.054) = 5.906 \times 10^{-4} \text{ ft}^2/\text{s}$$



Analysis (a) The velocity of exhaust gases at the inlet of the exhaust pipe is

$$\dot{m} = \rho V_{\text{avg}} A_c \longrightarrow V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{0.2 \text{ lbm/s}}{(0.03606 \text{ lbm/ft}^3)(\pi(3.5/12 \text{ ft})^2/4)} = \mathbf{83.01 \text{ ft/s}}$$

(b) The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(83.01 \text{ ft/s})(3.5/12 \text{ ft})}{5.906 \times 10^{-4} \text{ ft}^2/\text{s}} = 40,990$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(3.5/12 \text{ ft}) = 2.917 \text{ ft}$$

which are shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$\text{Nu} = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(40,990)^{0.8} (0.694)^{0.3} = 101.0$$

$$\text{and} \quad h_i = h = \frac{k}{D_h} \text{Nu} = \frac{0.0280 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F}}{(3.5/12 \text{ ft})} (101.0) = 9.70 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

$$A_s = \pi DL = \pi(3.5/12 \text{ ft})(8 \text{ ft}) = 7.33 \text{ ft}^2$$

In steady operation, heat transfer from exhaust gases to the duct must be equal to the heat transfer from the duct to the surroundings, which must be equal to the energy loss of the exhaust gases in the pipe. That is,

$$\dot{Q} = \dot{Q}_{\text{internal}} = \dot{Q}_{\text{external}} = \Delta \dot{E}_{\text{exhaust gases}}$$

Assuming the duct to be at an average temperature of T_s , the quantities above can be expressed as

$$\dot{Q}_{\text{internal}}: \quad \dot{Q} = h_i A_s \Delta T_{\text{ln}} = h_i A_s \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \rightarrow \dot{Q} = (9.70 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(7.33 \text{ ft}^2) \frac{T_e - 800^\circ\text{F}}{\ln\left(\frac{T_s - T_e}{T_s - 800}\right)}$$

$$\dot{Q}_{\text{external}}: \quad \dot{Q} = h_o A_s (T_s - T_o) \rightarrow \dot{Q} = (3 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(7.33 \text{ ft}^2)(T_s - 80^\circ\text{F})$$

$$\Delta \dot{E}_{\text{exhaust gases}}: \quad \dot{Q} = \dot{m} c_p (T_e - T_i) \rightarrow \dot{Q} = (0.2 \times 3600 \text{ lbm/h})(0.2535 \text{ Btu/lbm} \cdot ^\circ\text{F})(800 - T_e)^\circ\text{F}$$

This is a system of three equations with three unknowns whose solution is

$$\dot{Q} = 11,528 \text{ Btu/h}, T_e = \mathbf{736.8^\circ\text{F}}, \text{ and } T_s = 604.2^\circ\text{F}$$

13-77 Hot water enters a cast iron pipe whose outer surface is exposed to cold air with a specified heat transfer coefficient. The rate of heat loss from the water and the exit temperature of the water are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the pipe are smooth.

Properties We assume the water temperature not to drop significantly since the pipe is not very long. We will check this assumption later. The properties of water at 90°C are (Table A-15)

$$\rho = 965.3 \text{ kg/m}^3; \quad k = 0.675 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = \mu / \rho = 0.326 \times 10^{-6} \text{ m}^2/\text{s}; \quad c_p = 4206 \text{ J/kg} \cdot ^\circ\text{C}$$

$$\text{Pr} = 1.96$$

Analysis (a) The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = (965.3 \text{ kg/m}^3) \frac{\pi (0.04 \text{ m})^2}{4} (1.2 \text{ m/s}) = 1.456 \text{ kg/s}$$

The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(1.2 \text{ m/s})(0.04 \text{ m})}{0.326 \times 10^{-6} \text{ m}^2/\text{s}} = 147,240$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.04 \text{ m}) = 0.4 \text{ m}$$

which are much shorter than the total length of the pipe. Therefore, we can assume fully developed turbulent flow in the entire pipe. The friction factor corresponding to $\text{Re} = 147,240$ and $\varepsilon/D = (0.026 \text{ cm})/(4 \text{ cm}) = 0.0065$ is determined from the Moody chart to be $f = 0.032$. Then the Nusselt number becomes

$$\text{Nu} = \frac{hD_h}{k} = 0.125 f \text{ Re Pr}^{1/3} = 0.125 \times 0.032 \times 147,240 \times 1.96^{1/3} = 737.1$$

$$\text{and} \quad h_i = h = \frac{k}{D_h} \text{Nu} = \frac{0.675 \text{ W/m} \cdot ^\circ\text{C}}{0.04 \text{ m}} (737.1) = 12,440 \text{ W/m}^2 \cdot ^\circ\text{C}$$

which is much greater than the convection heat transfer coefficient of $12 \text{ W/m}^2 \cdot ^\circ\text{C}$. Therefore, the convection thermal resistance inside the pipe is negligible, and thus the inner surface temperature of the pipe can be taken to be equal to the water temperature. Also, we expect the pipe to be nearly isothermal since it is made of thin metal (we check this later). Then the rate of heat loss from the pipe will be the sum of the convection and radiation from the outer surface at a temperature of 90°C, and is determined to be

$$A_o = \pi D_o L = \pi (0.046 \text{ m})(15 \text{ m}) = 2.168 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = h_o A_o (T_s - T_{\text{surr}}) = (12 \text{ W/m}^2 \cdot ^\circ\text{C})(2.168 \text{ m}^2)(90 - 10)^\circ\text{C} = 2081 \text{ W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon A_o \sigma (T_s^4 - T_{\text{surr}}^4)$$

$$= (0.7)(2.168 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(90 + 273 \text{ K})^4 - (10 + 273 \text{ K})^4] = 942 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 2081 + 942 = \mathbf{3023 \text{ W}}$$

(b) The temperature at which water leaves the basement is

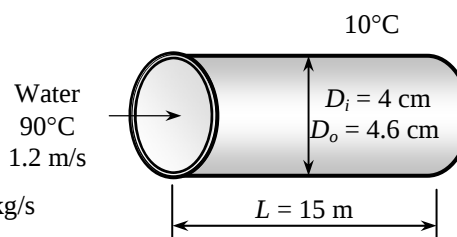
$$\dot{Q} = \dot{m} c_p (T_i - T_e) \longrightarrow T_e = T_i - \frac{\dot{Q}}{\dot{m} c_p} = 90^\circ\text{C} - \frac{3023 \text{ W}}{(1.456 \text{ kg/s})(4206 \text{ J/kg} \cdot ^\circ\text{C})} = \mathbf{89.5^\circ\text{C}}$$

The result justifies our assumption that the temperature drop of water is negligible. Also, the thermal resistance of the pipe and temperature drop across it are

$$R_{\text{pipe}} = \frac{\ln(D_2 / D_1)}{2\pi k L} = \frac{\ln(4.6 / 4)}{4\pi (52 \text{ W/m} \cdot ^\circ\text{C})(15 \text{ m})} = 2.85 \times 10^{-5} \text{ }^\circ\text{C/W}$$

$$\Delta T_{\text{pipe}} = \dot{Q}_{\text{total}} R_{\text{pipe}} = (3023 \text{ W})(2.85 \times 10^{-5} \text{ }^\circ\text{C/W}) = 0.09^\circ\text{C}$$

which justifies our assumption that the temperature drop across the pipe is negligible.



13-78 Hot water enters a copper pipe whose outer surface is exposed to cold air with a specified heat transfer coefficient. The rate of heat loss from the water and the exit temperature of the water are to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the pipe are smooth.

Properties We assume the water temperature not to drop significantly since the pipe is not very long. We will check this assumption later. The properties of water at 90°C are (Table A-22)

$$\rho = 965.3 \text{ kg/m}^3; \quad k = 0.675 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = \mu / \rho = 0.326 \times 10^{-6} \text{ m}^2/\text{s}; \quad c_p = 4206 \text{ J/kg} \cdot ^\circ\text{C}$$

$$\text{Pr} = 1.96$$

Analysis (a) The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = (965.3 \text{ kg/m}^3) \frac{\pi (0.04 \text{ m})^2}{4} (1.2 \text{ m/s}) = 1.456 \text{ kg/s}$$

The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(1.2 \text{ m/s})(0.04 \text{ m})}{0.326 \times 10^{-6} \text{ m}^2/\text{s}} = 147,240$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.04 \text{ m}) = 0.4 \text{ m}$$

which are much shorter than the total length of the pipe. Therefore, we can assume fully developed turbulent flow in the entire pipe. The friction factor corresponding to $\text{Re} = 147,240$ and $\varepsilon/D = (0.026 \text{ cm})/(4 \text{ cm}) = 0.0065$ is determined from the Moody chart to be $f = 0.032$. Then the Nusselt number becomes

$$\text{Nu} = \frac{hD_h}{k} = 0.125 f \text{ Re Pr}^{1/3} = 0.125 \times 0.032 \times 147,240 \times 1.96^{1/3} = 737.1$$

$$\text{and} \quad h_i = h = \frac{k}{D_h} \text{Nu} = \frac{0.675 \text{ W/m} \cdot ^\circ\text{C}}{0.04 \text{ m}} (737.1) = 12,440 \text{ W/m}^2 \cdot ^\circ\text{C}$$

which is much greater than the convection heat transfer coefficient of $12 \text{ W/m}^2 \cdot ^\circ\text{C}$. Therefore, the convection thermal resistance inside the pipe is negligible, and thus the inner surface temperature of the pipe can be taken to be equal to the water temperature. Also, we expect the pipe to be nearly isothermal since it is made of thin metal (we check this later). Then the rate of heat loss from the pipe will be the sum of the convection and radiation from the outer surface at a temperature of 90°C, and is determined to be

$$A_o = \pi D_o L = \pi (0.046 \text{ m})(15 \text{ m}) = 2.168 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = h_o A_o (T_s - T_{\text{surr}}) = (12 \text{ W/m}^2 \cdot ^\circ\text{C})(2.168 \text{ m}^2)(90 - 10)^\circ\text{C} = 2081 \text{ W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon A_o \sigma (T_s^4 - T_{\text{surr}}^4)$$

$$= (0.7)(2.168 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(90 + 273 \text{ K})^4 - (10 + 273 \text{ K})^4] = 942 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 2081 + 942 = \mathbf{3023 \text{ W}}$$

(b) The temperature at which water leaves the basement is

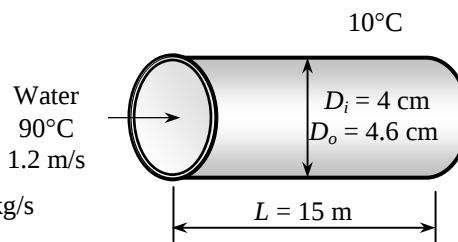
$$\dot{Q} = \dot{m} c_p (T_i - T_e) \longrightarrow T_e = T_i - \frac{\dot{Q}}{\dot{m} c_p} = 90^\circ\text{C} - \frac{3023 \text{ W}}{(1.456 \text{ kg/s})(4206 \text{ J/kg} \cdot ^\circ\text{C})} = \mathbf{89.5^\circ\text{C}}$$

The result justifies our assumption that the temperature drop of water is negligible. Also, the thermal resistance of the pipe and temperature drop across it are

$$R_{\text{pipe}} = \frac{\ln(D_2 / D_1)}{4\pi k L} = \frac{\ln(4.6 / 4)}{2\pi (386 \text{ W/m} \cdot ^\circ\text{C})(15 \text{ m})} = 3.84 \times 10^{-6} \text{ }^\circ\text{C/W}$$

$$\Delta T_{\text{pipe}} = \dot{Q}_{\text{total}} R_{\text{pipe}} = (3023 \text{ W})(3.84 \times 10^{-6} \text{ }^\circ\text{C/W}) = 0.012^\circ\text{C}$$

which justifies our assumption that the temperature drop across the pipe is negligible.



13-79 13-81 Design and Essay Problems

13-81 A computer is cooled by a fan blowing air through the case of the computer. The flow rate of the fan and the diameter of the casing of the fan are to be specified.

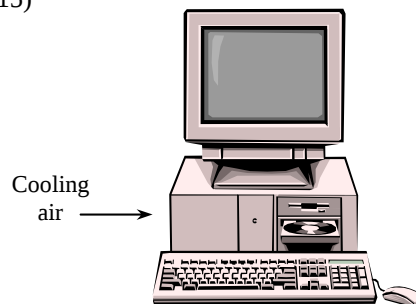
Assumptions 1 Steady flow conditions exist. 2 Heat flux is uniformly distributed. 3 Air is an ideal gas with constant properties.

Properties The relevant properties of air are (Tables A-1 and A-15)

$$c_p = 1007 \text{ J/kg} \cdot ^\circ\text{C}$$

$$R = 0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}$$

Analysis We need to determine the flow rate of air for the worst case scenario. Therefore, we assume the inlet temperature of air to be 50°C , the atmospheric pressure to be 70.12 kPa , and disregard any heat transfer from the outer surfaces of the computer case. The mass flow rate of air required to absorb heat at a rate of 80 W can be determined from



$$\dot{Q} = \dot{m} c_p (T_{out} - T_{in}) \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{80 \text{ J/s}}{(1007 \text{ J/kg} \cdot ^\circ\text{C})(60 - 50)^\circ\text{C}} = 0.007944 \text{ kg/s}$$

In the worst case the exhaust fan will handle air at 60°C . Then the density of air entering the fan and the volume flow rate becomes

$$\rho = \frac{P}{RT} = \frac{70.12 \text{ kPa}}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(60 + 273) \text{ K}} = 0.7337 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.007944 \text{ kg/s}}{0.7337 \text{ kg/m}^3} = 0.01083 \text{ m}^3/\text{s} = \mathbf{0.6497 \text{ m}^3/\text{min}}$$

For an average velocity of 120 m/min , the diameter of the duct in which the fan is installed can be determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.6497 \text{ m}^3/\text{min})}{\pi(120 \text{ m/min})}} = 0.083 \text{ m} = \mathbf{8.3 \text{ cm}}$$



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 14
NATURAL CONVECTION

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Physical Mechanisms of Natural Convection

14-1C Natural convection is the mode of heat transfer that occurs between a solid and a fluid which moves under the influence of natural means. Natural convection differs from forced convection in that fluid motion in natural convection is caused by natural effects such as buoyancy.

14-2C The convection heat transfer coefficient is usually higher in forced convection because of the higher fluid velocities involved.

14-3C The hot boiled egg in a spacecraft will cool faster when the spacecraft is on the ground since there is no gravity in space, and thus there will be no natural convection currents which is due to the buoyancy force.

14-4C The upward force exerted by a fluid on a body completely or partially immersed in it is called the buoyancy or “lifting” force. The buoyancy force is proportional to the density of the medium. Therefore, the buoyancy force is the largest in mercury, followed by in water, air, and the evacuated chamber. Note that in an evacuated chamber there will be no buoyancy force because of absence of any fluid in the medium.

14-5C The buoyancy force is proportional to the density of the medium, and thus is larger in sea water than it is in fresh water. Therefore, the hull of a ship will sink deeper in fresh water because of the smaller buoyancy force acting upwards.

14-6C A spring scale measures the “weight” force acting on it, and the person will weigh less in water because of the upward buoyancy force acting on the person’s body.

14-7C The greater the volume expansion coefficient, the greater the change in density with temperature, the greater the buoyancy force, and thus the greater the natural convection currents.

14-8C There cannot be any natural convection heat transfer in a medium that experiences no change in volume with temperature.

14-9C The lines on an interferometer photograph represent isotherms (constant temperature lines) for a gas, which correspond to the lines of constant density. Closely packed lines on a photograph represent a large temperature gradient.

14-10C The Grashof number represents the ratio of the buoyancy force to the viscous force acting on a fluid. The inertial forces in Reynolds number is replaced by the buoyancy forces in Grashof number.

14-11 The volume expansion coefficient is defined as $\beta = \frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_P$. For an ideal gas, $P = \rho RT$ or

$$\rho = \frac{P}{RT}, \text{ and thus } \beta = -\frac{1}{\rho} \left(\frac{\partial (P/RT)}{\partial T} \right)_P = -\frac{1}{\rho} \left(\frac{-P}{RT^2} \right) = \frac{1}{\rho T} \left(\frac{P}{RT} \right) = \frac{1}{\rho T} (\rho) = \frac{1}{T}$$

Natural Convection over Surfaces

14-12C Rayleigh number is the product of the Grashof and Prandtl numbers.

14-13C A vertical cylinder can be treated as a vertical plate when $D \geq \frac{35L}{Gr^{1/4}}$.

14-14C No, a hot surface will cool slower when facing down since the warmer air in this position cannot rise and escape easily.

14-15C The heat flux will be higher at the bottom of the plate since the thickness of the boundary layer which is a measure of thermal resistance is the lowest there.

14-16 Heat is generated in a horizontal plate while heat is lost from it by convection and radiation. The temperature of the plate when steady operating conditions are reached is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

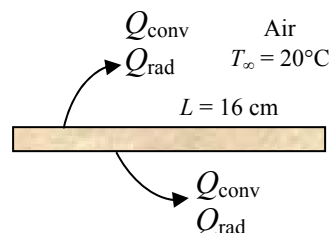
Properties We assume the surface temperature to be 50°C . Then the properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (50 + 20)/2 = 35^\circ\text{C}$ are (Table A-22)

$$k = 0.02625 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.655 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7268$$

$$\beta = \frac{1}{T_f} = \frac{1}{(35 + 273)\text{K}} = 0.003247 \text{ K}^{-1}$$



Analysis The characteristic length in this case is

$$L_c = \frac{A_s}{p} = \frac{(0.16 \text{ m})(0.20 \text{ m})}{2[(0.16 \text{ m}) + (0.20 \text{ m})]} = 0.04444 \text{ m}$$

The Rayleigh number is

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003247 \text{ K}^{-1})(50 - 20 \text{ K})(0.04444 \text{ m})^3}{(1.655 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7268) = 222,593$$

The Nusselt number relation for the top surface of the plate is

$$Nu = 0.54 Ra^{0.25} = 0.54(222,593)^{0.25} = 11.73$$

Then

$$h = \frac{k}{L_c} Nu = \frac{0.02625 \text{ W/m}\cdot^\circ\text{C}}{0.04444 \text{ m}} (11.73) = 6.928 \text{ W/m}^2\cdot^\circ\text{C}$$

and

$$\dot{Q}_{\text{top}} = hA(T_s - T_\infty) = (6.928 \text{ W/m}^2\cdot^\circ\text{C})(0.16 \times 0.20 \text{ m}^2)(T_s - 20)^\circ\text{C} = 0.2217(T_s - 20)$$

The Nusselt number relation for the bottom surface of the plate is

$$Nu = 0.27 Ra^{0.25} = 0.27(222,593)^{0.25} = 5.865$$

Then

$$h = \frac{k}{L_c} Nu = \frac{0.02625 \text{ W/m}\cdot^\circ\text{C}}{0.04444 \text{ m}} (5.865) = 3.464 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q}_{\text{bottom}} = hA(T_s - T_\infty) = (3.464 \text{ W/m}^2\cdot^\circ\text{C})(0.16 \times 0.20 \text{ m}^2)(T_s - 20)^\circ\text{C} = 0.1108(T_s - 20)$$

Considering that radiation heat loss to surroundings occur both from top and bottom surfaces, it may be expressed as

$$\begin{aligned} \dot{Q}_{\text{rad}} &= 2\varepsilon A \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.9)(2)(0.16 \times 0.20 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(T_s + 273 \text{ K})^4 - (17 + 273 \text{ K})^4] \\ &= 3.2659 \times 10^{-9} [(T_s + 273 \text{ K})^4 - (17 + 273 \text{ K})^4] \end{aligned}$$

When the heat lost from the plate equals to the heat generated, the steady operating conditions are reached. The surface temperature in this case can be determined by trial-error or using EES to be

$$\begin{aligned} \dot{Q}_{\text{total}} &= \dot{Q}_{\text{top}} + \dot{Q}_{\text{bottom}} + \dot{Q}_{\text{rad}} \\ 20 \text{ W} &= 0.2217(T_s - 20) + 0.1108(T_s - 20) + 3.2659 \times 10^{-9} [(T_s + 273 \text{ K})^4 - (17 + 273 \text{ K})^4] \\ T_s &= \mathbf{46.8^\circ\text{C}} \end{aligned}$$

14-17 Flue gases are released to atmosphere using a cylindrical stack. The rates of heat transfer from the stack with and without wind cases are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

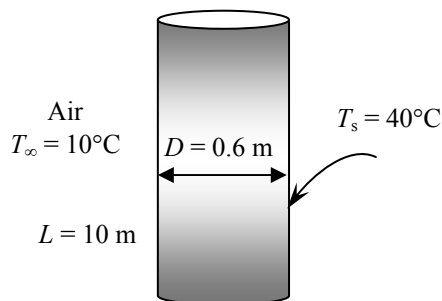
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (40 + 10)/2 = 25^\circ\text{C}$ are (Table A-22)

$$k = 0.02551 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7296$$

$$\beta = \frac{1}{T_f} = \frac{1}{(25 + 273)\text{K}} = 0.003356 \text{ K}^{-1}$$



Analysis (a) When there is no wind heat transfer is by natural convection. The characteristic length in this case is the height of the stack, $L_c = L = 10 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003356 \text{ K}^{-1})(40 - 10 \text{ K})(10 \text{ m})^3}{(1.562 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7296) = 2.953 \times 10^{12}$$

We can treat this vertical cylinder as a vertical plate since

$$\frac{35L}{Gr^{1/4}} = \frac{35(10)}{(2.953 \times 10^{12} / 0.7296)^{1/4}} = 0.246 < 0.6 \quad \text{and thus } D \geq \frac{35L}{Gr^{1/4}}$$

The Nusselt number is determined from

$$Nu = 0.1Ra^{1/3} = 0.1(2.953 \times 10^{12})^{1/3} = 1435 \quad (\text{from Table 14-1})$$

Then

$$h = \frac{k}{L_c} Nu = \frac{0.02551 \text{ W/m}\cdot^\circ\text{C}}{10 \text{ m}} (1435) = 3.660 \text{ W/m}^2\cdot^\circ\text{C}$$

and

$$\dot{Q} = hA(T_s - T_\infty) = (3.660 \text{ W/m}^2\cdot^\circ\text{C})(\pi \times 0.6 \times 10 \text{ m}^2)(40 - 10)^\circ\text{C} = \mathbf{2070 \text{ W}}$$

(b) When the stack is exposed to 20 km/h winds, the heat transfer will be by forced convection. We have flow of air over a cylinder and the heat transfer rate is determined as follows:

$$\text{Re} = \frac{VD}{\nu} = \frac{(20 \times 1000 / 3600 \text{ m/s})(0.6 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 213,400$$

$$\text{Nu} = 0.027 \text{Re}^{0.805} \text{Pr}^{1/3} = 0.027(213,400)^{0.805} (0.7296)^{1/3} = 473.9 \quad (\text{from Table 19-2})$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.02551 \text{ W/m}\cdot^\circ\text{C}}{0.6 \text{ m}} (473.9) = 20.15 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q} = hA(T_s - T_\infty) = (20.15 \text{ W/m}^2\cdot^\circ\text{C})(\pi \times 0.6 \times 10 \text{ m}^2)(40 - 10)^\circ\text{C} = \mathbf{11,390 \text{ W}}$$

Discussion There is more than five-fold increase in heat transfer due to winds.

14-18 Heat generated by the electrical resistance of a bare cable is dissipated to the surrounding air. The surface temperature of the cable is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The temperature of the surface of the cable is constant.

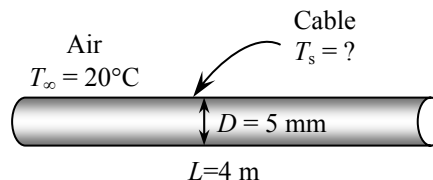
Properties We assume the surface temperature to be 100°C. Then the properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (100 + 20)/2 = 60^\circ\text{C}$ are (Table A-22)

$$k = 0.02808 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.896 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7202$$

$$\beta = \frac{1}{T_f} = \frac{1}{(60 + 273)\text{K}} = 0.003003 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the outer diameter of the pipe, $L_c = D = 0.005 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003003 \text{ K}^{-1})(100 - 20 \text{ K})(0.005 \text{ m})^3}{(1.896 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7202) = 590.2$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (590.2)^{1/6}}{\left[1 + (0.559 / 0.7202)^{9/16} \right]^{8/27}} \right\}^2 = 2.346$$

$$h = \frac{k}{D} Nu = \frac{0.02808 \text{ W/m}\cdot^\circ\text{C}}{0.005 \text{ m}} (2.346) = 13.17 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.005 \text{ m})(4 \text{ m}) = 0.06283 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$(60 \text{ V})(1.5 \text{ A}) = (13.17 \text{ W/m}^2\cdot^\circ\text{C})(0.06283 \text{ m}^2)(T_s - 20)^\circ\text{C}$$

$$T_s = 128.8^\circ\text{C}$$

which is not close to the assumed value of 100°C. Repeating calculations for an assumed surface temperature of 120°C, $[T_f = (T_s + T_\infty)/2 = (120 + 20)/2 = 70^\circ\text{C}]$

$$k = 0.02881 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.995 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7177$$

$$\beta = \frac{1}{T_f} = \frac{1}{(70 + 273)\text{K}} = 0.002915 \text{ K}^{-1}$$

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002915 \text{ K}^{-1})(120 - 20 \text{ K})(0.005 \text{ m})^3}{(1.995 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7177) = 644.6$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (644.6)^{1/6}}{\left[1 + (0.559 / 0.7177)^{9/16} \right]^{8/27}} \right\}^2 = 2.387$$

$$h = \frac{k}{D} Nu = \frac{0.02881 \text{ W/m}\cdot^\circ\text{C}}{0.005 \text{ m}} (2.387) = 13.76 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$(60 \text{ V})(1.5 \text{ A}) = (13.76 \text{ W/m}^2\cdot^\circ\text{C})(0.06283 \text{ m}^2)(T_s - 20)^\circ\text{C}$$

$$T_s = 124.1^\circ\text{C}$$

which is sufficiently close to the assumed value of 120°C.

14-19 A horizontal hot water pipe passes through a large room. The rate of heat loss from the pipe by natural convection and radiation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The temperature of the outer surface of the pipe is constant.

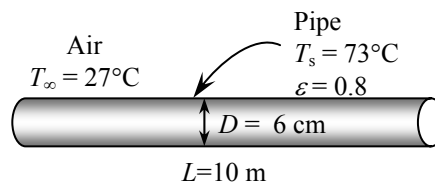
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (73 + 27)/2 = 50^\circ\text{C}$ are (Table A-22)

$$k = 0.02735 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7228$$

$$\beta = \frac{1}{T_f} = \frac{1}{(50 + 273)\text{K}} = 0.003096 \text{ K}^{-1}$$



Analysis (a) The characteristic length in this case is the outer diameter of the pipe, $L_c = D = 0.06 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003096 \text{ K}^{-1})(73 - 27 \text{ K})(0.06 \text{ m})^3}{(1.798 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7228) = 6.747 \times 10^5$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (6.747 \times 10^5)^{1/6}}{\left[1 + (0.559 / 0.7228)^{9/16} \right]^{8/27}} \right\}^2 = 13.05$$

$$h = \frac{k}{D} Nu = \frac{0.02735 \text{ W/m}\cdot^\circ\text{C}}{0.06 \text{ m}} (13.05) = 5.950 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.06 \text{ m})(10 \text{ m}) = 1.885 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.950 \text{ W/m}^2\cdot^\circ\text{C})(1.885 \text{ m}^2)(73 - 27)^\circ\text{C} = \mathbf{516 \text{ W}}$$

(b) The radiation heat loss from the pipe is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.8)(1.885 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) \left[(73 + 273 \text{ K})^4 - (27 + 273 \text{ K})^4 \right] = \mathbf{533 \text{ W}} \end{aligned}$$

14-20 A power transistor mounted on the wall dissipates 0.18 W. The surface temperature of the transistor is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Any heat transfer from the base surface is disregarded. 4 The local atmospheric pressure is 1 atm. 5 Air properties are evaluated at 100°C.

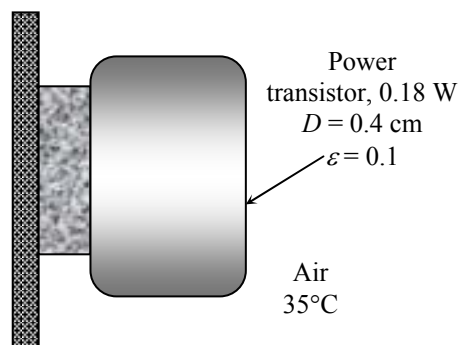
Properties The properties of air at 1 atm and the given film temperature of 100°C are (Table A-22)

$$k = 0.03095 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 2.306 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7111$$

$$\beta = \frac{1}{T_f} = \frac{1}{(100 + 273) \text{ K}} = 0.00268 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 165°C for the evaluation of h . This is the surface temperature that will give a film temperature of 100°C. We will check the accuracy of this guess later and repeat the calculations if necessary.

The transistor loses heat through its cylindrical surface as well as its top surface. For convenience, we take the heat transfer coefficient at the top surface of the transistor to be the same as that of its side surface. (The alternative is to treat the top surface as a vertical plate, but this will double the amount of calculations without providing much improvement in accuracy since the area of the top surface is much smaller and it is circular in shape instead of being rectangular). The characteristic length in this case is the outer diameter of the transistor, $L_c = D = 0.004 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00268 \text{ K}^{-1})(165 - 35 \text{ K})(0.004 \text{ m})^3}{(2.306 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7111) = 292.6$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (292.6)^{1/6}}{\left[1 + (0.559 / 0.7111)^{9/16} \right]^{8/27}} \right\}^2 = 2.039$$

$$h = \frac{k}{D} Nu = \frac{0.03095 \text{ W/m} \cdot ^\circ\text{C}}{0.004 \text{ m}} (2.039) = 15.78 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$A_s = \pi DL + \pi D^2 / 4 = \pi (0.004 \text{ m})(0.0045 \text{ m}) + \pi (0.004 \text{ m})^2 / 4 = 0.0000691 \text{ m}^2$$

and

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ 0.18 \text{ W} &= (15.78 \text{ W/m}^2 \cdot ^\circ\text{C})(0.0000691 \text{ m}^2)(T_s - 35)^\circ\text{C} \\ &\quad + (0.1)(0.0000691 \text{ m}^2)(5.67 \times 10^{-8}) \left[(T_s + 273)^4 - (25 + 273 \text{ K})^4 \right] \\ \longrightarrow T_s &= 187^\circ\text{C} \end{aligned}$$

which is relatively close to the assumed value of 165°C. To improve the accuracy of the result, we repeat the Rayleigh number calculation at new surface temperature of 187°C and determine the surface temperature to be

$$T_s = 183^\circ\text{C}$$

Discussion We evaluated the air properties again at 100°C when repeating the calculation at the new surface temperature. It can be shown that the effect of this on the calculated surface temperature is less than 1°C.

14-21 EES Prob. 14-20 is reconsidered. The effect of ambient temperature on the surface temperature of the transistor is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

Q_dot=0.18 [W]
 T_infinity=35 [C]
 L=0.0045 [m]
 D=0.004 [m]
 epsilon=0.1
 T_surr=T_infinity-10 [C]

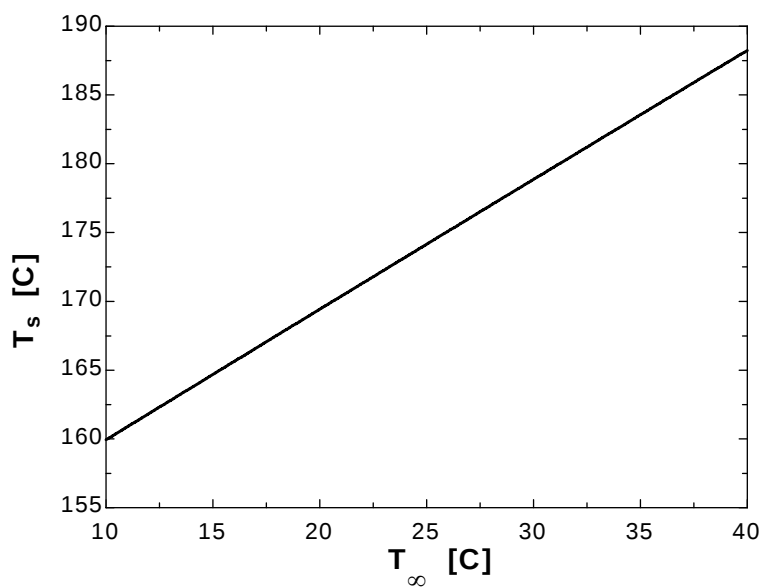
"PROPERTIES"

Fluid\$='air'
 k=Conductivity(Fluid\$, T=T_film)
 Pr=Prandtl(Fluid\$, T=T_film)
 rho=Density(Fluid\$, T=T_film, P=101.3)
 mu=Viscosity(Fluid\$, T=T_film)
 nu=mu/rho
 beta=1/(T_film+273)
 T_film=1/2*(T_s+T_infinity)
 sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzmann constant"
 g=9.807 [m/s^2] "gravitational acceleration"

"ANALYSIS"

delta=D
 Ra=(g*beta*(T_s-T_infinity)*delta^3)/nu^2*Pr
 Nusselt=(0.6+(0.387*Ra^(1/6))/(1+(0.559/Pr)^(9/16)))^(8/27)
 h=k/delta*Nusselt
 A=pi*D*L+pi*D^2/4
 Q_dot=h*A*(T_s-T_infinity)+epsilon*A*sigma*((T_s+273)^4-(T_surr+273)^4)

T _∞ [C]	T _s [C]
10	159.9
12	161.8
14	163.7
16	165.6
18	167.5
20	169.4
22	171.3
24	173.2
26	175.1
28	177
30	178.9
32	180.7
34	182.6
36	184.5
38	186.4
40	188.2



14-22E A hot plate with an insulated back is considered. The rate of heat loss by natural convection is to be determined for different orientations.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

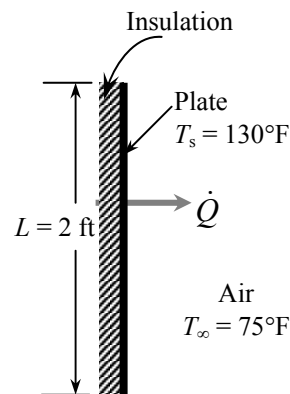
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (130 + 75)/2 = 102.5^\circ\text{F}$ are (Table A-22)

$$k = 0.01535 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1823 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7256$$

$$\beta = \frac{1}{T_f} = \frac{1}{(102.5 + 460)\text{R}} = 0.001778 \text{ R}^{-1}$$



Analysis (a) When the plate is vertical, the characteristic length is the height of the plate. $L_c = L = 2 \text{ ft}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001778 \text{ R}^{-1})(130 - 75 \text{ R})(2 \text{ ft})^3}{(0.1823 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7256) = 5.503 \times 10^8$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(5.503 \times 10^8)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7256} \right)^{9/16} \right]^{8/27}} \right\}^2 = 102.6$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.01535 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{2 \text{ ft}} (102.6) = 0.7869 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_s = L^2 = (2 \text{ ft})^2 = 4 \text{ ft}^2$$

and $\dot{Q} = hA_s(T_s - T_\infty) = (0.7869 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(4 \text{ ft}^2)(130 - 75)^\circ\text{C} = \mathbf{173.1 \text{ Btu/h}}$

(b) When the plate is horizontal with hot surface facing up, the characteristic length is determined from

$$L_s = \frac{A_s}{P} = \frac{L^2}{4L} = \frac{L}{4} = \frac{2 \text{ ft}}{4} = 0.5 \text{ ft}.$$

Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001778 \text{ R}^{-1})(130 - 75 \text{ R})(0.5 \text{ ft})^3}{(0.1823 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7256) = 8.598 \times 10^6$$

$$\text{Nu} = 0.54\text{Ra}^{1/4} = 0.54(8.598 \times 10^6)^{1/4} = 29.24$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.01535 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.5 \text{ ft}} (29.24) = 0.8975 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

and $\dot{Q} = hA_s(T_s - T_\infty) = (0.8975 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(4 \text{ ft}^2)(130 - 75)^\circ\text{C} = \mathbf{197.4 \text{ Btu/h}}$

(c) When the plate is horizontal with hot surface facing down, the characteristic length is again $\delta = 0.5 \text{ ft}$ and the Rayleigh number is $\text{Ra} = 8.598 \times 10^6$. Then,

$$\text{Nu} = 0.27\text{Ra}^{1/4} = 0.27(8.598 \times 10^6)^{1/4} = 14.62$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.01535 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.5 \text{ ft}} (14.62) = 0.4487 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

and $\dot{Q} = hA_s(T_s - T_\infty) = (0.4487 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(4 \text{ ft}^2)(130 - 75)^\circ\text{C} = \mathbf{98.7 \text{ Btu/h}}$

14-23E EES Prob. 14-22E is reconsidered. The rate of natural convection heat transfer for different orientations of the plate as a function of the plate temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

L=2 [ft]
T_infinity=75 [F]
T_s=130 [F]

"PROPERTIES"

Fluid\$='air'
k=Conductivity(Fluid\$, T=T_film)
Pr=Prandtl(Fluid\$, T=T_film)
rho=Density(Fluid\$, T=T_film, P=14.7)
mu=Viscosity(Fluid\$, T=T_film)*Convert(lbm/ft-h, lbm/ft-s)
nu=mu/rho
beta=1/(T_film+460)
T_film=1/2*(T_s+T_infinity)
g=32.2 [ft/s^2]

"ANALYSIS"

"(a), plate is vertical"

delta_a=L
Ra_a=(g*beta*(T_s-T_infinity)*delta_a^3)/nu^2*Pr
Nusselt_a=0.59*Ra_a^0.25
h_a=k/delta_a*Nusselt_a
A=L^2
Q_dot_a=h_a*A*(T_s-T_infinity)

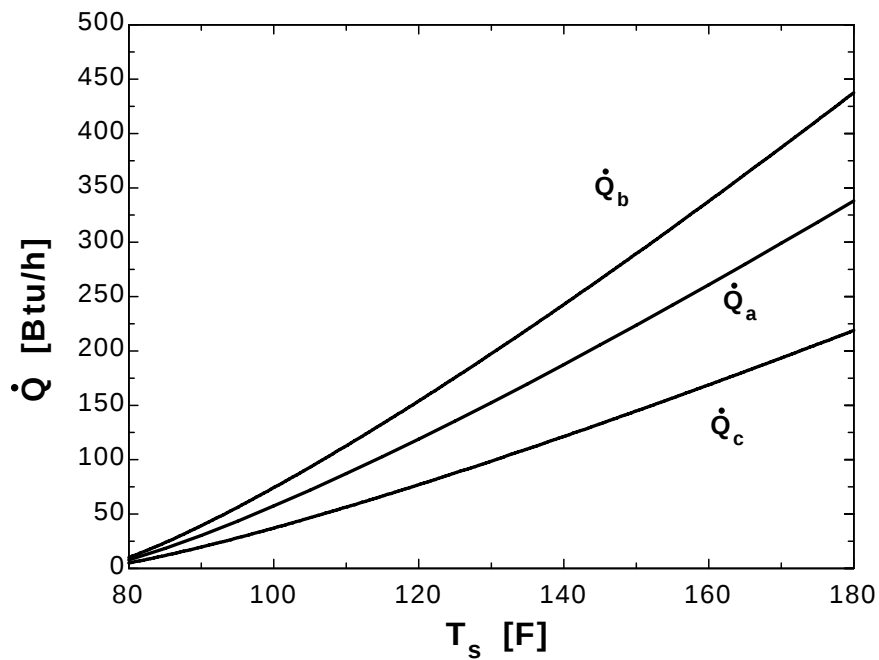
"(b), plate is horizontal with hot surface facing up"

delta_b=A/p
p=4*L
Ra_b=(g*beta*(T_s-T_infinity)*delta_b^3)/nu^2*Pr
Nusselt_b=0.54*Ra_b^0.25
h_b=k/delta_b*Nusselt_b
Q_dot_b=h_b*A*(T_s-T_infinity)

"(c), plate is horizontal with hot surface facing down"

delta_c=delta_b
Ra_c=Ra_b
Nusselt_c=0.27*Ra_c^0.25
h_c=k/delta_c*Nusselt_c
Q_dot_c=h_c*A*(T_s-T_infinity)

T_s [F]	Q_a [Btu/h]	Q_b [Btu/h]	Q_c [Btu/h]
80	7.714	9.985	4.993
85	18.32	23.72	11.86
90	30.38	39.32	19.66
95	43.47	56.26	28.13
100	57.37	74.26	37.13
105	71.97	93.15	46.58
110	87.15	112.8	56.4
115	102.8	133.1	66.56
120	119	154	77.02
125	135.6	175.5	87.75
130	152.5	197.4	98.72
135	169.9	219.9	109.9
140	187.5	242.7	121.3
145	205.4	265.9	132.9
150	223.7	289.5	144.7
155	242.1	313.4	156.7
160	260.9	337.7	168.8
165	279.9	362.2	181.1
170	299.1	387.1	193.5
175	318.5	412.2	206.1
180	338.1	437.6	218.8



14-24 A cylindrical resistance heater is placed horizontally in a fluid. The outer surface temperature of the resistance wire is to be determined for two different fluids.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Any heat transfer by radiation is ignored. 5 Properties are evaluated at 500°C for air and 40°C for water.

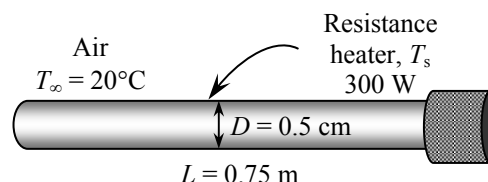
Properties The properties of air at 1 atm and 500°C are (Table A-22)

$$k = 0.05572 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 7.804 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.6986,$$

$$\beta = \frac{1}{T_f} = \frac{1}{(500 + 273)\text{K}} = 0.001294 \text{ K}^{-1}$$



The properties of water at 40°C are (Table A-15)

$$k = 0.631 \text{ W/m} \cdot ^\circ\text{C}, \quad \nu = \mu / \rho = 0.6582 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 4.32, \quad \beta = 0.000377 \text{ K}^{-1}$$

Analysis (a) The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 1200°C for the calculation of h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the wire, $L_c = D = 0.005 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.001294 \text{ K}^{-1})(1200 - 20)^\circ\text{C}(0.005 \text{ m})^3}{(7.804 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.6986) = 214.7$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (214.7)^{1/6}}{\left[1 + (0.559 / 0.6986)^{9/16} \right]^{8/27}} \right\}^2 = 1.919$$

$$h = \frac{k}{D} Nu = \frac{0.05572 \text{ W/m} \cdot ^\circ\text{C}}{0.005 \text{ m}} (1.919) = 21.38 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.005 \text{ m})(0.75 \text{ m}) = 0.01178 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow 300 \text{ W} = (21.38 \text{ W/m}^2 \cdot ^\circ\text{C})(0.01178 \text{ m}^2)(T_s - 20)^\circ\text{C} \rightarrow T_s = \mathbf{1211^\circ\text{C}}$$

which is sufficiently close to the assumed value of 1200°C used in the evaluation of h , and thus it is not necessary to repeat calculations.

(b) For the case of water, we “guess” the surface temperature to be 40°C. The characteristic length in this case is the outer diameter of the wire, $L_c = D = 0.005 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.000377 \text{ K}^{-1})(40 - 20)^\circ\text{C}(0.005 \text{ m})^3}{(0.6582 \times 10^{-6} \text{ m}^2/\text{s})^2} (4.32) = 92,197$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (92,197)^{1/6}}{\left[1 + (0.559 / 4.32)^{9/16} \right]^{8/27}} \right\}^2 = 8.986$$

$$h = \frac{k}{D} Nu = \frac{0.631 \text{ W/m} \cdot ^\circ\text{C}}{0.005 \text{ m}} (8.986) = 1134 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\text{and } \dot{Q} = hA_s(T_s - T_\infty) \rightarrow 300 \text{ W} = (1134 \text{ W/m}^2 \cdot ^\circ\text{C})(0.01178 \text{ m}^2)(T_s - 20)^\circ\text{C} \rightarrow T_s = \mathbf{42.5^\circ\text{C}}$$

which is sufficiently close to the assumed value of 40°C in the evaluation of the properties and h . The film temperature in this case is $(T_s + T_\infty)/2 = (42.5 + 20)/2 = 31.3^\circ\text{C}$, which is close to the value of 40°C used in the evaluation of the properties.

14-25 Water is boiling in a pan that is placed on top of a stove. The rate of heat loss from the cylindrical side surface of the pan by natural convection and radiation and the ratio of heat lost from the side surfaces of the pan to that by the evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

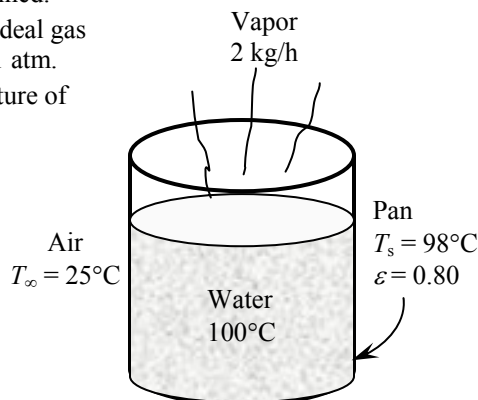
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (98 + 25)/2 = 61.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02819 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.910 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7198$$

$$\beta = \frac{1}{T_f} = \frac{1}{(61.5 + 273)\text{K}} = 0.00299 \text{ K}^{-1}$$



Analysis (a) The characteristic length in this case is the height of the pan, $L_c = L = 0.12 \text{ m}$. Then

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00299 \text{ K}^{-1})(98 - 25 \text{ K})(0.12 \text{ m})^3}{(1.910 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7198) = 7.299 \times 10^6$$

We can treat this vertical cylinder as a vertical plate since

$$\frac{35L}{\text{Gr}^{1/4}} = \frac{35(0.12)}{(7.299 \times 10^6 / 0.7198)^{1/4}} = 0.07443 < 0.25 \quad \text{and thus } D \geq \frac{35L}{\text{Gr}^{1/4}}$$

Therefore,

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(7.299 \times 10^6)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7198} \right)^{9/16} \right]^{8/27}} \right\}^2 = 28.60$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02819 \text{ W/m}\cdot^\circ\text{C}}{0.12 \text{ m}} (28.60) = 6.720 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.25 \text{ m})(0.12 \text{ m}) = 0.09425 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (6.720 \text{ W/m}^2\cdot^\circ\text{C})(0.09425 \text{ m}^2)(98 - 25)^\circ\text{C} = \mathbf{46.2 \text{ W}}$$

(b) The radiation heat loss from the pan is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.80)(0.09425 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(98 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] = \mathbf{47.3 \text{ W}} \end{aligned}$$

(c) The heat loss by the evaporation of water is

$$\dot{Q} = \dot{m} h_{fg} = (1.5 / 3600 \text{ kg/s})(2257 \text{ kJ/kg}) = 0.9404 \text{ kW} = 940 \text{ W}$$

Then the ratio of the heat lost from the side surfaces of the pan to that by the evaporation of water then becomes

$$f = \frac{46.2 + 47.3}{940} = 0.099 = \mathbf{9.9\%}$$

14-26 Water is boiling in a pan that is placed on top of a stove. The rate of heat loss from the cylindrical side surface of the pan by natural convection and radiation and the ratio of heat lost from the side surfaces of the pan to that by the evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

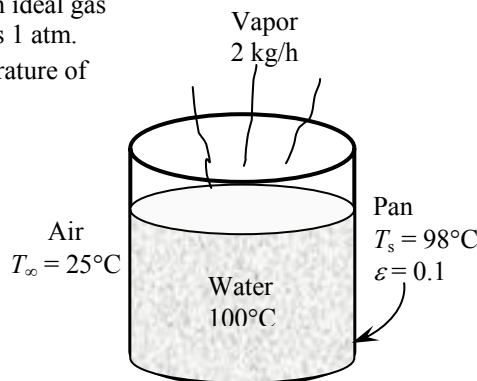
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (98 + 25)/2 = 61.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02819 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.910 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7198$$

$$\beta = \frac{1}{T_f} = \frac{1}{(61.5 + 273)\text{K}} = 0.00299 \text{ K}^{-1}$$



Analysis (a) The characteristic length in this case is the height of the pan, $L_c = L = 0.12 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00299 \text{ K}^{-1})(98 - 25 \text{ K})(0.12 \text{ m})^3}{(1.910 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7198) = 7.299 \times 10^6$$

We can treat this vertical cylinder as a vertical plate since

$$\frac{35L}{\text{Gr}^{1/4}} = \frac{35(0.12)}{(7.299 \times 10^6 / 0.7198)^{1/4}} = 0.07443 < 0.25 \quad \text{and thus } D \geq \frac{35L}{\text{Gr}^{1/4}}$$

Therefore,

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(7.299 \times 10^6)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7198} \right)^{9/16} \right]^{8/27}} \right\}^2 = 28.60$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02819 \text{ W/m}\cdot^\circ\text{C}}{0.12 \text{ m}} (28.60) = 6.720 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.25 \text{ m})(0.12 \text{ m}) = 0.09425 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (6.720 \text{ W/m}^2\cdot^\circ\text{C})(0.09425 \text{ m}^2)(98 - 25)^\circ\text{C} = \mathbf{46.2 \text{ W}}$$

(b) The radiation heat loss from the pan is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.10)(0.09425 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(98 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] = \mathbf{5.9 \text{ W}} \end{aligned}$$

(c) The heat loss by the evaporation of water is

$$\dot{Q} = \dot{m} h_{fg} = (1.5 / 3600 \text{ kg/s})(2257 \text{ kJ/kg}) = 0.9404 \text{ kW} = 940 \text{ W}$$

Then the ratio of the heat lost from the side surfaces of the pan to that by the evaporation of water then becomes

$$f = \frac{46.2 + 5.9}{940} = 0.055 = \mathbf{5.5\%}$$

14-27 Some cans move slowly in a hot water container made of sheet metal. The rate of heat loss from the four side surfaces of the container and the annual cost of those heat losses are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 3 Heat loss from the top surface is disregarded.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (55 + 20)/2 = 37.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02644 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.678 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7262$$

$$\beta = \frac{1}{T_f} = \frac{1}{(37.5 + 273)\text{K}} = 0.003221 \text{ K}^{-1}$$

Analysis The characteristic length in this case is the height of the bath, $L_c = L = 0.5 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003221 \text{ K}^{-1})(55 - 20 \text{ K})(0.5 \text{ m})^3}{(1.678 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7262) = 3.565 \times 10^8$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(3.565 \times 10^8)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7261} \right)^{9/16} \right]^{8/27}} \right\}^2 = 89.84$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02644 \text{ W/m}\cdot^\circ\text{C}}{0.5 \text{ m}} (89.84) = 4.75 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = 2[(0.5 \text{ m})(1 \text{ m}) + (0.5 \text{ m})(3.5 \text{ m})] = 4.5 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (4.75 \text{ W/m}^2\cdot^\circ\text{C})(4.5 \text{ m}^2)(55 - 20)^\circ\text{C} = 748.1 \text{ W}$$

The radiation heat loss is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.7)(4.5 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(55 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4] = 750.9 \text{ W} \end{aligned}$$

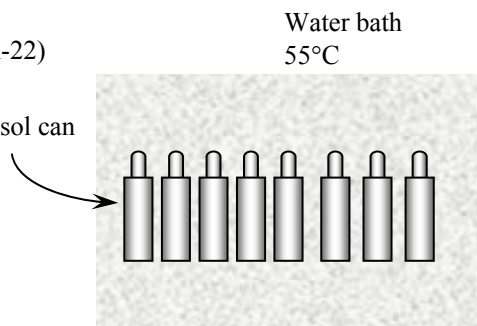
Then the total rate of heat loss becomes

$$\dot{Q}_{total} = \dot{Q}_{\text{natural convection}} + \dot{Q}_{rad} = 748.1 + 750.9 = \mathbf{1499 \text{ W}}$$

The amount and cost of the heat loss during one year is

$$Q_{total} = \dot{Q}_{total} \Delta t = (1.499 \text{ kW})(8760 \text{ h}) = 13,131 \text{ kWh}$$

$$\text{Cost} = (13,131 \text{ kWh})(\$0.085 / \text{kWh}) = \mathbf{\$1116}$$



14-28 Some cans move slowly in a hot water container made of sheet metal. It is proposed to insulate the side and bottom surfaces of the container for \$350. The simple payback period of the insulation to pay for itself from the energy it saves is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 3 Heat loss from the top surface is disregarded.

Properties Insulation will drop the outer surface temperature to a value close to the ambient temperature. The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature, which is unknown. We assume the surface temperature to be 26°C. The properties of air at the anticipated film temperature of $(26+20)/2=23^\circ\text{C}$ are (Table A-22)

$$k = 0.02536 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.543 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7301$$

$$\beta = \frac{1}{T_f} = \frac{1}{(23 + 273)\text{K}} = 0.00338 \text{ K}^{-1}$$

Analysis We start the solution process by “guessing” the outer surface temperature to be 26°C. We will check the accuracy of this guess later and repeat the calculations if necessary with a better guess based on the results obtained. The characteristic length in this case is the height of the tank, $L_c = L = 0.5 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00338 \text{ K}^{-1})(26 - 20 \text{ K})(0.5 \text{ m})^3}{(1.543 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7301) = 7.622 \times 10^7$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(7.622 \times 10^7)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7301} \right)^{9/16} \right]^{8/27}} \right\}^2 = 56.53$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02536 \text{ W/m}\cdot^\circ\text{C}}{0.5 \text{ m}} (56.53) = 2.868 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = 2[(0.5 \text{ m})(1.10 \text{ m}) + (0.5 \text{ m})(3.60 \text{ m})] = 4.7 \text{ m}^2$$

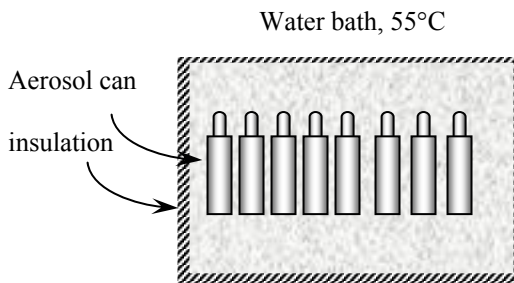
Then the total rate of heat loss from the outer surface of the insulated tank by convection and radiation becomes

$$\begin{aligned} \dot{Q} &= \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (2.868 \text{ W/m}^2\cdot^\circ\text{C})(4.7 \text{ m}^2)(26 - 20)^\circ\text{C} \\ &\quad + (0.1)(4.7 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(26 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4] \\ &= 97.5 \text{ W} \end{aligned}$$

In steady operation, the heat lost by the side surfaces of the tank must be equal to the heat lost from the exposed surface of the insulation by convection and radiation, which must be equal to the heat conducted through the insulation. The second conditions requires the surface temperature to be

$$\dot{Q} = \dot{Q}_{\text{insulation}} = kA_s \frac{T_{\text{tank}} - T_s}{L} \rightarrow 97.5 \text{ W} = (0.035 \text{ W/m}\cdot^\circ\text{C})(4.7 \text{ m}^2) \frac{(55 - T_s)^\circ\text{C}}{0.05 \text{ m}}$$

It gives $T_s = 25.38^\circ\text{C}$, which is very close to the assumed temperature, 26°C. Therefore, there is no need to repeat the calculations.



The total amount of heat loss and its cost during one year are

$$Q_{total} = \dot{Q}_{total} \Delta t = (97.5 \text{ W})(8760 \text{ h}) = 853.7 \text{ kWh}$$

$$\text{Cost} = (853.7 \text{ kWh})(\$0.085 / \text{kWh}) = \$72.6$$

Then money saved during a one-year period due to insulation becomes

$$\text{Money saved} = \text{Cost}_{\text{without insulation}} - \text{Cost}_{\text{with insulation}} = \$1116 - \$72.6 = \$1043$$

where \$1116 is obtained from the solution of Problem 14-28. The insulation will pay for itself in

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$350}{\$1043 / \text{yr}} = \mathbf{0.3354 \text{ yr} = 122 \text{ days}}$$

Discussion We would definitely recommend the installation of insulation in this case.

14-29 A printed circuit board (PCB) is placed in a room. The average temperature of the hot surface of the board is to be determined for different orientations.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 3 The heat loss from the back surface of the board is negligible.

Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (45 + 20)/2 = 32.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02607 \text{ W/m}\cdot^\circ\text{C}$$

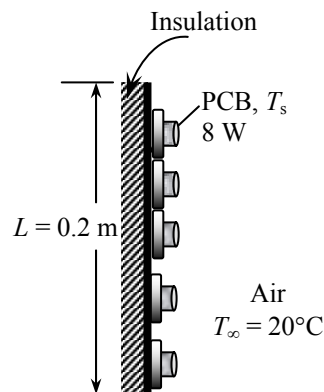
$$\nu = 1.631 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7275$$

$$\beta = \frac{1}{T_f} = \frac{1}{(32.5 + 273)\text{K}} = 0.003273 \text{ K}^{-1}$$

Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown

(a) **Vertical PCB**. We start the solution process by “guessing” the surface temperature to be 45°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the height of the PCB, $L_c = L = 0.2 \text{ m}$. Then,



$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003273 \text{ K}^{-1})(45 - 20 \text{ K})(0.2 \text{ m})^3}{(1.631 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7275) = 1.756 \times 10^7$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.756 \times 10^7)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7275} \right)^{9/16} \right]^{8/27}} \right\}^2 = 36.78$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02607 \text{ W/m}\cdot^\circ\text{C}}{0.2 \text{ m}} (36.78) = 4.794 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (0.15 \text{ m})(0.2 \text{ m}) = 0.03 \text{ m}^2$$

Heat loss by both natural convection and radiation heat can be expressed as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ 8 \text{ W} &= (4.794 \text{ W/m}^2\cdot^\circ\text{C})(0.03 \text{ m}^2)(T_s - 20)^\circ\text{C} + (0.8)(0.03 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (20 + 273 \text{ K})^4] \end{aligned}$$

Its solution is

$$T_s = 46.6^\circ\text{C}$$

which is sufficiently close to the assumed value of 45°C for the evaluation of the properties and h .

(b) **Horizontal, hot surface facing up** Again we assume the surface temperature to be 45°C and use the properties evaluated above. The characteristic length in this case is

$$L_c = \frac{A_s}{p} = \frac{(0.20 \text{ m})(0.15 \text{ m})}{2(0.2 \text{ m} + 0.15 \text{ m})} = 0.0429 \text{ m}.$$

Then

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003273 \text{ K}^{-1})(45 - 20 \text{ K})(0.0429 \text{ m})^3}{(1.631 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7275) = 1.728 \times 10^5$$

$$\text{Nu} = 0.54\text{Ra}^{1/4} = 0.54(1.728 \times 10^5)^{1/4} = 11.01$$

$$h = \frac{k}{L_c} Nu = \frac{0.02607 \text{ W/m} \cdot ^\circ\text{C}}{0.0429 \text{ m}} (11.01) = 6.696 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Heat loss by both natural convection and radiation heat can be expressed as

$$\begin{aligned}\dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ 8 \text{ W} &= (6.696 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03 \text{ m}^2)(T_s - 20)^\circ\text{C} + (0.8)(0.03 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (20 + 273 \text{ K})^4]\end{aligned}$$

Its solution is

$$T_s = \mathbf{42.6^\circ\text{C}}$$

which is sufficiently close to the assumed value of 45°C in the evaluation of the properties and h .

(c) **Horizontal, hot surface facing down** This time we expect the surface temperature to be higher, and assume the surface temperature to be 50°C . We will check this assumption after obtaining result and repeat calculations with a better assumption, if necessary. The properties of air at the film temperature of $(50+20)/2=35^\circ\text{C}$ are (Table A-22)

$$k = 0.02625 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.655 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7268$$

$$\beta = \frac{1}{T_f} = \frac{1}{(35 + 273)\text{K}} = 0.003247 \text{ K}^{-1}$$

The characteristic length in this case is, from part (b), $L_c = 0.0429 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003247 \text{ K}^{-1})(50 - 20 \text{ K})(0.0429 \text{ m})^3}{(1.655 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7268) = 200,200$$

$$Nu = 0.27 Ra^{1/4} = 0.27(200,200)^{1/4} = 5.711$$

$$h = \frac{k}{L_c} Nu = \frac{0.02625 \text{ W/m} \cdot ^\circ\text{C}}{0.0429 \text{ m}} (5.711) = 3.494 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Considering both natural convection and radiation heat losses

$$\begin{aligned}\dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ 8 \text{ W} &= (3.494 \text{ W/m}^2 \cdot ^\circ\text{C})(0.03 \text{ m}^2)(T_s - 20)^\circ\text{C} + (0.8)(0.03 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (20 + 273 \text{ K})^4]\end{aligned}$$

Its solution is

$$T_s = \mathbf{50.3^\circ\text{C}}$$

which is very close to the assumed value. Therefore, there is no need to repeat calculations.

14-30 EES Prob. 14-29 is reconsidered. The effects of the room temperature and the emissivity of the board on the temperature of the hot surface of the board for different orientations of the board are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

L=0.2 [m]
w=0.15 [m]
T_infinity=20 [C]
Q_dot=8 [W]
epsilon=0.8
T_surr=T_infinity

"PROPERTIES"

Fluid\$='air'
k=Conductivity(Fluid\$, T=T_film)
Pr=Prandtl(Fluid\$, T=T_film)
rho=Density(Fluid\$, T=T_film, P=101.3)
mu=Viscosity(Fluid\$, T=T_film)
nu=mu/rho
beta=1/(T_film+273)
T_film=1/2*(T_s_a+T_infinity)
sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzmann constant"
g=9.807 [m/s^2] "gravitational acceleration"

"ANALYSIS"

"(a), plate is vertical"

delta_a=L
Ra_a=(g*beta*(T_s_a-T_infinity)*delta_a^3)/nu^2*Pr
Nusselt_a=0.59*Ra_a^0.25
h_a=k/delta_a*Nusselt_a
A=w*L
Q_dot=h_a*A*(T_s_a-T_infinity)+epsilon*A*sigma*((T_s_a+273)^4-(T_surr+273)^4)

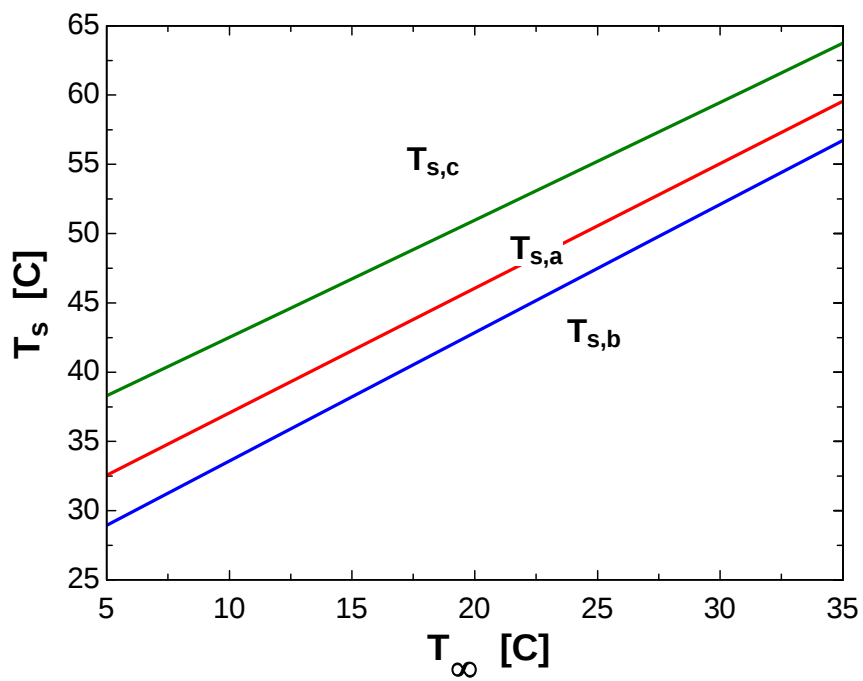
"(b), plate is horizontal with hot surface facing up"

delta_b=A/p
p=2*(w+L)
Ra_b=(g*beta*(T_s_b-T_infinity)*delta_b^3)/nu^2*Pr
Nusselt_b=0.54*Ra_b^0.25
h_b=k/delta_b*Nusselt_b
Q_dot=h_b*A*(T_s_b-T_infinity)+epsilon*A*sigma*((T_s_b+273)^4-(T_surr+273)^4)

"(c), plate is horizontal with hot surface facing down"

delta_c=delta_b
Ra_c=Ra_b
Nusselt_c=0.27*Ra_c^0.25
h_c=k/delta_c*Nusselt_c
Q_dot=h_c*A*(T_s_c-T_infinity)+epsilon*A*sigma*((T_s_c+273)^4-(T_surr+273)^4)

T_{∞} [F]	$T_{s,a}$ [C]	$T_{s,b}$ [C]	$T_{s,c}$ [C]
5	32.54	28.93	38.29
7	34.34	30.79	39.97
9	36.14	32.65	41.66
11	37.95	34.51	43.35
13	39.75	36.36	45.04
15	41.55	38.22	46.73
17	43.35	40.07	48.42
19	45.15	41.92	50.12
21	46.95	43.78	51.81
23	48.75	45.63	53.51
25	50.55	47.48	55.21
27	52.35	49.33	56.91
29	54.16	51.19	58.62
31	55.96	53.04	60.32
33	57.76	54.89	62.03
35	59.56	56.74	63.74



14-31 CD EES Absorber plates whose back side is heavily insulated is placed horizontally outdoors. Solar radiation is incident on the plate. The equilibrium temperature of the plate is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

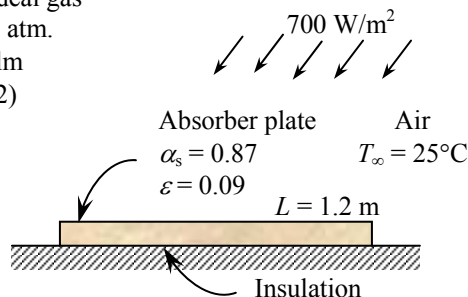
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (115 + 25)/2 = 70^\circ\text{C}$ are (Table A-22)

$$k = 0.02881 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.995 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7177$$

$$\beta = \frac{1}{T_f} = \frac{1}{(70 + 273)\text{K}} = 0.002915 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 115°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The

characteristic length in this case is $L_c = \frac{A_s}{p} = \frac{(1.2 \text{ m})(0.8 \text{ m})}{2(1.2 \text{ m} + 0.8 \text{ m})} = 0.24 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002915 \text{ K}^{-1})(115 - 25 \text{ K})(0.24 \text{ m})^3}{(1.995 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7177) = 6.414 \times 10^7$$

$$Nu = 0.54 Ra^{1/4} = 0.54 (6.414 \times 10^7)^{1/4} = 48.33$$

$$h = \frac{k}{L_c} Nu = \frac{0.02881 \text{ W/m}\cdot^\circ\text{C}}{0.24 \text{ m}} (48.33) = 5.801 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (0.8 \text{ m})(1.2 \text{ m}) = 0.96 \text{ m}^2$$

In steady operation, the heat gain by the plate by absorption of solar radiation must be equal to the heat loss by natural convection and radiation. Therefore,

$$\dot{Q} = \alpha \dot{q} A_s = (0.87)(700 \text{ W/m}^2)(0.96 \text{ m}^2) = 584.6 \text{ W}$$

$$\dot{Q} = h A_s (T_s - T_\infty) + \epsilon A_s \sigma (T_s^4 - T_{sky}^4)$$

$$584.6 \text{ W} = (5.801 \text{ W/m}^2\cdot^\circ\text{C})(0.96 \text{ m}^2)(T_s - 25)^\circ\text{C} + (0.09)(0.96 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (10 + 273 \text{ K})^4]$$

Its solution is $T_s = 115.6^\circ\text{C}$

which is identical to the assumed value. Therefore there is no need to repeat calculations.

If the absorber plate is made of ordinary aluminum which has a solar absorptivity of 0.28 and an emissivity of 0.07, the rate of solar gain becomes

$$\dot{Q} = \alpha \dot{q} A_s = (0.28)(700 \text{ W/m}^2)(0.96 \text{ m}^2) = 188.2 \text{ W}$$

Again noting that in steady operation the heat gain by the plate by absorption of solar radiation must be equal to the heat loss by natural convection and radiation, and using the convection coefficient determined above for convenience,

$$\dot{Q} = h A_s (T_s - T_\infty) + \epsilon A_s \sigma (T_s^4 - T_{sky}^4)$$

$$188.2 \text{ W} = (5.801 \text{ W/m}^2\cdot^\circ\text{C})(0.96 \text{ m}^2)(T_s - 25)^\circ\text{C} + (0.07)(0.96 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (10 + 273 \text{ K})^4]$$

Its solution is $T_s = 55.2^\circ\text{C}$

Repeating the calculations at the new film temperature of 40°C , we obtain

$$h = 4.524 \text{ W/m}^2\cdot^\circ\text{C} \text{ and } T_s = 62.8^\circ\text{C}$$

14-32 An absorber plate whose back side is heavily insulated is placed horizontally outdoors. Solar radiation is incident on the plate. The equilibrium temperature of the plate is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

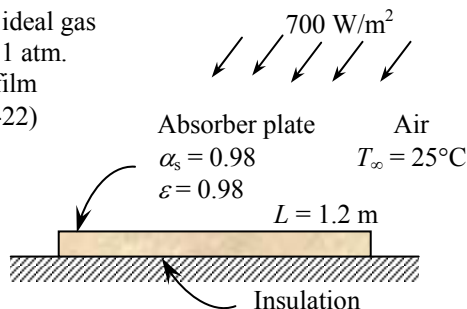
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (70 + 25)/2 = 47.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02717 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.774 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7235$$

$$\beta = \frac{1}{T_f} = \frac{1}{(47.5 + 273)\text{K}} = 0.00312 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 70°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The

characteristic length in this case is $L_c = \frac{A_s}{p} = \frac{(1.2 \text{ m})(0.8 \text{ m})}{2(1.2 \text{ m} + 0.8 \text{ m})} = 0.24 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00312 \text{ K}^{-1})(70 - 25 \text{ K})(0.24 \text{ m})^3}{(1.774 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7235) = 4.379 \times 10^7$$

$$\text{Nu} = 0.54 \text{Ra}^{1/4} = 0.54(4.379 \times 10^7)^{1/4} = 43.93$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.02717 \text{ W/m}\cdot^\circ\text{C}}{0.24 \text{ m}} (43.93) = 4.973 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (0.8 \text{ m})(1.2 \text{ m}) = 0.96 \text{ m}^2$$

In steady operation, the heat gain by the plate by absorption of solar radiation must be equal to the heat loss by natural convection and radiation. Therefore,

$$\dot{Q} = \alpha \dot{q} A_s = (0.98)(700 \text{ W/m}^2)(0.96 \text{ m}^2) = 658.6 \text{ W}$$

$$\dot{Q} = h A_s (T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4)$$

$$658.6 \text{ W} = (4.973 \text{ W/m}^2\cdot^\circ\text{C})(0.96 \text{ m}^2)(T_s - 25)^\circ\text{C} + (0.98)(0.96 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (10 + 273 \text{ K})^4]$$

Its solution is $T_s = 73.5^\circ\text{C}$

which is close to the assumed value. Therefore there is no need to repeat calculations.

For a white painted absorber plate, the solar absorptivity is 0.26 and the emissivity is 0.90. Then the rate of solar gain becomes

$$\dot{Q} = \alpha \dot{q} A_s = (0.26)(700 \text{ W/m}^2)(0.96 \text{ m}^2) = 174.7 \text{ W}$$

Again noting that in steady operation the heat gain by the plate by absorption of solar radiation must be equal to the heat loss by natural convection and radiation, and using the convection coefficient determined above for convenience (actually, we should calculate the new h using data at a lower temperature, and iterating if necessary for better accuracy),

$$\dot{Q} = h A_s (T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4)$$

$$174.7 \text{ W} = (4.973 \text{ W/m}^2\cdot^\circ\text{C})(0.96 \text{ m}^2)(T_s - 25)^\circ\text{C} + (0.90)(0.96 \text{ m}^2)(5.67 \times 10^{-8})[(T_s + 273)^4 - (10 + 273 \text{ K})^4]$$

Its solution is $T_s = 35.0^\circ\text{C}$

Discussion If we recalculated the h using air properties at 30°C , we would obtain

$$h = 3.47 \text{ W/m}^2\cdot^\circ\text{C} \text{ and } T_s = 36.6^\circ\text{C}$$

14-33 A resistance heater is placed along the centerline of a horizontal cylinder whose two circular side surfaces are well insulated. The natural convection heat transfer coefficient and whether the radiation effect is negligible are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

Analysis The heat transfer surface area of the cylinder is

$$A = \pi DL = \pi(0.02 \text{ m})(0.8 \text{ m}) = 0.05027 \text{ m}^2$$

Noting that in steady operation the heat dissipated from the outer surface must equal to the electric power consumed, and radiation is negligible, the convection heat transfer is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow h = \frac{\dot{Q}}{A_s(T_s - T_\infty)} = \frac{60 \text{ W}}{(0.05027 \text{ m}^2)(120 - 20)^\circ\text{C}} = \mathbf{11.9 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

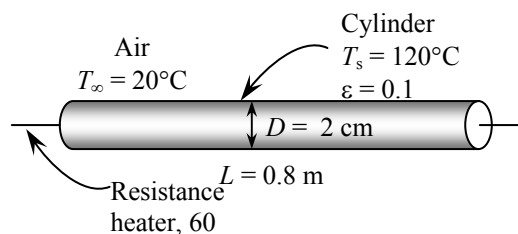
The radiation heat loss from the cylinder is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.1)(0.05027 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(120 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4] = 4.7 \text{ W} \end{aligned}$$

Therefore, the fraction of heat loss by radiation is

$$\text{Radiation fraction} = \frac{\dot{Q}_{radiation}}{\dot{Q}_{total}} = \frac{4.7 \text{ W}}{60 \text{ W}} = 0.078 = 7.8\%$$

which is greater than 5%. Therefore, the radiation effect is still more than acceptable, and corrections must be made for the radiation effect.



14-34 A thick fluid flows through a pipe in calm ambient air. The pipe is heated electrically. The power rating of the electric resistance heater and the cost of electricity during a 10-h period are to be determined. ✓

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

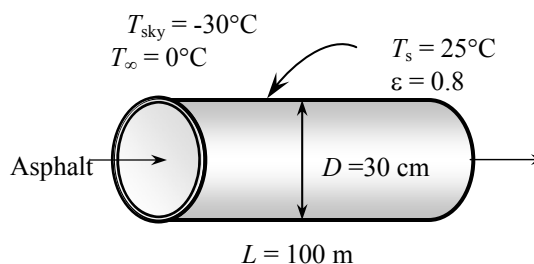
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (25 + 0)/2 = 12.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02458 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.448 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7330$$

$$\beta = \frac{1}{T_f} = \frac{1}{(12.5 + 273)\text{K}} = 0.003503 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the outer diameter of the pipe, $L_c = D = 0.3 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003503 \text{ K}^{-1})(25 - 0 \text{ K})(0.3 \text{ m})^3}{(1.448 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7330) = 8.106 \times 10^7$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (8.106 \times 10^7)^{1/6}}{\left[1 + (0.559 / 0.7330)^{9/16} \right]^{8/27}} \right\}^2 = 53.29$$

$$h = \frac{k}{L_c} Nu = \frac{0.02458 \text{ W/m}\cdot^\circ\text{C}}{0.3 \text{ m}} (53.29) = 4.366 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.3 \text{ m})(100 \text{ m}) = 94.25 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (4.366 \text{ W/m}^2\cdot^\circ\text{C})(94.25 \text{ m}^2)(25 - 0)^\circ\text{C} = 10,287 \text{ W}$$

The radiation heat loss from the cylinder is

$$\begin{aligned} \dot{Q}_{rad} &= \epsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.8)(94.25 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(25 + 273 \text{ K})^4 - (-30 + 273 \text{ K})^4] = 18,808 \text{ W} \end{aligned}$$

Then,

$$\dot{Q}_{total} = \dot{Q}_{\text{natural convection}} + \dot{Q}_{\text{radiation}} = 10,287 + 18,808 = 29,094 \text{ W} = \mathbf{29.1 \text{ kW}}$$

The total amount and cost of heat loss during a 10 hour period is

$$Q = \dot{Q} \Delta t = (29.1 \text{ kW})(10 \text{ h}) = 290.9 \text{ kWh}$$

$$\text{Cost} = (290.9 \text{ kWh})(\$0.09/\text{kWh}) = \mathbf{\$26.18}$$

14-35 A fluid flows through a pipe in calm ambient air. The pipe is heated electrically. The thickness of the insulation needed to reduce the losses by 85% and the money saved during 10-h are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

Properties Insulation will drop the outer surface temperature to a value close to the ambient temperature, and possible below it because of the very low sky temperature for radiation heat loss. For convenience, we use the properties of air at 1 atm and 5°C (the anticipated film temperature) (Table A-22),

$$k = 0.02401 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.382 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7350$$

$$\beta = \frac{1}{T_f} = \frac{1}{(5 + 273)\text{K}} = 0.003597 \text{ K}^{-1}$$

Analysis The rate of heat loss in the previous problem was obtained to be 29,094 W. Noting that insulation will cut down the heat losses by 85%, the rate of heat loss will be

$$\dot{Q} = (1 - 0.85)\dot{Q}_{\text{no insulation}} = 0.15 \times 29,094 \text{ W} = 4364 \text{ W}$$

The amount of energy and money insulation will save during a 10-h period is simply determined from

$$Q_{\text{saved, total}} = \dot{Q}_{\text{saved}} \Delta t = (0.85 \times 29.094 \text{ kW})(10 \text{ h}) = 247.3 \text{ kWh}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (247.3 \text{ kWh})(\$0.09 / \text{kWh}) = \text{\$22.26}$$

The characteristic length in this case is the outer diameter of the insulated pipe,

$L_c = D + 2t_{\text{insul}} = 0.3 + 2t_{\text{insul}}$ where t_{insul} is the thickness of insulation in m. Then the problem can be formulated for T_s and t_{insul} as follows:

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2 \text{Pr}} = \frac{(9.81 \text{ m/s}^2)(0.003597 \text{ K}^{-1})[(T_s - 273)\text{K}](0.3 + 2t_{\text{insul}})^3}{(1.382 \times 10^{-5} \text{ m}^2/\text{s})^2 (0.7350)}$$

$$Nu = \left\{ 0.6 + \frac{0.387Ra^{1/6}}{\left[1 + (0.559/\text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387Ra^{1/6}}{\left[1 + (0.559/0.7350)^{9/16} \right]^{8/27}} \right\}^2$$

$$h = \frac{k}{L_c} Nu = \frac{0.02401 \text{ W/m} \cdot ^\circ\text{C}}{L_c} Nu$$

$$A_s = \pi D_o L = \pi(0.3 + 2t_{\text{insul}})(100 \text{ m})$$

The total rate of heat loss from the outer surface of the insulated pipe by convection and radiation becomes

$$\dot{Q} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4)$$

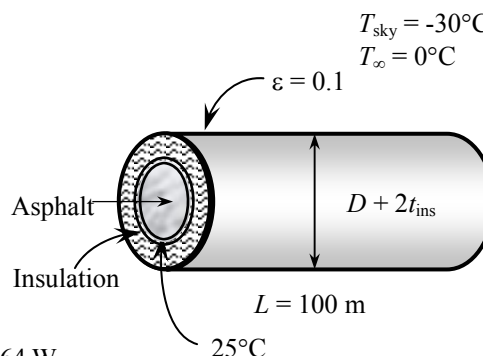
$$4364 = hA_s(T_s - 273) + (0.1)A_s(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_s^4 - (-30 + 273 \text{ K})^4]$$

In steady operation, the heat lost by the side surfaces of the pipe must be equal to the heat lost from the exposed surface of the insulation by convection and radiation, which must be equal to the heat conducted through the insulation. Therefore,

$$\dot{Q} = \dot{Q}_{\text{insulation}} = \frac{2\pi k L (T_{\text{tank}} - T_s)}{\ln(D_o / D)} \rightarrow 4364 \text{ W} = \frac{2\pi(0.035 \text{ W/m} \cdot ^\circ\text{C})(100 \text{ m})(298 - T_s) \text{ K}}{\ln[(0.3 + 2t_{\text{insul}}) / 0.3]}$$

The solution of all of the equations above simultaneously using an equation solver gives $T_s = 281.5 \text{ K} = 8.5^\circ\text{C}$ and $t_{\text{insul}} = \mathbf{0.013 \text{ m} = 1.3 \text{ cm}}$.

Note that the film temperature is $(8.5 + 0)/2 = 4.25^\circ\text{C}$ which is very close to the assumed value of 5°C . Therefore, there is no need to repeat the calculations using properties at this new film temperature.



14-36E An industrial furnace that resembles a horizontal cylindrical enclosure whose end surfaces are well insulated. The highest allowable surface temperature of the furnace and the annual cost of this loss to the plant are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

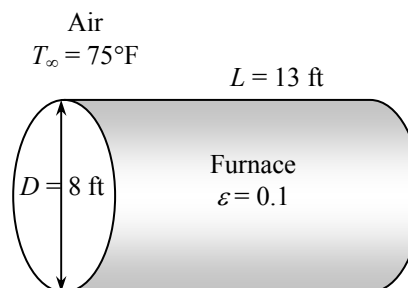
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (140 + 75)/2 = 107.5^\circ\text{F}$ are (Table A-22E)

$$k = 0.01546 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1852 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7249$$

$$\beta = \frac{1}{T_f} = \frac{1}{(107.5 + 460)\text{R}} = 0.001762 \text{ R}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 140°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the furnace, $L_c = D = 8 \text{ ft}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001762 \text{ R}^{-1})(140 - 75 \text{ R})(8 \text{ ft})^3}{(0.1852 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7249) = 3.991 \times 10^{10}$$

$$\text{Nu} = \left\{ 0.6 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387(3.991 \times 10^{10})^{1/6}}{\left[1 + (0.559 / 0.7249)^{9/16} \right]^{8/27}} \right\}^2 = 376.8$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.01546 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{8 \text{ ft}} (376.8) = 0.7287 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_s = \pi DL = \pi(8 \text{ ft})(13 \text{ ft}) = 326.7 \text{ ft}^2$$

The total rate of heat generated in the furnace is

$$\dot{Q}_{\text{gen}} = (0.82)(48 \text{ therms/h})(100,000 \text{ Btu/therm}) = 3.936 \times 10^6 \text{ Btu/h}$$

Noting that 1% of the heat generated can be dissipated by natural convection and radiation ,

$$\dot{Q} = (0.01)(3.936 \times 10^6 \text{ Btu/h}) = 39,360 \text{ Btu/h}$$

The total rate of heat loss from the furnace by natural convection and radiation can be expressed as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ 39,360 \text{ Btu/h} &= (0.7287 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(326.7 \text{ ft}^2)[T_s - (75 + 460 \text{ R})] \\ &\quad + (0.85)(326.7 \text{ m}^2)(0.1714 \times 10^{-8} \text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)[T_s^4 - (75 + 460 \text{ R})^4] \end{aligned}$$

Its solution is

$$T_s = 601.8 \text{ R} = \mathbf{141.8^\circ\text{F}}$$

which is very close to the assumed value. Therefore, there is no need to repeat calculations. The total amount of heat loss and its cost during a 2800 hour period is

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{total}} \Delta t = (39,360 \text{ Btu/h})(2800 \text{ h}) = 1.102 \times 10^8 \text{ Btu}$$

$$\text{Cost} = (1.102 \times 10^8 / 100,000 \text{ therm})(\$1.15 / \text{therm}) = \mathbf{\$1267}$$

14-37 A glass window is considered. The convection heat transfer coefficient on the inner side of the window, the rate of total heat transfer through the window, and the combined natural convection and radiation heat transfer coefficient on the outer surface of the window are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

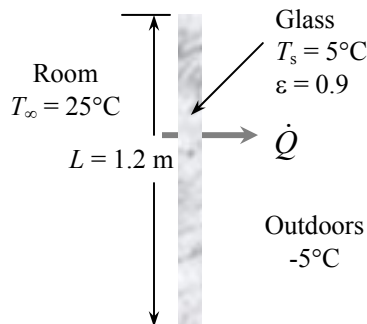
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (5 + 25)/2 = 15^\circ\text{C}$ are (Table A-22)

$$k = 0.02476 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.470 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7323$$

$$\beta = \frac{1}{T_f} = \frac{1}{(15 + 273)\text{K}} = 0.003472 \text{ K}^{-1}$$



Analysis (a) The characteristic length in this case is the height of the window, $L_c = L = 1.2 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003472 \text{ K}^{-1})(25 - 5 \text{ K})(1.2 \text{ m})^3}{(1.470 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7323) = 3.989 \times 10^9$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(3.989 \times 10^9)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7323} \right)^{9/16} \right]^{8/27}} \right\}^2 = 189.7$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02476 \text{ W/m}\cdot^\circ\text{C}}{1.2 \text{ m}} (189.7) = \mathbf{3.915 \text{ W/m}^2\cdot^\circ\text{C}}$$

$$A_s = (1.2 \text{ m})(2 \text{ m}) = 2.4 \text{ m}^2$$

(b) The sum of the natural convection and radiation heat transfer from the room to the window is

$$\dot{Q}_{\text{convection}} = hA_s(T_\infty - T_s) = (3.915 \text{ W/m}^2\cdot^\circ\text{C})(2.4 \text{ m}^2)(25 - 5)^\circ\text{C} = 187.9 \text{ W}$$

$$\begin{aligned} \dot{Q}_{\text{radiation}} &= \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) \\ &= (0.9)(2.4 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(25 + 273 \text{ K})^4 - (5 + 273 \text{ K})^4] = 234.3 \text{ W} \end{aligned}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{convection}} + \dot{Q}_{\text{radiation}} = 187.9 + 234.3 = \mathbf{422.2 \text{ W}}$$

(c) The outer surface temperature of the window can be determined from

$$\dot{Q}_{\text{total}} = \frac{kA_s}{t}(T_{s,i} - T_{s,o}) \longrightarrow T_{s,o} = T_{s,i} - \frac{\dot{Q}_{\text{total}} t}{kA_s} = 5^\circ\text{C} - \frac{(422.2 \text{ W})(0.006 \text{ m})}{(0.78 \text{ W/m}\cdot^\circ\text{C})(2.4 \text{ m}^2)} = 3.65^\circ\text{C}$$

Then the combined natural convection and radiation heat transfer coefficient on the outer window surface becomes

$$\dot{Q}_{\text{total}} = h_{\text{combined}} A_s (T_{s,o} - T_{\infty,o})$$

$$\text{or } h_{\text{combined}} = \frac{\dot{Q}_{\text{total}}}{A_s (T_{s,o} - T_{\infty,o})} = \frac{422.2 \text{ W}}{(2.4 \text{ m}^2)[3.65 - (-5)]^\circ\text{C}} = \mathbf{20.35 \text{ W/m}^2\cdot^\circ\text{C}}$$

Note that $\Delta T = \dot{Q}R$ and thus the thermal resistance R of a layer is proportional to the temperature drop across that layer. Therefore, the fraction of thermal resistance of the glass is equal to the ratio of the temperature drop across the glass to the overall temperature difference,

$$\frac{R_{\text{glass}}}{R_{\text{total}}} = \frac{\Delta T_{\text{glass}}}{\Delta T R_{\text{total}}} = \frac{5 - 3.65}{25 - (-5)} = 0.045 \text{ (or 4.5\%)}$$

which is low. Thus it is reasonable to neglect the thermal resistance of the glass.

14-38 An insulated electric wire is exposed to calm air. The temperature at the interface of the wire and the plastic insulation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

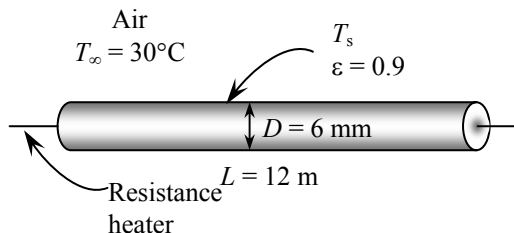
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (50 + 30)/2 = 40^\circ\text{C}$ are (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7255$$

$$\beta = \frac{1}{T_f} = \frac{1}{(40 + 273)\text{K}} = 0.003195 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 50°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the insulated wire $L_c = D = 0.006 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003195 \text{ K}^{-1})(50 - 30 \text{ K})(0.006 \text{ m})^3}{(1.702 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7255) = 339.3$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (339.3)^{1/6}}{\left[1 + (0.559 / 0.7255)^{9/16} \right]^{8/27}} \right\}^2 = 2.101$$

$$h = \frac{k}{D} Nu = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{0.006 \text{ m}} (2.101) = 9.327 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.006 \text{ m})(12 \text{ m}) = 0.2262 \text{ m}^2$$

The rate of heat generation, and thus the rate of heat transfer is

$$\dot{Q} = VI = (7 \text{ V})(10 \text{ A}) = 70 \text{ W}$$

Considering both natural convection and radiation, the total rate of heat loss can be expressed as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \epsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ 70 \text{ W} &= (9.327 \text{ W/m}^2\cdot^\circ\text{C})(0.226 \text{ m}^2)(T_s - 30)^\circ\text{C} \\ &\quad + (0.9)(0.2262 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273)^4 - (30 + 273 \text{ K})^4] \end{aligned}$$

Its solution is

$$T_s = 49.9^\circ\text{C}$$

which is very close to the assumed value of 50°C . Then the temperature at the interface of the wire and the plastic cover in steady operation becomes

$$\dot{Q} = \frac{2\pi kL}{\ln(D_2/D_1)}(T_i - T_s) \longrightarrow T_i = T_s + \frac{\dot{Q} \ln(D_2/D_1)}{2\pi kL} = 49.9^\circ\text{C} + \frac{(70 \text{ W}) \ln(6/3)}{2\pi(0.20 \text{ W/m}\cdot^\circ\text{C})(12 \text{ m})} = \mathbf{53.1^\circ\text{C}}$$

14-39 A steam pipe extended from one end of a plant to the other with no insulation on it. The rate of heat loss from the steam pipe and the annual cost of those heat losses are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

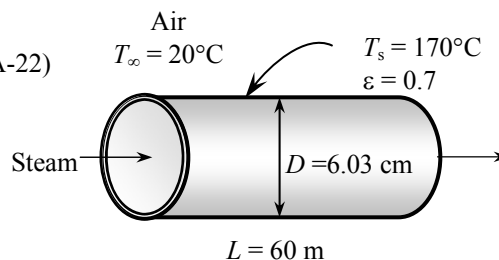
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (170 + 20)/2 = 95^\circ\text{C}$ are (Table A-22)

$$k = 0.0306 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 2.254 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7122$$

$$\beta = \frac{1}{T_f} = \frac{1}{(95 + 273)\text{K}} = 0.002717 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the outer diameter of the pipe, $L_c = D = 0.0603 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002717 \text{ K}^{-1})(170 - 20 \text{ K})(0.0603 \text{ m})^3}{(2.254 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7122) = 1.229 \times 10^6$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (1.229 \times 10^6)^{1/6}}{\left[1 + (0.559 / 0.7122)^{9/16} \right]^{8/27}} \right\}^2 = 15.41$$

$$h = \frac{k}{D} Nu = \frac{0.0306 \text{ W/m}\cdot^\circ\text{C}}{0.0603 \text{ m}} (15.41) = 7.821 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.0603 \text{ m})(60 \text{ m}) = 11.37 \text{ m}^2$$

Then the total rate of heat transfer by natural convection and radiation becomes

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (7.821 \text{ W/m}^2\cdot^\circ\text{C})(11.37 \text{ m}^2)(170 - 20)^\circ\text{C} \\ &\quad + (0.7)(11.37 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(170 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4] \\ &= 27,393 \text{ W} = \mathbf{27.4 \text{ kW}} \end{aligned}$$

The total amount of gas consumption and its cost during a one-year period is

$$Q_{\text{gas}} = \frac{\dot{Q}\Delta t}{\eta} = \frac{27.393 \text{ kJ/s}}{0.78} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) (8760 \text{ h/yr} \times 3600 \text{ s/h}) = 10,498 \text{ therms/yr}$$

$$\text{Cost} = (10,498 \text{ therms/yr})(\$1.10 / \text{therm}) = \mathbf{\$11,550/\text{yr}}$$

14-40 EES Prob. 14-39 is reconsidered. The effect of the surface temperature of the steam pipe on the rate of heat loss from the pipe and the annual cost of this heat loss is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

L=60 [m]
D=0.0603 [m]
T_s=170 [C]
T_infinity=20 [C]
epsilon=0.7
T_surr=T_infinity
eta_furnace=0.78
UnitCost=1.10 [\$/therm]
time=24*365 [h]

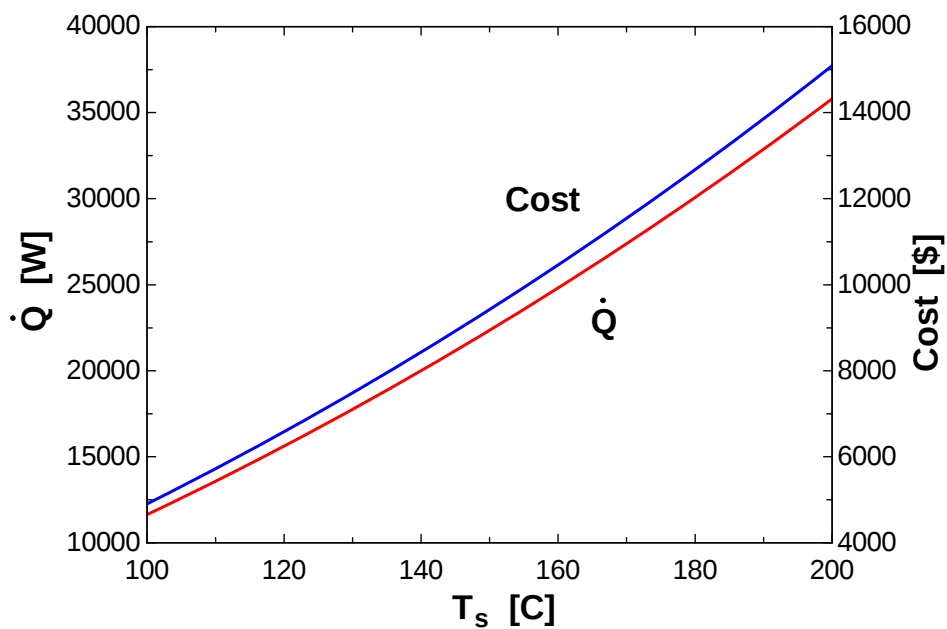
"PROPERTIES"

Fluid\$='air'
k=Conductivity(Fluid\$, T=T_film)
Pr=Prandtl(Fluid\$, T=T_film)
rho=Density(Fluid\$, T=T_film, P=101.3)
mu=Viscosity(Fluid\$, T=T_film)
nu=mu/rho
beta=1/(T_film+273)
T_film=1/2*(T_s+T_infinity)
sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzmann constant"
g=9.807 [m/s^2] "gravitational acceleration"

"ANALYSIS"

delta=D
Ra=(g*beta*(T_s-T_infinity)*delta^3)/nu^2*Pr
Nusselt=(0.6+(0.387*Ra^(1/6))/(1+(0.559/Pr)^(9/16)))^(8/27))^2
h=k/delta*Nusselt
A=pi*D*L
Q_dot=h*A*(T_s-T_infinity)+epsilon*A*sigma*((T_s+273)^4-(T_surr+273)^4)
Q_gas=(Q_dot*time)/eta_furnace*Convert(h, s)*Convert(J, kJ)*Convert(kJ, therm)
Cost=Q_gas*UnitCost

T_s [C]	$\dot{Q}$ [W]	Cost [\$]
100	11636	4905
105	12594	5309
110	13577	5723
115	14585	6148
120	15618	6584
125	16676	7030
130	17760	7486
135	18869	7954
140	20004	8432
145	21166	8922
150	22355	9423
155	23570	9936
160	24814	10460
165	26085	10996
170	27385	11543
175	28713	12103
180	30071	12676
185	31459	13261
190	32877	13859
195	34327	14470
200	35807	15094



14-41 A steam pipe extended from one end of a plant to the other. It is proposed to insulate the steam pipe for \$750. The simple payback period of the insulation to pay for itself from the energy it saves are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

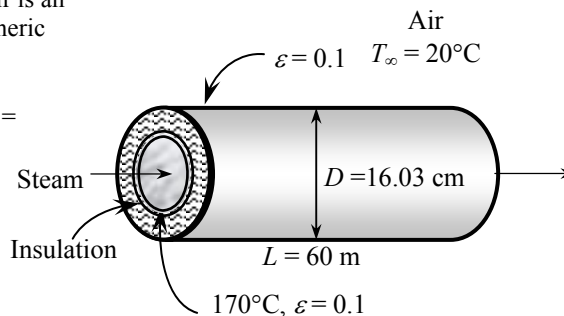
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (35 + 20)/2 = 27.5^\circ\text{C}$ are (Table A-22)

$$k = 0.0257 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.584 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7289$$

$$\beta = \frac{1}{T_f} = \frac{1}{(27.5 + 273)\text{K}} = 0.003328 \text{ K}^{-1}$$



Analysis Insulation will drop the outer surface temperature to a value close to the ambient temperature. The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the outer surface temperature to be 35°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the insulated pipe, $L_c = D = 0.1603 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003328 \text{ K}^{-1})(35 - 20 \text{ K})(0.1603 \text{ m})^3}{(1.584 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7289) = 5.856 \times 10^6$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (5.856 \times 10^6)^{1/6}}{\left[1 + (0.559 / 0.7289)^{9/16} \right]^{8/27}} \right\}^2 = 24.23$$

$$h = \frac{k}{D} Nu = \frac{0.0257 \text{ W/m}\cdot^\circ\text{C}}{0.1603 \text{ m}} (24.23) = 3.884 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi (0.1603 \text{ m})(60 \text{ m}) = 30.22 \text{ m}^2$$

Then the total rate of heat loss from the outer surface of the insulated pipe by convection and radiation becomes

$$\begin{aligned} \dot{Q} &= \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (3.884 \text{ W/m}^2\cdot^\circ\text{C})(30.22 \text{ m}^2)(35 - 20)^\circ\text{C} \\ &\quad + (0.1)(30.22 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(35 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4] \\ &= 2039 \text{ W} \end{aligned}$$

In steady operation, the heat lost from the exposed surface of the insulation by convection and radiation must be equal to the heat conducted through the insulation. This requirement gives the surface temperature to be

$$\dot{Q} = \dot{Q}_{\text{insulation}} = \frac{T_{s,i} - T_s}{R_{\text{ins}}} = \frac{T_{s,i} - T_s}{\frac{\ln(D_2 / D_1)}{2\pi kL}} \rightarrow 2039 \text{ W} = \frac{(170 - T_s)^\circ\text{C}}{\frac{\ln(16.03 / 6.03)}{2\pi(0.038 \text{ W/m}\cdot^\circ\text{C})(60 \text{ m})}}$$

It gives 30.8°C for the surface temperature, which is somewhat different than the assumed value of 35°C .

Repeating the calculations with other surface temperatures gives

$$T_s = 34.3^\circ\text{C} \quad \text{and} \quad \dot{Q} = 1988 \text{ W}$$

Heat loss and its cost without insulation was determined in the Prob. 14-39 to be 27.388 kW and \$11,550. Then the reduction in the heat losses becomes

$$\dot{Q}_{\text{saved}} = 27.388 - 1.988 \cong 25.40 \text{ kW} \quad \text{or} \quad 25.388/27.40 = 0.927 \quad (92.7\%)$$

Therefore, the money saved by insulation will be $0.927 \times (\$11,550/\text{yr}) = \mathbf{\$10,700/\text{yr}}$ which will pay for the cost of \$750 in $\$750/(\$10,640/\text{yr}) = 0.0701 \text{ year} = \mathbf{26 \text{ days}}$.

14-42 A circuit board containing square chips is mounted on a vertical wall in a room. The surface temperature of the chips is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The heat transfer from the back side of the circuit board is negligible.

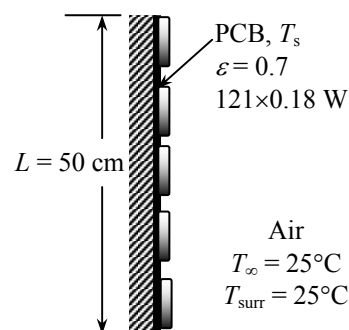
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (35 + 25)/2 = 30^\circ\text{C}$ are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7282$$

$$\beta = \frac{1}{T_f} = \frac{1}{(30 + 273)\text{K}} = 0.0033 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 35°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the height of the board, $L_c = L = 0.5 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.0033 \text{ K}^{-1})(35 - 25 \text{ K})(0.5 \text{ m})^3}{(1.608 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7282) = 1.140 \times 10^8$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.140 \times 10^8)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7282} \right)^{9/16} \right]^{8/27}} \right\}^2 = 63.72$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{0.5 \text{ m}} (63.72) = 3.30 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (0.5 \text{ m})^2 = 0.25 \text{ m}^2$$

Considering both natural convection and radiation, the total rate of heat loss can be expressed as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \epsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ (121 \times 0.18) \text{ W} &= (3.30 \text{ W/m}^2\cdot^\circ\text{C})(0.25 \text{ m}^2)(T_s - 25)^\circ\text{C} \\ &\quad + (0.7)(0.25 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] \end{aligned}$$

Its solution is

$$T_s = 36.2^\circ\text{C}$$

which is sufficiently close to the assumed value in the evaluation of properties and h . Therefore, there is no need to repeat calculations by reevaluating the properties and h at the new film temperature.

14-43 A circuit board containing square chips is positioned horizontally in a room. The surface temperature of the chips is to be determined for two orientations.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The heat transfer from the back side of the circuit board is negligible.

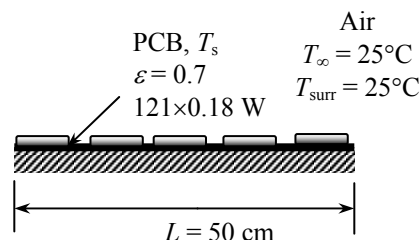
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (35 + 25)/2 = 30^\circ\text{C}$ are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7282$$

$$\beta = \frac{1}{T_f} = \frac{1}{(30 + 273)\text{K}} = 0.0033 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 35°C for the evaluation of the properties and h . The characteristic length for both cases is determined from

$$L_c = \frac{A_s}{p} = \frac{(0.5 \text{ m})^2}{2[(0.5 \text{ m}) + (0.5 \text{ m})]} = 0.125 \text{ m}.$$

Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.0033 \text{ K}^{-1})(35 - 25 \text{ K})(0.125 \text{ m})^3}{(1.608 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7282) = 1.781 \times 10^6$$

(a) Chips (hot surface) facing up:

$$\text{Nu} = 0.54 \text{Ra}^{1/4} = 0.54(1.781 \times 10^6)^{1/4} = 19.73$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{0.125 \text{ m}} (19.73) = 4.08 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (0.5 \text{ m})^2 = 0.25 \text{ m}^2$$

Considering both natural convection and radiation, the total rate of heat loss can be expressed as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \epsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ (121 \times 0.18) \text{ W} &= (4.08 \text{ W/m}^2\cdot^\circ\text{C})(0.25 \text{ m}^2)(T_s - 25)^\circ\text{C} \\ &\quad + (0.7)(0.25 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] \end{aligned}$$

Its solution is $T_s = 35.2^\circ\text{C}$

which is sufficiently close to the assumed value. Therefore, there is no need to repeat calculations.

(b) Chips (hot surface) facing up:

$$\text{Nu} = 0.27 \text{Ra}^{1/4} = 0.27(1.781 \times 10^6)^{1/4} = 9.863$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{0.125 \text{ m}} (9.863) = 2.04 \text{ W/m}^2\cdot^\circ\text{C}$$

Considering both natural convection and radiation, the total rate of heat loss can be expressed as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \epsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ (121 \times 0.18) \text{ W} &= (2.04 \text{ W/m}^2\cdot^\circ\text{C})(0.25 \text{ m}^2)(T_s - 25)^\circ\text{C} \\ &\quad + (0.7)(0.25 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] \end{aligned}$$

Its solution is $T_s = 38.3^\circ\text{C}$

which is identical to the assumed value in the evaluation of properties and h . Therefore, there is no need to repeat calculations.

14-44 CD EES It is proposed that the side surfaces of a cubic industrial furnace be insulated for \$550 in order to reduce the heat loss by 90 percent. The thickness of the insulation and the payback period of the insulation to pay for itself from the energy it saves are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

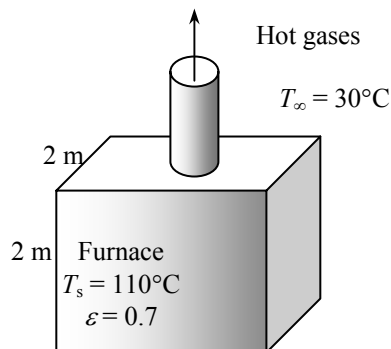
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (110 + 30)/2 = 70^\circ\text{C}$ are (Table A-22)

$$k = 0.02881 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.995 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7177$$

$$\beta = \frac{1}{T_f} = \frac{1}{(70 + 273)\text{K}} = 0.002915 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the height of the furnace, $L_c = L = 2 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002915 \text{ K}^{-1})(110 - 30 \text{ K})(2 \text{ m})^3}{(1.995 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7177) = 3.301 \times 10^{10}$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(3.301 \times 10^{10})^{1/6}}{\left[1 + \left(\frac{0.492}{0.7177} \right)^{9/16} \right]^{8/27}} \right\}^2 = 369.2$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.02881 \text{ W/m}\cdot^\circ\text{C}}{2 \text{ m}} (369.2) = 5.318 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = 4(2 \text{ m})^2 = 16 \text{ m}^2$$

Then the heat loss by combined natural convection and radiation becomes

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (5.318 \text{ W/m}^2\cdot^\circ\text{C})(16 \text{ m}^2)(110 - 30)^\circ\text{C} \\ &\quad + (0.7)(16 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(110 + 273 \text{ K})^4 - (30 + 273 \text{ K})^4] \\ &= 15,119 \text{ W} \end{aligned}$$

Noting that insulation will reduce the heat losses by 90%, the rate of heat loss after insulation will be

$$\dot{Q}_{\text{saved}} = 0.9 \dot{Q}_{\text{no insulation}} = 0.9 \times 15,119 \text{ W} = 13,607 \text{ W}$$

$$\dot{Q}_{\text{loss}} = (1 - 0.9) \dot{Q}_{\text{no insulation}} = 0.1 \times 15,119 \text{ W} = 1512 \text{ W}$$

The furnace operates continuously and thus 8760 h. Then the amount of energy and money the insulation will save becomes

$$\text{Energy saved} = \dot{Q}_{\text{saved}} \Delta t = \frac{13.607 \text{ kJ/s}}{0.78} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) (8760 \times 3600 \text{ s/yr}) = 5215 \text{ therms/yr}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (5215 \text{ therms})(\$0.55 / \text{therm}) = \$2868$$

Therefore, the money saved by insulation will pay for the cost of \$550 in

$$550 / (\$2868/\text{yr}) = 0.1918 \text{ yr} = \mathbf{70 \text{ days.}}$$

Insulation will lower the outer surface temperature, the Rayleigh and Nusselt numbers, and thus the convection heat transfer coefficient. For the evaluation of the heat transfer coefficient, we assume the surface temperature in this case to be 50°C. The properties of air at the film temperature of $(T_s + T_\infty)/2 = (50 + 30)/2 = 40^\circ\text{C}$ are (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7255$$

$$\beta = \frac{1}{T_f} = \frac{1}{(40 + 273)\text{K}} = 0.003195 \text{ K}^{-1}$$

Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003195 \text{ K}^{-1})(50 - 30 \text{ K})(2 \text{ m})^3}{(1.702 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7255) = 1.256 \times 10^{10}$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.256 \times 10^{10})^{1/6}}{\left[1 + \left(\frac{0.492}{0.7255} \right)^{9/16} \right]^{8/27}} \right\}^2 = 272.0$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{2 \text{ m}} (272.0) = 3.620 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = 4 \times (2 \text{ m})(2 + 2t_{\text{insul}}) \text{ m}$$

The total rate of heat loss from the outer surface of the insulated furnace by convection and radiation becomes

$$\begin{aligned} \dot{Q} &= \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ 1512 \text{ W} &= (3.620 \text{ W/m}^2\cdot^\circ\text{C})A_s(T_s - 30)^\circ\text{C} \\ &\quad + (0.7)A_s(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273 \text{ K})^4 - (30 + 273 \text{ K})^4] \end{aligned}$$

In steady operation, the heat lost by the side surfaces of the pipe must be equal to the heat lost from the exposed surface of the insulation by convection and radiation, which must be equal to the heat conducted through the insulation. Therefore,

$$\dot{Q} = \dot{Q}_{\text{insulation}} = kA_s \frac{(T_{\text{furnace}} - T_s)}{t_{\text{ins}}} \rightarrow 1512 \text{ W} = (0.038 \text{ W/m}\cdot^\circ\text{C})A_s \frac{(110 - T_s)^\circ\text{C}}{t_{\text{insul}}}$$

Solving the two equations above by trial-and-error (or better yet, an equation solver) gives

$$T_s = 41.1^\circ\text{C} \text{ and } t_{\text{insul}} = 0.0285 \text{ m} = \mathbf{2.85 \text{ cm}}$$

14-45 A cylindrical propane tank is exposed to calm ambient air. The propane is slowly vaporized due to a crack developed at the top of the tank. The time it will take for the tank to empty is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Radiation heat transfer is negligible.

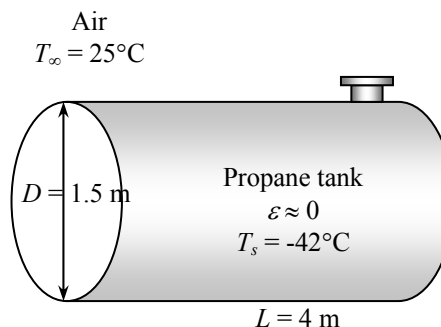
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (-42 + 25)/2 = -8.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02299 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.265 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7383$$

$$\beta = \frac{1}{T_f} = \frac{1}{(-8.5 + 273)\text{K}} = 0.003781 \text{ K}^{-1}$$



Analysis The tank gains heat through its cylindrical surface as well as its circular end surfaces. For convenience, we take the heat transfer coefficient at the end surfaces of the tank to be the same as that of its side surface. (The alternative is to treat the end surfaces as a vertical plate, but this will double the amount of calculations without providing much improvement in accuracy since the area of the end surfaces is much smaller and it is circular in shape rather than being rectangular). The characteristic length in this case is the outer diameter of the tank, $L_c = D = 1.5 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003781 \text{ K}^{-1})[(25 - (-42))\text{K}](1.5 \text{ m})^3}{(1.265 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7383) = 3.869 \times 10^{10}$$

$$\text{Nu} = \left\{ 0.6 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387(3.869 \times 10^{10})^{1/6}}{\left[1 + (0.559 / 0.7383)^{9/16} \right]^{8/27}} \right\}^2 = 374.1$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.02299 \text{ W/m}\cdot^\circ\text{C}}{1.5 \text{ m}} (374.1) = 5.733 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL + 2\pi D^2 / 4 = \pi(1.5 \text{ m})(4 \text{ m}) + 2\pi(1.5 \text{ m})^2 / 4 = 22.38 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_\infty - T_s) = (5.733 \text{ W/m}^2\cdot^\circ\text{C})(22.38 \text{ m}^2)[(25 - (-42))^\circ\text{C}] = 8598 \text{ W}$$

The total mass and the rate of evaporation of propane are

$$m = \rho V = \rho \frac{\pi D^2}{4} L = (581 \text{ kg/m}^3) \frac{\pi(1.5 \text{ m})^2}{4} (4 \text{ m}) = 4107 \text{ kg}$$

$$\dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{8.598 \text{ kJ/s}}{425 \text{ kJ/kg}} = 0.02023 \text{ kg/s}$$

and it will take

$$\Delta t = \frac{m}{\dot{m}} = \frac{4107 \text{ kg}}{0.02023 \text{ kg/s}} = 202,996 \text{ s} = \mathbf{56.4 \text{ hours}}$$

for the propane tank to empty.

14-46E The average surface temperature of a human head is to be determined when it is not covered.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The head can be approximated as a 12-in.-diameter sphere.

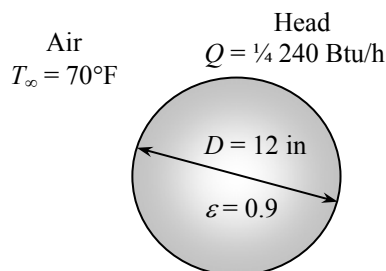
Properties The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 90°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (90 + 70)/2 = 80^\circ\text{F}$ are (Table A-22E)

$$k = 0.01481 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 1.697 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7290$$

$$\beta = \frac{1}{T_f} = \frac{1}{(80 + 460)\text{R}} = 0.001852 \text{ R}^{-1}$$



Analysis The characteristic length for a spherical object is $L_c = D/6 = 12/6 = 2 \text{ in} = 1/6 \text{ ft}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001852 \text{ R}^{-1})(90 - 70 \text{ R})(1/6 \text{ ft})^3}{(1.697 \times 10^{-4} \text{ ft}^2/\text{s})^2} (0.7290) = 3.019 \times 10^7$$

$$\text{Nu} = 2 + \frac{0.589 \text{Ra}^{1/4}}{\left[1 + \left(\frac{0.469}{\text{Pr}}\right)^{9/16}\right]^{4/9}} = 2 + \frac{0.589(3.019 \times 10^7)^{1/4}}{\left[1 + \left(\frac{0.469}{0.7290}\right)^{9/16}\right]^{4/9}} = 35.79$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.01481 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{1/6 \text{ ft}} (35.79) = 0.5300 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_s = \pi D^2 = \pi (1 \text{ ft})^2 = 3.142 \text{ ft}^2$$

Considering both natural convection and radiation, the total rate of heat loss can be written as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ (240/4 \text{ Btu/h}) &= (0.5300 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(3.142 \text{ ft}^2)(T_s - 70)^\circ\text{F} \\ &\quad + (0.9)(3.142 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)[(T_s + 460 \text{ R})^4 - (70 + 460 \text{ R})^4] \end{aligned}$$

Its solution is

$$T_s = \mathbf{82.9^\circ\text{F}}$$

which is sufficiently close to the assumed value in the evaluation of the properties and h . Therefore, there is no need to repeat calculations.

14-47 The equilibrium temperature of a light glass bulb in a room is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The light bulb is approximated as an 8-cm-diameter sphere.

Properties The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 170°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (170 + 25)/2 = 97.5^\circ\text{C}$ are (Table A-22)

$$k = 0.03077 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 2.279 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7116$$

$$\beta = \frac{1}{T_f} = \frac{1}{(97.5 + 273)\text{K}} = 0.002699 \text{ K}^{-1}$$

Analysis The characteristic length in this case is $L_c = D = 0.08 \text{ m}$. Then,

$$\begin{aligned} Ra &= \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} \\ &= \frac{(9.81 \text{ m/s}^2)(0.002699 \text{ K}^{-1})(170 - 25 \text{ K})(0.08 \text{ m})^3}{(2.279 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7116) = 2.694 \times 10^6 \end{aligned}$$

$$Nu = 2 + \frac{0.589 Ra^{1/4}}{\left[1 + (0.469 / \text{Pr})^{9/16}\right]^{4/9}} = 2 + \frac{0.589(2.694 \times 10^6)^{1/4}}{\left[1 + (0.469 / 0.7116)^{9/16}\right]^{4/9}} = 20.42$$

Then

$$h = \frac{k}{D} Nu = \frac{0.03077 \text{ W/m}\cdot^\circ\text{C}}{0.08 \text{ m}} (20.42) = 7.854 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi D^2 = \pi(0.08 \text{ m})^2 = 0.02011 \text{ m}^2$$

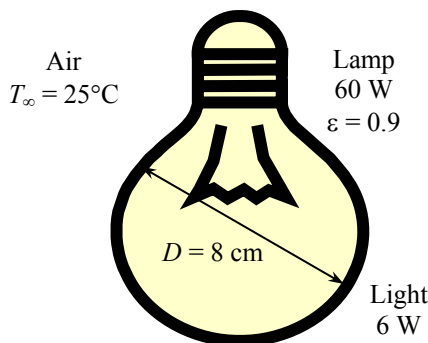
Considering both natural convection and radiation, the total rate of heat loss can be written as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ (0.90 \times 60) \text{ W} &= (7.854 \text{ W/m}^2\cdot^\circ\text{C})(0.02011 \text{ m}^2)(T_s - 25)^\circ\text{C} \\ &\quad + (0.9)(0.02011 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273)^4 - (25 + 273 \text{ K})^4] \end{aligned}$$

Its solution is

$$T_s = \mathbf{169.4^\circ\text{C}}$$

which is sufficiently close to the value assumed in the evaluation of properties and h . Therefore, there is no need to repeat calculations.



14-48 A vertically oriented cylindrical hot water tank is located in a bathroom. The rate of heat loss from the tank by natural convection and radiation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The temperature of the outer surface of the tank is constant.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (44 + 20)/2 = 32^\circ\text{C}$ are (Table A-22)

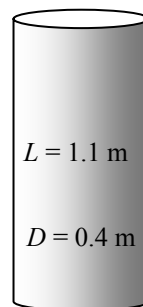
$$k = 0.02603 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.627 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7276$$

$$\beta = \frac{1}{T_f} = \frac{1}{(32 + 273)\text{K}} = 0.003279 \text{ K}^{-1}$$

Air
 $T_\infty = 20^\circ\text{C}$



Tank
 $T_s = 44^\circ\text{C}$
 $\varepsilon = 0.4$

Analysis The characteristic length in this case is the height of the cylinder, $L_c = L = 1.1 \text{ m}$. Then,

$$\text{Gr} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} = \frac{(9.81 \text{ m/s}^2)(0.003279 \text{ K}^{-1})(44 - 20 \text{ K})(1.1 \text{ m})^3}{(1.627 \times 10^{-5} \text{ m}^2/\text{s})^2} = 3.883 \times 10^9$$

A vertical cylinder can be treated as a vertical plate when

$$D (= 0.4 \text{ m}) \geq \frac{35L}{\text{Gr}^{1/4}} = \frac{35(1.1 \text{ m})}{(3.883 \times 10^9)^{1/4}} = 0.1542 \text{ m}$$

which is satisfied. That is, the Nusselt number relation for a vertical plate can be used for the side surfaces. For the top and bottom surfaces we use the relevant Nusselt number relations. First, for the side surfaces,

$$\text{Ra} = \text{GrPr} = (3.883 \times 10^9)(0.7276) = 2.825 \times 10^9$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(2.825 \times 10^9)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7276} \right)^{9/16} \right]^{8/27}} \right\}^2 = 170.2$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02603 \text{ W/m}\cdot^\circ\text{C}}{1.1 \text{ m}} (170.2) = 4.027 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.4 \text{ m})(1.1 \text{ m}) = 1.382 \text{ m}^2$$

$$\dot{Q}_{\text{side}} = hA_s(T_s - T_\infty) = (4.027 \text{ W/m}^2\cdot^\circ\text{C})(1.382 \text{ m}^2)(44 - 20)^\circ\text{C} = 133.6 \text{ W}$$

For the top surface,

$$L_c = \frac{A_s}{p} = \frac{\pi D^2 / 4}{\pi D} = \frac{D}{4} = \frac{0.4 \text{ m}}{4} = 0.1 \text{ m}$$

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003279 \text{ K}^{-1})(44 - 20 \text{ K})(0.1 \text{ m})^3}{(1.627 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7276) = 2.123 \times 10^6$$

$$\text{Nu} = 0.54\text{Ra}^{1/4} = 0.54(2.123 \times 10^6)^{1/4} = 20.61$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.02603 \text{ W/m}\cdot^\circ\text{C}}{0.1 \text{ m}} (20.61) = 5.365 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi D^2 / 4 = \pi(0.4 \text{ m})^2 / 4 = 0.1257 \text{ m}^2$$

$$\dot{Q}_{\text{top}} = hA_s(T_s - T_\infty) = (5.365 \text{ W/m}^2\cdot^\circ\text{C})(0.1257 \text{ m}^2)(44 - 20)^\circ\text{C} = 16.2 \text{ W}$$

For the bottom surface,

$$Nu = 0.27Ra^{1/4} = 0.27(2.123 \times 10^6)^{1/4} = 10.31$$

$$h = \frac{k}{L_c} Nu = \frac{0.02603 \text{ W/m} \cdot ^\circ\text{C}}{0.1 \text{ m}} (10.31) = 2.683 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q}_{\text{bottom}} = hA_s(T_s - T_\infty) = (2.683 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1257 \text{ m}^2)(44 - 20)^\circ\text{C} = 8.1 \text{ W}$$

The total heat loss by natural convection is

$$\dot{Q}_{\text{conv}} = \dot{Q}_{\text{side}} + \dot{Q}_{\text{top}} + \dot{Q}_{\text{bottom}} = 133.6 + 16.2 + 8.1 = \mathbf{157.9 \text{ W}}$$

The radiation heat loss from the tank is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.4)(1.382 + 0.1257 + 0.1257 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) \left[(44 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4 \right] \\ &= \mathbf{101.1 \text{ W}} \end{aligned}$$

14-49 A rectangular container filled with cold water is gaining heat from its surroundings by natural convection and radiation. The water temperature in the container after 3 hours and the average rate of heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The heat transfer coefficient at the top and bottom surfaces is the same as that on the side surfaces.

Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (10 + 24)/2 = 17^\circ\text{C}$ are (Table A-22)

$$k = 0.02491 \text{ W/m}\cdot^\circ\text{C}$$

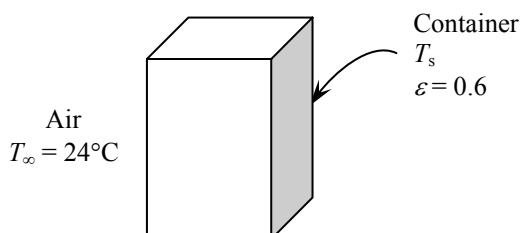
$$\nu = 1.489 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7317$$

$$\beta = \frac{1}{T_f} = \frac{1}{(17 + 273)\text{K}} = 0.003448 \text{ K}^{-1}$$

The properties of water at 2°C are (Table A-15)

$$\rho = 1000 \text{ kg/m}^3 \text{ and } c_p = 4214 \text{ J/kg}\cdot^\circ\text{C}$$



Analysis We first evaluate the heat transfer coefficient on the side surfaces. The characteristic length in this case is the height of the container,

$L_c = L = 0.28 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003448 \text{ K}^{-1})(24 - 10 \text{ K})(0.28 \text{ m})^3}{(1.489 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7317) = 1.133 \times 10^7$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.133 \times 10^7)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7317} \right)^{9/16} \right]^{8/27}} \right\}^2 = 30.52$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02491 \text{ W/m}\cdot^\circ\text{C}}{0.28 \text{ m}} (30.52) = 4.224 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = 2(0.28 \times 0.18 + 0.28 \times 0.18 + 0.18 \times 0.18) = 0.2664 \text{ m}^2$$

The rate of heat transfer can be expressed as

$$\begin{aligned} \dot{Q} &= \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s \left(T_\infty - \frac{T_1 + T_2}{2} \right) + \varepsilon \sigma A_s \left[T_{\text{surr}}^4 - \left(\frac{T_1 + T_2}{2} \right)^4 \right] \\ &= (4.224 \text{ W/m}^2\cdot^\circ\text{C})(0.2664 \text{ m}^2) \left[297 - \left(\frac{275 + T_2}{2} \right) \right] \\ &\quad + (0.6)(0.2664 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) \left[297^4 - \left(\frac{275 + T_2}{2} \right)^4 \right] \end{aligned} \quad (\text{Eq. 1})$$

where $(T_1 + T_2)/2$ is the average temperature of water (or the container surface). The mass of water in the container is

$$m = \rho V = (1000 \text{ kg/m}^3)(0.28 \times 0.18 \times 0.18) \text{ m}^3 = 9.072 \text{ kg}$$

Then the amount of heat transfer to the water is

$$Q = mc_p(T_2 - T_1) = (9.072 \text{ kg})(4214 \text{ J/kg}\cdot^\circ\text{C})(T_2 - 275)^\circ\text{C} = 38,229(T_2 - 275)$$

The average rate of heat transfer can be expressed as

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{38,229(T_2 - 275)}{3 \times 3600 \text{ s}} = 3.53976(T_2 - 275) \quad (\text{Eq. 2})$$

Setting Eq. 1 and Eq. 2 equal to each other, we obtain the final water temperature.

$$T_2 = 284.7 \text{ K} = \mathbf{11.7^\circ\text{C}}$$

We could repeat the solution using air properties at the new film temperature using this value to increase the accuracy. However, this would only affect the heat transfer value somewhat, which would not have significant effect on the final water temperature. The average rate of heat transfer can be determined from Eq. 2

$$\dot{Q} = 3.53976(11.7 - 2) = \mathbf{34.3 \text{ W}}$$

14-50 EES Prob. 14-49 is reconsidered. The water temperature in the container as a function of the heating time is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

height=0.28 [m]
 L=0.18 [m]
 w=0.18 [m]
 T_infinity=24 [C]
 T_w1=2 [C]
 epsilon=0.6
 T_surr=T_infinity
 time=3 [h]

"PROPERTIES"

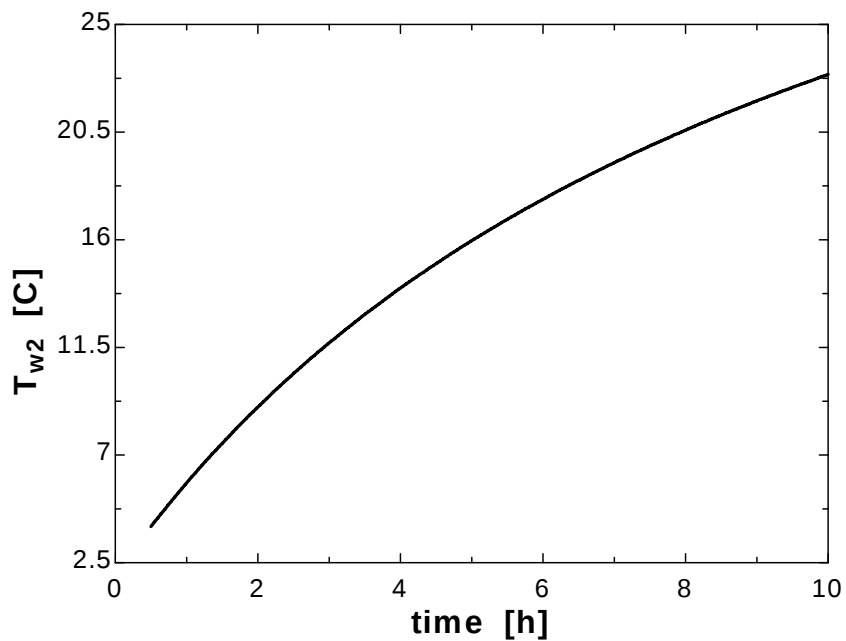
Fluid\$='air'
 k=Conductivity(Fluid\$, T=T_film)
 Pr=Prandtl(Fluid\$, T=T_film)
 rho=Density(Fluid\$, T=T_film, P=101.3)
 mu=Viscosity(Fluid\$, T=T_film)
 nu=mu/rho
 beta=1/(T_film+273)
 T_film=1/2*(T_w_ave+T_infinity)
 T_w_ave=1/2*(T_w1+T_w2)
 rho_w=Density(water, T=T_w_ave, P=101.3)
 c_p_w=CP(water, T=T_w_ave, P=101.3)*Convert(kJ/kg-C, J/kg-C)
 sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzmann constant"
 g=9.807 [m/s^2] "gravitational acceleration"

"ANALYSIS"

delta=height
 Ra=(g*beta*(T_infinity-T_w_ave)*delta^3)/nu^2*Pr
 Nusselt=0.59*Ra^0.25
 h=k/delta*Nusselt
 A=2*(height*L+height*w+w*L)
 Q_dot=h*A*(T_infinity-T_w_ave)+epsilon*A*sigma*((T_surr+273)^4-(T_w_ave+273)^4)

 m_w=rho_w*V_w
 V_w=height*L*w
 Q=m_w*c_p_w*(T_w2-T_w1)
 Q_dot=Q/(time*Convert(h, s))

time [h]	T_{w2} [C]
0.5	4.013
1	5.837
1.5	7.496
2	9.013
2.5	10.41
3	11.69
3.5	12.88
4	13.98
4.5	15
5	15.96
5.5	16.85
6	17.69
6.5	18.48
7	19.22
7.5	19.92
8	20.59
8.5	21.21
9	21.81
9.5	22.37
10	22.91



14-51 A room is to be heated by a cylindrical coal-burning stove. The surface temperature of the stove and the amount of coal burned during a 14-h period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 The temperature of the outer surface of the stove is constant. 5 The heat transfer from the bottom surface is negligible. 6 The heat transfer coefficient at the top surface is the same as that on the side surface.

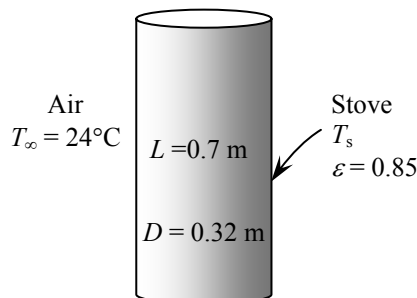
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (130 + 24)/2 = 77^\circ\text{C}$ are (Table A-22)

$$k = 0.02931 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 2.066 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7161$$

$$\beta = \frac{1}{T_f} = \frac{1}{(77 + 273)\text{K}} = 0.002857 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the height of the cylinder, $L_c = L = 0.7 \text{ m}$. Then,

$$\text{Gr} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} = \frac{(9.81 \text{ m/s}^2)(0.002857 \text{ K}^{-1})(130 - 24 \text{ K})(0.70 \text{ m})^3}{(2.066 \times 10^{-5} \text{ m}^2/\text{s})^2} = 2.387 \times 10^9$$

A vertical cylinder can be treated as a vertical plate when

$$D (= 0.32 \text{ m}) \geq \frac{35L}{\text{Gr}^{1/4}} = \frac{35(0.7 \text{ m})}{(2.387 \times 10^9)^{1/4}} = 0.1108 \text{ m}$$

which is satisfied. That is, the Nusselt number relation for a vertical plate can be used for side surfaces.

$$\text{Ra} = \text{GrPr} = (2.387 \times 10^9)(0.7161) = 1.709 \times 10^9$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.709 \times 10^9)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7161} \right)^{9/16} \right]^{8/27}} \right\}^2 = 145.2$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02931 \text{ W/m}\cdot^\circ\text{C}}{0.7 \text{ m}} (145.2) = 6.080 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL + \pi D^2 / 4 = \pi(0.32 \text{ m})(0.7 \text{ m}) + \pi(0.32 \text{ m})^2 / 4 = 0.7841 \text{ m}^2$$

Then the surface temperature of the stove is determined from

$$\begin{aligned} \dot{Q} &= \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s(T_s^4 - T_{\text{surr}}^4) \\ 1500 \text{ W} &= (6.080 \text{ W/m}^2\cdot^\circ\text{C})(0.7841 \text{ m}^2)(T_s - 297) \\ &\quad + (0.85)(0.7841 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)(T_s^4 - 287^4) \\ T_s &= 419.6 \text{ K} = \mathbf{146.6^\circ\text{C}} \end{aligned}$$

The amount of coal used is determined from

$$\begin{aligned} Q &= \dot{Q}\Delta t = (1.5 \text{ kJ/s})(14 \text{ h/day} \times 3600 \text{ s/h}) = 75,600 \text{ kJ} \\ m_{\text{coal}} &= \frac{Q/\eta}{HV} = \frac{(75,600 \text{ kJ})/0.65}{30,000 \text{ kJ/kg}} = \mathbf{3.88 \text{ kg}} \end{aligned}$$

14-52 Water in a tank is to be heated by a spherical heater. The heating time is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The temperature of the outer surface of the sphere is constant.

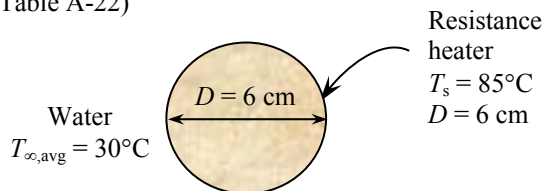
Properties Using the average temperature for water $(15+45)/2=30^\circ\text{C}$ as the fluid temperature, the properties of water at the film temperature of $(T_s+T_\infty)/2 = (85+30)/2 = 57.5^\circ\text{C}$ are (Table A-22)

$$k = 0.6515 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 0.493 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 3.12$$

$$\beta = 0.501 \times 10^{-3} \text{ K}^{-1}$$



Also, the properties of water at 30°C are (Table A-22)

$$\rho = 996 \text{ kg/m}^3 \text{ and } c_p = 4178 \text{ J/kg}\cdot^\circ\text{C}$$

Analysis The characteristic length in this case is $L_c = D = 0.06 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.501 \times 10^{-3} \text{ K}^{-1})(85 - 30 \text{ K})(0.06 \text{ m})^3}{(0.493 \times 10^{-6} \text{ m}^2/\text{s})^2} (3.12) = 7.495 \times 10^8$$

$$\text{Nu} = 2 + \frac{0.589 \text{Ra}^{1/4}}{\left[1 + (0.469 / \text{Pr})^{9/16}\right]^{4/9}} = 2 + \frac{0.589(7.495 \times 10^8)^{1/4}}{\left[1 + (0.469 / 3.12)^{9/16}\right]^{4/9}} = 87.44$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.6515 \text{ W/m}\cdot^\circ\text{C}}{0.06 \text{ m}} (87.44) = 949.5 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi D^2 = \pi (0.06 \text{ m})^2 = 0.01131 \text{ m}^2$$

The rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) = (949.5 \text{ W/m}^2\cdot^\circ\text{C})(0.01131 \text{ m}^2)(85 - 30) = 590.6 \text{ W}$$

The mass of water in the container is

$$m = \rho V = (996 \text{ kg/m}^3)(0.040 \text{ m}^3) = 39.84 \text{ kg}$$

The amount of heat transfer to the water is

$$Q = mc_p(T_2 - T_1) = (39.84 \text{ kg})(4178 \text{ J/kg}\cdot^\circ\text{C})(45 - 15)^\circ\text{C} = 4.994 \times 10^6 \text{ J}$$

Then the time the heater should be on becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{4.994 \times 10^6 \text{ J}}{590.6 \text{ J/s}} = 8456 \text{ s} = \mathbf{2.35 \text{ hours}}$$

Natural Convection from Finned Surfaces and PCBs

14-53C Finned surfaces are frequently used in practice to enhance heat transfer by providing a larger heat transfer surface area. Finned surfaces are referred to as heat sinks in the electronics industry since they provide a medium to which the waste heat generated in the electronic components can be transferred effectively.

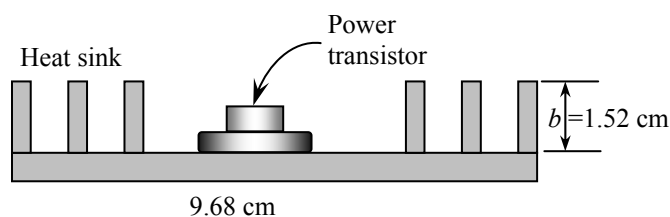
14-54C A heat sink with closely packed fins will have greater surface area for heat transfer, but smaller heat transfer coefficient because of the extra resistance the additional fins introduce to fluid flow through the interfin passages.

14-55C Removing some of the fins on the heat sink will decrease heat transfer surface area, but will increase heat transfer coefficient. The decrease on heat transfer surface area more than offsets the increase in heat transfer coefficient, and thus heat transfer rate will decrease. In the second case, the decrease on heat transfer coefficient more than offsets the increase in heat transfer surface area, and thus heat transfer rate will again decrease.

14-56 An aluminum heat sink of rectangular profile oriented vertically is used to cool a power transistor. The average natural convection heat transfer coefficient is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Radiation heat transfer from the sink is negligible. 4 The entire sink is at the base temperature.

Analysis The total surface area of the heat sink is



$$A_{fins} = 2nLb = (2)(6)(0.0762 \text{ m})(0.0152 \text{ m}) + (2)(0.0048 \text{ m})(0.0762 \text{ m}) = 0.01463 \text{ m}^2$$

$$A_{unfinned} = (4)(0.0145 \text{ m})(0.0762 \text{ m}) + (0.0317 \text{ m})(0.0762 \text{ m}) = 0.006835 \text{ m}^2$$

$$A_{total} = A_{fins} + A_{unfinned} = 0.01463 + 0.006835 = 0.021465 \text{ m}^2$$

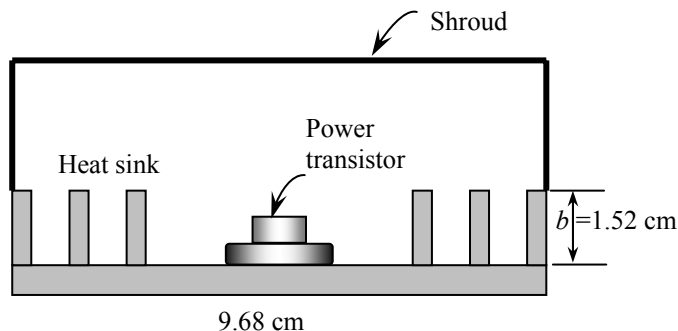
Then the average natural convection heat transfer coefficient becomes

$$\dot{Q} = hA_{total}(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A_{total}(T_s - T_\infty)} = \frac{15 \text{ W}}{(0.021465 \text{ m}^2)(120 - 22)^\circ\text{C}} = \mathbf{7.13 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

14-57 Aluminum heat sinks of rectangular profile oriented vertically are used to cool a power transistor. A shroud is placed very close to the tips of fins. The average natural convection heat transfer coefficient is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Radiation heat transfer from the sink is negligible. 4 The entire sink is at the base temperature.

Analysis The total surface area of the shrouded heat sink is



$$A_{fins} = 2nLb = (2)(6)(0.0762 \text{ m})(0.0152 \text{ m}) = 0.013898 \text{ m}^2$$

$$A_{unfanned} = (4)(0.0145 \text{ m})(0.0762 \text{ m}) + (0.0317 \text{ m})(0.0762 \text{ m}) = 0.006835 \text{ m}^2$$

$$A_{shroud} = (2)(0.0968 \text{ m})(0.0762 \text{ m}) = 0.014752 \text{ m}^2$$

$$A_{total} = A_{fins} + A_{unfanned} + A_{shroud} = 0.013898 + 0.006835 + 0.014752 = 0.035486 \text{ m}^2$$

Then the average natural convection heat transfer coefficient becomes

$$\dot{Q} = hA_{total}(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A_{total}(T_s - T_\infty)} = \frac{15 \text{ W}}{(0.035486 \text{ m}^2)(108 - 22)^\circ\text{C}} = \mathbf{4.92 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

14-58E A heat sink with equally spaced rectangular fins is to be used to cool a hot surface. The optimum fin spacing and the rate of heat transfer from the heat sink are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The atmospheric pressure at that location is 1 atm. 4 The thickness t of the fins is very small relative to the fin spacing S so that Eqs. 14-32 and 14-33 for optimum fin spacing are applicable.

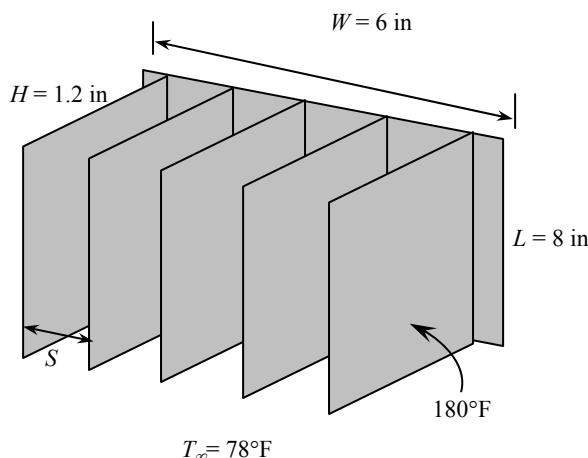
Properties The properties of air at 1 atm and 1 atm and the film temperature of $(T_s + T_\infty)/2 = (180 + 78)/2 = 129^\circ\text{F}$ are (Table A-22E)

$$k = 0.01597 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1975 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7217$$

$$\beta = \frac{1}{T_f} = \frac{1}{(129 + 460) \text{ R}} = 0.001698 \text{ R}^{-1}$$



Analysis The characteristic length in this case is the fin height, $L_c = L = 8 \text{ in}$. Then,

$$Ra = \frac{g\beta(T_1 - T_2)L^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001698 \text{ R}^{-1})(180 - 78 \text{ R})(8/12 \text{ ft})^3}{(0.1975 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7217) = 3.058 \times 10^7$$

The optimum fin spacing is

$$S = 2.714 \frac{L}{Ra^{1/4}} = 2.714 \frac{8/12 \text{ ft}}{(3.058 \times 10^7)^{1/4}} = 0.02433 \text{ ft} = \mathbf{0.292 \text{ in}}$$

The heat transfer coefficient for this optimum spacing case is

$$h = 1.307 \frac{k}{S} = 1.307 \frac{0.01597 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.02433 \text{ ft}} = 0.8578 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

The number of fins and the total heat transfer surface area is

$$n = \frac{w}{S + t} = \frac{6}{0.2916 + 0.08} = 16 \text{ fins}$$

$$A_s = 2nLH + ntL + 2ntH = 2 \times 16 \times (8/12 \text{ ft})(1.2/12 \text{ ft}) + 16 \times (0.08/12 \text{ ft})(8/12 \text{ ft}) + 2 \times 16 \times (0.08/12 \text{ ft})(1.2/12 \text{ ft}) = 2.226 \text{ ft}^2$$

Then the rate of natural convection heat transfer becomes

$$\dot{Q} = hA_s(T_s - T_\infty) = (0.862 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(2.226 \text{ ft}^2)(180 - 78)^\circ\text{F} = \mathbf{196 \text{ Btu/h}}$$

Discussion If the fin height is disregarded, the number of fins and the rate of heat transfer become

$$n = \frac{w}{s + t} \cong \frac{w}{s} = \frac{6}{0.2916} \cong 21 \text{ fins}$$

$$A_s = 2nLH = 2 \times 21 \times (8/12 \text{ ft})(1.2/12 \text{ ft}) = 2.8 \text{ ft}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (0.8578 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(2.8 \text{ ft}^2)(180 - 78)^\circ\text{F} = \mathbf{245 \text{ Btu/h}}$$

Therefore, the fin tip area is significant in this case.

14-59E EES Prob. 14-58E is reconsidered. The effect of the length of the fins in the vertical direction on the optimum fin spacing and the rate of heat transfer by natural convection is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

w_s=(6/12) [ft]
H_s=(8/12) [ft]
T_infinity=78 [F]
t_fin=(0.08/12) [ft]
L_fin=8 [in]
H_fin=(1.2/12) [ft]
T_s=180 [F]

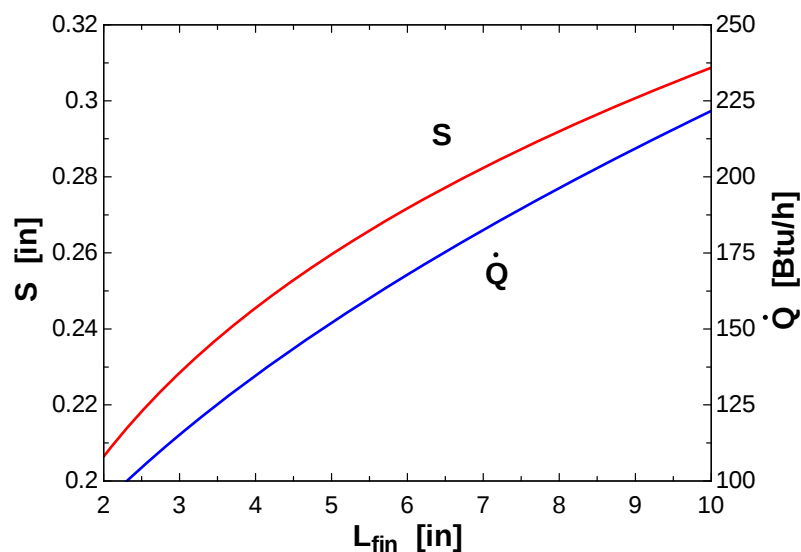
"PROPERTIES"

Fluid\$='air'
k=Conductivity(Fluid\$, T=T_film)
Pr=Prandtl(Fluid\$, T=T_film)
rho=Density(Fluid\$, T=T_film, P=14.7)
mu=Viscosity(Fluid\$, T=T_film)*Convert(lbm/ft-h, lbm/ft-s)
nu=mu/rho
beta=1/(T_film+460)
T_film=1/2*(T_s+T_infinity)
g=32.2 [ft/s^2] "gravitational acceleration"

"ANALYSIS"

L_fin_ft=L_fin*Convert(in, ft)
delta=L_fin_ft
Ra=(g*beta*(T_s-T_infinity)*delta^3)/nu^2*Pr
S_ft=2.714*L_fin_ft/Ra^0.25
S=S_ft*Convert(ft, in)
h=1.307*k/S_ft
n_fin=w_s/(S_ft+t_fin)
A=2*n_fin*L_fin_ft*H_fin+n_fin*t_fin*L_fin_ft+2*n_fin*t_fin*H_fin
Q_dot=h*A*(T_s-T_infinity)

L _{fin} [in]	S [in]	Q [Btu/h]
2	0.2065	92.73
2.5	0.2183	104.5
3	0.2285	115.3
3.5	0.2375	125.3
4	0.2455	134.7
4.5	0.2529	143.6
5	0.2596	152
5.5	0.2659	160.1
6	0.2717	167.9
6.5	0.2772	175.4
7	0.2824	182.6
7.5	0.2873	189.6
8	0.292	196.3
8.5	0.2964	202.9
9	0.3007	209.3
9.5	0.3048	215.6
10	0.3087	221.7



14-60 A heat sink with equally spaced rectangular fins is to be used to cool a hot surface. The optimum fin height and the rate of heat transfer from the heat sink are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The atmospheric pressure at that location is 1 atm.

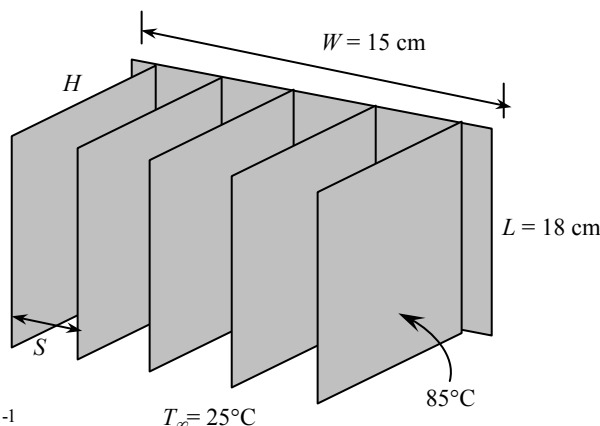
Properties The properties of air at 1 atm and 1 atm and the film temperature of $(T_s + T_\infty)/2 = (85 + 25)/2 = 55^\circ\text{C}$ are (Table A-22)

$$k = 0.02772 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.847 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7215$$

$$\beta = \frac{1}{T_f} = \frac{1}{(55 + 273)\text{K}} = 0.003049 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the height of the surface $L_c = L = 0.18 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003049 \text{ K}^{-1})(85 - 25 \text{ K})(0.18 \text{ m})^3}{(1.847 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7215) = 2.214 \times 10^7$$

The optimum fin spacing is

$$S = 2.714 \frac{L}{Ra^{1/4}} = 2.714 \frac{0.18 \text{ m}}{(2.214 \times 10^7)^{1/4}} = 0.007122 \text{ m} = 7.122 \text{ mm}$$

The heat transfer coefficient for this optimum fin spacing case is

$$h = 1.307 \frac{k}{S} = 1.307 \frac{0.02772 \text{ W/m}\cdot^\circ\text{C}}{0.007122 \text{ m}} = 5.087 \text{ W/m}^2\cdot^\circ\text{C}$$

The criteria for optimum fin height H in the literature is given by $H = \sqrt{hA_c / pk}$ (not in the text) where $A_c/p \cong t/2$ for rectangular fins. Therefore,

$$H = \sqrt{\frac{ht}{2k}} = \sqrt{\frac{(5.087 \text{ W/m}^2\cdot^\circ\text{C})(0.001 \text{ m})}{2 \times (177 \text{ W/m}\cdot^\circ\text{C})}} = 0.00379 \text{ m}$$

The number of fins and the total heat transfer surface area is

$$n = \frac{w}{S+t} \cong \frac{w}{s} = \frac{0.15}{0.007122} \cong 21 \text{ fins}$$

$$A_s = 2nLH = 2 \times 21 \times (0.18 \text{ m})(0.00379 \text{ m}) = 0.02865 \text{ m}^2$$

Then the rate of natural convection heat transfer becomes

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.087 \text{ W/m}^2\cdot^\circ\text{C})(0.02865 \text{ m}^2)(85 - 25)^\circ\text{C} = 8.75 \text{ W}$$

Natural Convection inside Enclosures

14-61C We would recommend putting the hot fluid into the upper compartment of the container. In this case no convection currents will develop in the enclosure since the lighter (hot) fluid will always be on top of the heavier (cold) fluid.

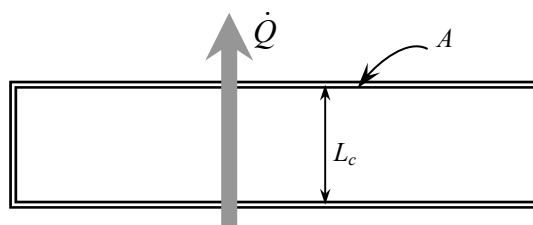
14-62C We would disagree with this recommendation since the air space introduces some thermal resistance to heat transfer. The thermal resistance of air space will be zero only when the convection coefficient approaches infinity, which is never the case. However, when the air space is eliminated, so is its thermal resistance.

14-63C Yes, dividing the air space into two compartments will retard air motion in the air space, and thus slow down heat transfer by natural convection. The vinyl sheet will also act as a radiation shield and reduce heat transfer by radiation.

14-64C The effective thermal conductivity of an enclosure represents the enhancement on heat transfer as result of convection currents relative to conduction. The ratio of the effective thermal conductivity to the ordinary thermal conductivity yields Nusselt number $Nu = k_{eff} / k$.

14-65 Conduction thermal resistance of a medium is expressed as $R = L / (kA)$. Thermal resistance of a rectangular enclosure can be expressed by replacing L with characteristic length of enclosure L_c , and thermal conductivity k with effective thermal conductivity k_{eff} to give

$$R = L_c / (k_{eff} A) = L_c / (kNuA)$$



14-66 The U-factors for the center-of-glass section of a double-pane window and a triple-pane window are to be determined. Also, the percentage decrease in total heat transfer when triple-pane window is used is to be estimated.

Assumptions 1 Steady operating conditions exist. 2 Heat transfer through the window is one-dimensional. 3 The thermal resistance of glass sheets is negligible.

Properties The thermal conductivity of air space is given to be $0.025 \text{ W/m}\cdot^\circ\text{C}$.

Analysis The convection heat transfer coefficient of the air space is determined from

$$h_{\text{conv}} = \frac{k}{L} Nu = \frac{0.025 \text{ W/m}\cdot^\circ\text{C}}{0.015 \text{ m}} (1.2) = 2 \text{ W/m}^2\cdot^\circ\text{C}$$

Noting that the radiation across the air space is of the same magnitude as the convection, the combined heat transfer coefficient of the space is

$$h_{\text{space}} = h_{\text{conv}} + h_{\text{rad}} = 2h_{\text{conv}} = 2(2 \text{ W/m}^2\cdot^\circ\text{C}) = 4 \text{ W/m}^2\cdot^\circ\text{C}$$

Disregarding the thermal resistance of glass sheets, which are small, the U-factor for the center region of a double pane window is determined from

$$\frac{1}{U_{\text{double}}} = \frac{1}{h_i} + \frac{1}{h_{\text{space}}} + \frac{1}{h_o} = \frac{1}{6} + \frac{1}{4} + \frac{1}{25} \longrightarrow U_{\text{double}} = \mathbf{2.190 \text{ W/m}^2\cdot^\circ\text{C}}$$

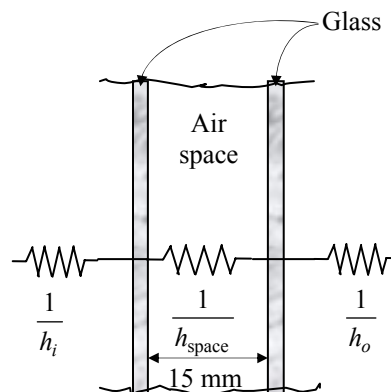
Noting that there are two air spaces, the U-factor for triple-pane window is

$$\frac{1}{U_{\text{triple}}} = \frac{1}{h_i} + \frac{1}{h_{\text{space}}} + \frac{1}{h_{\text{space}}} + \frac{1}{h_o} = \frac{1}{6} + \frac{1}{4} + \frac{1}{4} + \frac{1}{25} \longrightarrow U_{\text{triple}} = \mathbf{1.415 \text{ W/m}^2\cdot^\circ\text{C}}$$

Considering that about 70 percent of total heat transfer through a window is due to center-of-glass section, the percentage decrease in total heat transfer when triple-pane window is used in place of double-pane window is

$$\% \text{ Decrease} = (0.70) \frac{U_{\text{double}} - U_{\text{triple}}}{U_{\text{double}}} = (0.70) \frac{2.190 - 1.415}{2.190} = (0.70)(0.354) = 0.248 = \mathbf{24.8\%}$$

That is, triple-pane window decreases the heat transfer through the center region by 35.4 percent while the decrease for the entire window is 24.8 percent. The use of triple-pane window is usually not justified economically except for extremely cold regions.



14-67 Two glasses of a double pane window are maintained at specified temperatures. The fraction of heat transferred through the enclosure by radiation is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The air pressure in the enclosure is 1 atm.

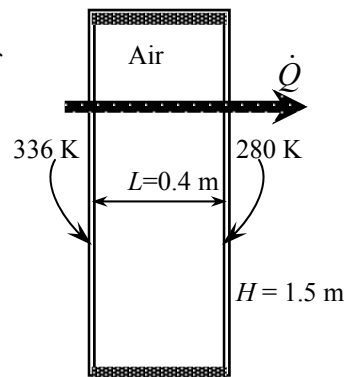
Properties The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (280 + 336)/2 = 308 \text{ K} = 35^\circ\text{C}$ are (Table A-22E)

$$k = 0.02625 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.655 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7268$$

$$\beta = \frac{1}{T_f} = \frac{1}{308 \text{ K}} = 0.003247 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the distance between the two glasses, $L_c = L = 0.4 \text{ m}$. Then,

$$Ra_L = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003247 \text{ K}^{-1})(336 - 280 \text{ K})(0.4 \text{ m})^3}{(1.655 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7268) = 3.029 \times 10^8$$

The aspect ratio of the geometry is $H/L = 1.5/0.4 = 3.75$. For this value of H/L the Nusselt number can be determined from

$$Nu = 0.22 \left(\frac{\text{Pr}}{0.2 + \text{Pr}} Ra \right)^{0.28} \left(\frac{H}{L} \right)^{-1/4} = 0.22 \left(\frac{0.7268}{0.2 + 0.7268} (3.029 \times 10^8) \right)^{0.28} \left(\frac{1.5}{0.4} \right)^{-1/4} = 35.00$$

Then,

$$A_s = H \times W = (1.5 \text{ m})(3 \text{ m}) = 4.5 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = kNu_s \frac{T_1 - T_2}{L} = (0.02625 \text{ W/m}\cdot^\circ\text{C})(35.00)(4.5 \text{ ft}^2) \frac{(336 - 280) \text{ K}}{0.4 \text{ m}} = 578.9 \text{ W}$$

The effective emissivity is

$$\frac{1}{\varepsilon_{\text{eff}}} = \frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1 = \frac{1}{0.15} + \frac{1}{0.90} - 1 = 6.778 \longrightarrow \varepsilon_{\text{eff}} = 0.1475$$

The rate of heat transfer by radiation is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon_{\text{eff}} A_s \sigma (T_1^4 - T_2^4) \\ &= (0.1475)(4.5 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(336 \text{ K})^4 - (280 \text{ K})^4] = 248.4 \text{ W} \end{aligned}$$

Then the fraction of heat transferred through the enclosure by radiation becomes

$$f_{\text{rad}} = \frac{\dot{Q}_{\text{rad}}}{\dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}}} = \frac{248.4}{578.9 + 248.4} = \mathbf{0.30}$$

14-68E Two glasses of a double pane window are maintained at specified temperatures. The rate of heat transfer through the window by natural convection and radiation, and the R-value of insulation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The air pressure in the enclosure is 1 atm.

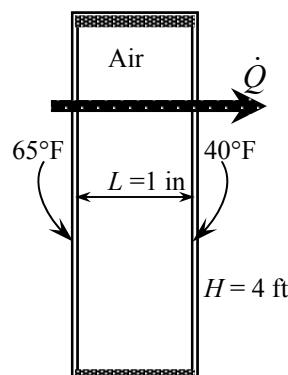
Properties The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (65 + 40)/2 = 52.5^\circ\text{F}$ are (Table A-22E)

$$k = 0.01415 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1548 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7332$$

$$\beta = \frac{1}{T_f} = \frac{1}{(52.5 + 460) \text{ R}} = 0.001951 \text{ R}^{-1}$$



Analysis (a) The characteristic length in this case is the distance between the two glasses, $L_c = L = 1 \text{ in}$. Then,

$$\text{Ra}_L = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001951 \text{ R}^{-1})(65 - 40 \text{ R})(1/12 \text{ ft})^3}{(0.1548 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7332) = 27,824$$

The aspect ratio of the geometry is $H/L = 4 \times 12/1 = 48$ (which is a little over 40, but still close enough for an approximate analysis). For these values of H/L and Ra_L , the Nusselt number can be determined from

$$\text{Nu} = 0.42 \text{Ra}^{1/4} \text{Pr}^{0.012} \left(\frac{H}{L} \right)^{-0.3} = 0.42(27,824)^{1/4} (0.7332)^{0.012} \left(\frac{4 \text{ ft}}{1/12 \text{ ft}} \right)^{-0.3} = 1.692$$

Then,

$$A_s = H \times W = (4 \text{ ft})(6 \text{ ft}) = 24 \text{ ft}^2$$

$$\dot{Q} = k \text{Nu} A_s \frac{T_1 - T_2}{L} = (0.01415 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F})(1.692)(24 \text{ ft}^2) \frac{(65 - 40)^\circ\text{F}}{(1/12) \text{ ft}} = \mathbf{172.4 \text{ Btu/h}}$$

(b) The rate of heat transfer by radiation is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_1^4 - T_2^4) \\ &= (0.82)(24 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h}\cdot\text{ft}^2 \cdot \text{R}^4)[(65 + 460 \text{ R})^4 - (40 + 460 \text{ R})^4] = \mathbf{454.3 \text{ Btu/h}} \end{aligned}$$

Then the total rate of heat transfer is

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{convection}} + \dot{Q}_{\text{rad}} = 172.4 + 454.3 = 626.7 \text{ Btu/h}$$

Then the effective thermal conductivity of the air, which also accounts for the radiation effect and the R-value become

$$\dot{Q}_{\text{total}} = k_{\text{eff}} A_s \frac{T_1 - T_2}{L} \longrightarrow k_{\text{eff}} = \frac{\dot{Q} L}{A_s (T_1 - T_2)} = \frac{(626.7 \text{ Btu/h})(1/12 \text{ ft})}{(24 \text{ ft}^2)(65 - 40)^\circ\text{F}} = 0.08704 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$R_{\text{value}} = \frac{L}{k_{\text{eff}}} = \frac{(1/12 \text{ ft})}{0.08704 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}} = 0.957 \text{ h}\cdot\text{ft}^2 \cdot ^\circ\text{F}/\text{Btu} = \mathbf{R - 0.96}$$

14-69E EES Prob. 14-68E is reconsidered. The effect of the air gap thickness on the rates of heat transfer by natural convection and radiation, and the R -value of insulation is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

H=4 [ft]
W=6 [ft]
L=1 [in]
T_1=65 [F]
T_2=40 [F]
epsilon_eff=0.82

"PROPERTIES"

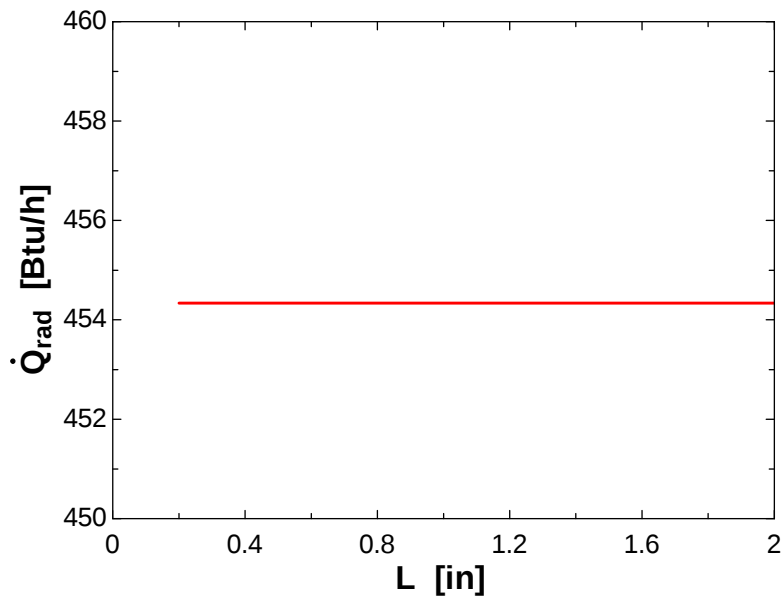
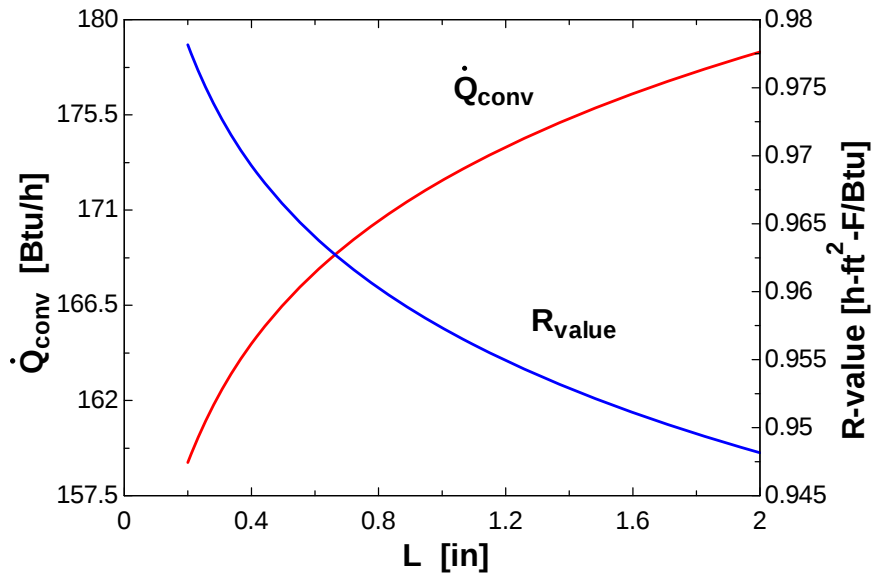
Fluid\$='air'
k=Conductivity(Fluid\$, T=T_ave)
Pr=Prandtl(Fluid\$, T=T_ave)
rho=Density(Fluid\$, T=T_ave, P=14.7)
mu=Viscosity(Fluid\$, T=T_ave)*Convert(lbm/ft-h, lbm/ft-s)
nu=mu/rho
beta=1/(T_ave+460)
T_ave=1/2*(T_1+T_2)
g=32.2 [ft/s^2]
sigma=0.1714E-8 [Btu/h-ft^2-R^4]

"ANALYSIS"

L_ft=L*Convert(in, ft)
Ra=(g*beta*(T_1-T_2)*L_ft^3)/nu^2*Pr
Ratio=H/L_ft
Nusselt=0.42*Ra^0.25*Pr^0.012*(H/L_ft)^(-0.3)
A=H*W
Q_dot_conv=k*Nusselt*A*(T_1-T_2)/L_ft
Q_dot_rad=epsilon_eff*A*sigma*((T_1+460)^4-(T_2+460)^4)
Q_dot_total=Q_dot_conv+Q_dot_rad
Q_dot_total=k_eff*A*(T_1-T_2)/L_ft
R_value=L_ft/k_eff

L [in]	Q _{conv} [Btu/h]	Q _{rad} [Btu/h]	R-value [h.ft ² .F/Btu]
0.2	159.1	454.3	0.9781
0.3	162.3	454.3	0.973
0.4	164.7	454.3	0.9693
0.5	166.5	454.3	0.9664
0.6	168.1	454.3	0.964
0.7	169.4	454.3	0.962
0.8	170.5	454.3	0.9603
0.9	171.5	454.3	0.9587
1	172.4	454.3	0.9573
1.1	173.2	454.3	0.9561
1.2	174	454.3	0.9549
1.3	174.7	454.3	0.9539
1.4	175.3	454.3	0.9529
1.5	175.9	454.3	0.952
1.6	176.5	454.3	0.9511

1.7	177	454.3	0.9503
1.8	177.5	454.3	0.9496
1.9	178	454.3	0.9488
2	178.5	454.3	0.9481



14-70 Two surfaces of a spherical enclosure are maintained at specified temperatures. The rate of heat transfer through the enclosure is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The air pressure in the enclosure is 1 atm.

Properties The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (350 + 275)/2 = 312.5 \text{ K} = 39.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02658 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.697 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7256$$

$$\beta = \frac{1}{T_f} = \frac{1}{312.5 \text{ K}} = 0.003200 \text{ K}^{-1}$$

Analysis The characteristic length in this case is determined from

$$L_c = \frac{D_2 - D_1}{2} = \frac{25 - 15}{2} = 5 \text{ cm}.$$

Then,

$$Ra = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003200 \text{ K}^{-1})(350 - 275 \text{ K})(0.05 \text{ m})^3}{(1.697 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7256) = 7.415 \times 10^5$$

The effective thermal conductivity is

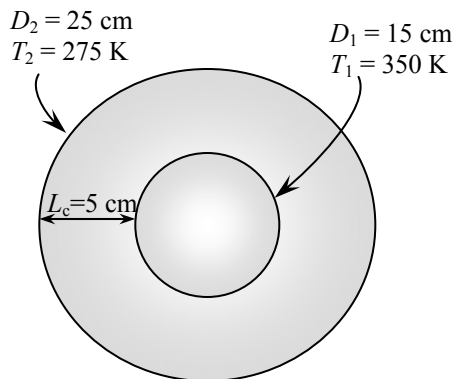
$$F_{\text{sph}} = \frac{L_c}{(D_i D_o)^4 (D_i^{-7/5} + D_o^{-7/5})^5} = \frac{0.05 \text{ m}}{[(0.15 \text{ m})(0.25 \text{ m})]^4 [(0.15 \text{ m})^{-7/5} + (0.25 \text{ m})^{-7/5}]^5} = 0.005900$$

$$k_{\text{eff}} = 0.74k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{sph}} Ra)^{1/4}$$

$$= 0.74(0.02658 \text{ W/m}\cdot^\circ\text{C}) \left(\frac{0.7256}{0.861 + 0.7256} \right)^{1/4} [(0.00590)(7.415 \times 10^5)]^{1/4} = 0.1315 \text{ W/m}\cdot^\circ\text{C}$$

Then the rate of heat transfer between the spheres becomes

$$\dot{Q} = k_{\text{eff}} \pi \left(\frac{D_i D_o}{L_c} \right) (T_i - T_o) = (0.1315 \text{ W/m}\cdot^\circ\text{C}) \pi \left[\frac{(0.15 \text{ m})(0.25 \text{ m})}{(0.05 \text{ m})} \right] (350 - 275) \text{ K} = \mathbf{23.3 \text{ W}}$$



14-71 EES Prob. 14-70 is reconsidered. The rate of natural convection heat transfer as a function of the hot surface temperature of the sphere is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$D_1 = 0.15 \text{ [m]}$

$D_2 = 0.25 \text{ [m]}$

$T_1 = 350 \text{ [K]}$

$T_2 = 275 \text{ [K]}$

"PROPERTIES"

Fluid\$='air'

$k = \text{Conductivity}(\text{Fluid}\$, T = T_{\text{ave}})$

$\text{Pr} = \text{Prandtl}(\text{Fluid}\$, T = T_{\text{ave}})$

$\rho = \text{Density}(\text{Fluid}\$, T = T_{\text{ave}}, P = 101.3)$

$\mu = \text{Viscosity}(\text{Fluid}\$, T = T_{\text{ave}})$

$\nu = \mu / \rho$

$\beta = 1 / T_{\text{ave}}$

$T_{\text{ave}} = 1/2 * (T_1 + T_2)$

$g = 9.807 \text{ [m/s}^2\text{]}$

"ANALYSIS"

$L = (D_2 - D_1) / 2$

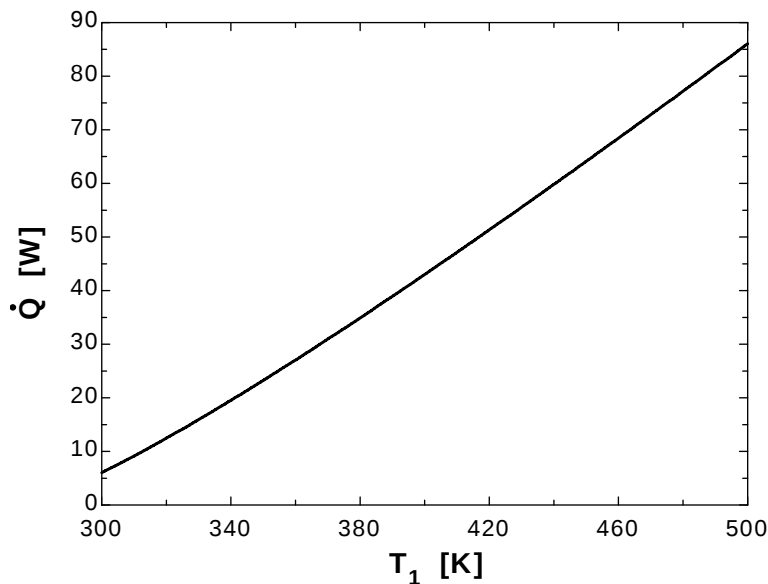
$\text{Ra} = (g * \beta * (T_1 - T_2) * L^3) / \nu^2 * \text{Pr}$

$F_{\text{sph}} = L / ((D_1 * D_2)^4 * (D_1^{(-7/5)} + D_2^{(-7/5)})^5)$

$k_{\text{eff}} = 0.74 * k * (\text{Pr} / (0.861 + \text{Pr}))^{0.25} * (F_{\text{sph}} * \text{Ra})^{0.25}$

$\dot{Q}_{\text{dot}} = k_{\text{eff}} * \pi * (D_1 * D_2) / L * (T_1 - T_2)$

$T_1 \text{ [K]}$	$\dot{Q} \text{ [W]}$
300	6.038
310	9.147
320	12.46
330	15.93
340	19.53
350	23.25
360	27.05
370	30.94
380	34.9
390	38.93
400	43.01
410	47.15
420	51.33
430	55.55
440	59.81
450	64.11
460	68.44
470	72.8
480	77.19
490	81.61
500	86.05



14-72 The absorber plate and the glass cover of a flat-plate solar collector are maintained at specified temperatures. The rate of heat loss from the absorber plate by natural convection is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Heat loss by radiation is negligible. 4 The air pressure in the enclosure is 1 atm.

Properties The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (80 + 40)/2 = 60^\circ\text{C}$ are (Table A-22)

$$k = 0.02808 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.896 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7202$$

$$\beta = \frac{1}{T_f} = \frac{1}{(60 + 273)\text{K}} = 0.003003 \text{ K}^{-1}$$

Analysis For $\theta = 0^\circ$, we have horizontal rectangular enclosure. The characteristic length in this case is the distance between the two glasses $L_c = L = 0.025 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003003 \text{ K}^{-1})(80 - 40 \text{ K})(0.025 \text{ m})^3}{(1.896 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7202) = 3.689 \times 10^4$$

$$\begin{aligned} \text{Nu} &= 1 + 1.44 \left[1 - \frac{1708}{\text{Ra}} \right]^+ + \left[\frac{\text{Ra}^{1/3}}{18} - 1 \right]^+ \\ &= 1 + 1.44 \left[1 - \frac{1708}{3.689 \times 10^4} \right]^+ + \left[\frac{(3.689 \times 10^4)^{1/3}}{18} - 1 \right]^+ = 3.223 \end{aligned}$$

Then

$$A_s = H \times W = (1.5 \text{ m})(3 \text{ m}) = 4.5 \text{ m}^2$$

$$\dot{Q} = k \text{Nu} A_s \frac{T_1 - T_2}{L} = (0.02808 \text{ W/m}\cdot^\circ\text{C})(3.223)(4.5 \text{ m}^2) \frac{(80 - 40)^\circ\text{C}}{0.025 \text{ m}} = \mathbf{652 \text{ W}}$$

For $\theta = 30^\circ$, we obtain

$$\begin{aligned} \text{Nu} &= 1 + 1.44 \left[1 - \frac{1708}{\text{Ra} \cos \theta} \right]^+ + \left[1 - \frac{1708(\sin 1.8\theta)^{1.6}}{\text{Ra} \cos \theta} \right]^+ + \left[\frac{(\text{Ra} \cos \theta)^{1/3}}{18} - 1 \right]^+ \\ &= 1 + 1.44 \left[1 - \frac{1708}{(3.689 \times 10^4) \cos(30)} \right]^+ + \left[1 - \frac{1708[\sin(1.8 \times 30)]^{1.6}}{(3.689 \times 10^4) \cos(30)} \right]^+ + \left[\frac{[(3.689 \times 10^4) \cos(30)]^{1/3}}{18} - 1 \right]^+ \\ &= 3.074 \end{aligned}$$

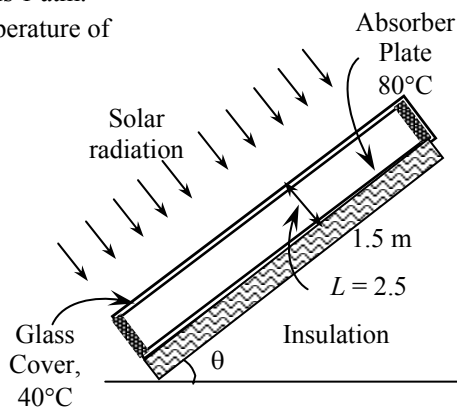
$$\dot{Q} = k \text{Nu} A_s \frac{T_1 - T_2}{L} = (0.02808 \text{ W/m}\cdot^\circ\text{C})(3.074)(4.5 \text{ m}^2) \frac{(80 - 40)^\circ\text{C}}{0.025 \text{ m}} = \mathbf{621 \text{ W}}$$

For $\theta = 90^\circ$, we have vertical rectangular enclosure. The Nusselt number for this geometry and orientation can be determined from $(\text{Ra} = 3.689 \times 10^4)$ - same as that for horizontal case)

$$\text{Nu} = 0.42 \text{Ra}^{1/4} \text{Pr}^{0.012} \left(\frac{H}{L} \right)^{-0.3} = 0.42(3.689 \times 10^4)^{1/4} (0.7202)^{0.012} \left(\frac{2 \text{ m}}{0.025 \text{ m}} \right)^{-0.3} = 1.557$$

$$\dot{Q} = k \text{Nu} A_s \frac{T_1 - T_2}{L} = (0.02808 \text{ W/m}\cdot^\circ\text{C})(1.557)(4.5 \text{ m}^2) \frac{(80 - 40)^\circ\text{C}}{0.025 \text{ m}} = \mathbf{315 \text{ W}}$$

Discussion Caution is advised for the vertical case since the condition $H/L < 40$ is not satisfied.



14-73 A simple solar collector is built by placing a clear plastic tube around a garden hose. The rate of heat loss from the water in the hose per meter of its length by natural convection is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Heat loss by radiation is negligible. 3 The air pressure in the enclosure is 1 atm.

Properties The properties of air at 1 atm and the anticipated average temperature of $(T_i + T_o)/2 = (65 + 35)/2 = 50^\circ\text{C}$ are (Table A-22)

$$k = 0.02735 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7228$$

$$\beta = \frac{1}{T_f} = \frac{1}{(50 + 273)\text{K}} = 0.003096 \text{ K}^{-1}$$

Analysis We assume the plastic tube temperature to be 35°C . We will check this assumption later, and repeat calculations, if necessary. The characteristic length in this case is

$$L_c = \frac{D_o - D_i}{2} = \frac{5 - 1.6}{2} = 1.7 \text{ cm}$$

Then,

$$\text{Ra} = \frac{g\beta(T_i - T_o)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003096 \text{ K}^{-1})(65 - 35 \text{ K})(0.017 \text{ m})^3}{(1.798 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7228) = 10,000$$

The effective thermal conductivity is

$$F_{\text{cyl}} = \frac{[\ln(D_o / D_i)]^4}{L_c^3 (D_i^{-3/5} + D_o^{-3/5})^5} = \frac{[\ln(0.05 / 0.016)]^4}{(0.017 \text{ m})^3 [(0.016 \text{ m})^{-3/5} + (0.05 \text{ m})^{-3/5}]^5} = 0.1821$$

$$k_{\text{eff}} = 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyl}} \text{Ra})^{1/4}$$

$$= 0.386(0.02735 \text{ W/m}\cdot^\circ\text{C}) \left(\frac{0.7228}{0.861 + 0.7228} \right)^{1/4} [(0.1821)(10,000)]^{1/4} = 0.05670 \text{ W/m}\cdot^\circ\text{C}$$

Then the rate of heat transfer between the cylinders becomes

$$\dot{Q} = \frac{2\pi k_{\text{eff}}}{\ln(D_o / D_i)} (T_i - T_o) = \frac{2\pi(0.05670 \text{ W/m}\cdot^\circ\text{C})}{\ln(0.05 / 0.016)} (65 - T_o) \quad (\text{Eq. 1})$$

Now we will calculate heat transfer from plastic tube to the ambient air by natural convection. Note that we should find a result close to the value we have already calculated since in steady operation they must be equal to each other. Also note that we neglect radiation heat transfer. We will use the same assumption for the plastic tube temperature (i.e., 35°C). The properties of air at 1 atm and the film temperature of $T_{\text{avg}} = (T_s + T_\infty) / 2 = (35 + 26) / 2 = 30.5^\circ\text{C}$ are

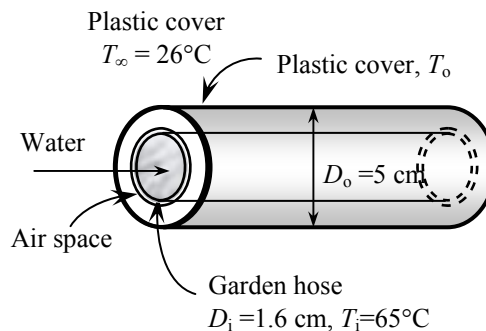
$$k = 0.02592 \text{ W/m}\cdot^\circ\text{C}, \quad \nu = 1.613 \times 10^{-5} \text{ m}^2/\text{s},$$

$$\text{Pr} = 0.7281, \text{ and } \beta = 1/T_f = 1/(30.5 + 273)\text{K} = 0.003295 \text{ K}^{-1}$$

The characteristic length in this case is the outer diameter of the solar collector $L_c = D_o = 0.05 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D_o^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003295 \text{ K}^{-1})(35 - 26 \text{ K})(0.05 \text{ m})^3}{(1.613 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7281) = 1.018 \times 10^5$$

$$\text{Nu} = \left\{ 0.6 + \frac{0.387 \text{Ra}^{1/6}}{[1 + (0.559 / \text{Pr})^{9/16}]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387(1.018 \times 10^5)^{1/6}}{[1 + (0.559 / 0.7281)^{9/16}]^{8/27}} \right\}^2 = 7.838$$



$$A_o = \pi D_o L = \pi(0.05 \text{ m})(1 \text{ m}) = 0.1571 \text{ m}^2$$

$$h_o = \frac{k}{D_o} Nu = \frac{0.02592 \text{ W/m} \cdot ^\circ\text{C}}{0.05 \text{ m}} (7.838) = 4.063 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q} = hA_o(T_o - T_\infty) = (4.063 \text{ W/m}^2 \cdot ^\circ\text{C})(0.1571 \text{ m}^2)(T_o - 26)^\circ\text{C} \quad (\text{Eq. 2})$$

Solving Eq. 1 and Eq. 2 simultaneously, we find

$$T_o = 38.8^\circ\text{C}, \quad \dot{Q} = 8.18 \text{ W}$$

Repeating the calculations at the new average temperature for enclosure analysis and at the new film temperature for convection at the outer surface analysis using the new calculated temperature 38.8°C , we find

$$T_o = 39.0^\circ\text{C}, \quad \dot{Q} = \mathbf{8.22 \text{ W}}$$

14-74 EES Prob. 14-73 is reconsidered. The rate of heat loss from the water by natural convection as a function of the ambient air temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D_1=0.016 [m]
 D_2=0.05 [m]
 T_1=65 "[C]"
 T_infinity=26 [C]
 Length=1 [m] "unit length of the tube is considered"

"PROPERTIES for enclosure"

Fluid\$='air'
 k_1=Conductivity(Fluid\$, T=T_ave)
 Pr_1=Prandtl(Fluid\$, T=T_ave)
 rho_1=Density(Fluid\$, T=T_ave, P=101.3)
 mu_1=Viscosity(Fluid\$, T=T_ave)
 nu_1=mu_1/rho_1
 beta_1=1/(T_ave+273)
 T_ave=1/2*(T_1+T_2)
 g=9.807 [m/s^2]

"ANALYSIS for enclosure"

L=(D_2-D_1)/2
 $Ra_1 = (g \cdot \beta_1 \cdot (T_1 - T_2) \cdot L^3) / (\nu_1^2 \cdot Pr_1)$
 $F_{cyl} = (\ln(D_2/D_1))^4 / (L^3 \cdot (D_1^{-3/5} + D_2^{-3/5}))^5$
 $k_{eff} = 0.386 \cdot k_1 \cdot (Pr_1 / (0.861 + Pr_1))^{0.25} \cdot (F_{cyl} \cdot Ra_1)^{0.25}$
 $\dot{Q} = (2 \cdot \pi \cdot k_{eff} / \ln(D_2/D_1)) \cdot (T_1 - T_2)$

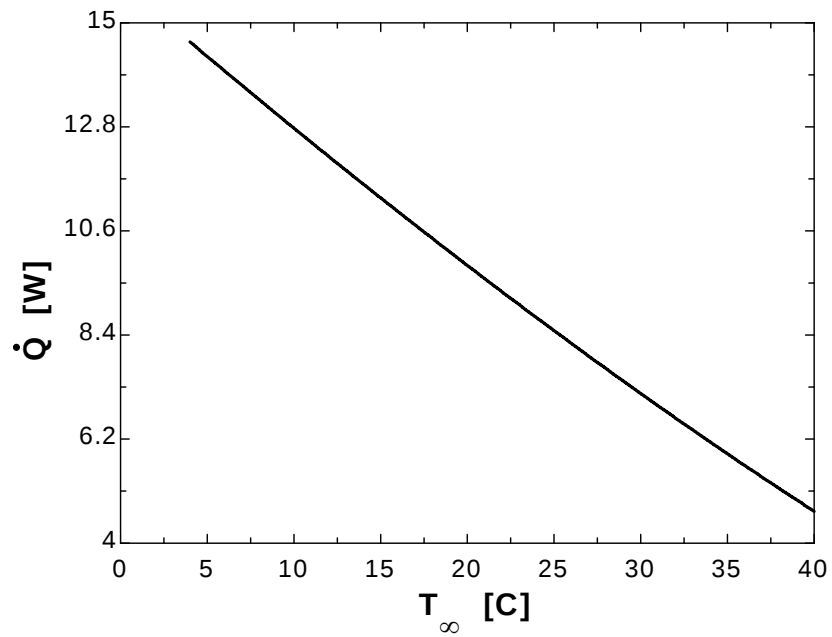
"PROPERTIES for convection on the outer surface"

k_2=Conductivity(Fluid\$, T=T_film)
 Pr_2=Prandtl(Fluid\$, T=T_film)
 rho_2=Density(Fluid\$, T=T_film, P=101.3)
 mu_2=Viscosity(Fluid\$, T=T_film)
 nu_2=mu_2/rho_2
 beta_2=1/(T_film+273)
 T_film=1/2*(T_2+T_infinity)

"ANALYSIS for convection on the outer surface"

delta=D_2
 $Ra_2 = (g \cdot \beta_2 \cdot (T_2 - T_{infinity}) \cdot \delta^3) / (\nu_2^2 \cdot Pr_2)$
 $Nusselt = (0.6 + (0.387 \cdot Ra_2^{1/6}) / (1 + (0.559 / Pr_2)^{(9/16)}))^{(8/27)}^2$
 $h = k_2 / \delta \cdot Nusselt$
 $A = \pi \cdot D_2 \cdot Length$
 $\dot{Q} = h \cdot A \cdot (T_2 - T_{infinity})$

T_{∞} [W]	Q [W]
4	14.6
6	13.98
8	13.37
10	12.77
12	12.18
14	11.59
16	11.01
18	10.44
20	9.871
22	9.314
24	8.764
26	8.222
28	7.688
30	7.163
32	6.647
34	6.139
36	5.641
38	5.153
40	4.675



14-75 A double pane window with an air gap is considered. The rate of heat transfer through the window by natural convection the temperature of the outer glass layer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The air pressure in the enclosure is 1 atm. 4 Radiation heat transfer is neglected.

Properties For natural convection between the inner surface of the window and the room air, the properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (18 + 26)/2 = 22^\circ\text{C}$ are (Table A-22)

$$k = 0.02529 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.534 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7304$$

$$\beta = \frac{1}{T_f} = \frac{1}{(22 + 273)\text{K}} = 0.00339 \text{ K}^{-1}$$

For natural convection between the two glass sheets separated by an air gap, the properties of air at 1 atm and the anticipated average temperature of $(T_1 + T_2)/2 = (18 + 0)/2 = 9^\circ\text{C}$ are (Table A-22)

$$k = 0.02431 \text{ W/m}\cdot^\circ\text{C}, \quad \nu = 1.417 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7339, \quad \beta = \frac{1}{T_f} = \frac{1}{(9 + 273)\text{K}} = 0.003546 \text{ K}^{-1}$$

Analysis We first calculate the natural convection heat transfer between the room air and the inner surface of the window.

$$L_c = H = 1.3 \text{ m}$$

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)H^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00339 \text{ K}^{-1})(26 - 18)\text{K}(1.3 \text{ m})^3}{(1.534 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7304) = 1.813 \times 10^9$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.813 \times 10^9)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7304} \right)^{9/16} \right]^{8/27}} \right\}^2 = 148.3$$

$$h = \frac{k}{H} \text{Nu} = \frac{0.02529 \text{ W/m}\cdot^\circ\text{C}}{1.3 \text{ m}} (148.3) = 2.884 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = H \times W = (1.3 \text{ m})(2.8 \text{ m}) = 3.64 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = hA_s(T_\infty - T_s) = (2.884 \text{ W/m}^2\cdot^\circ\text{C})(3.64 \text{ m}^2)(26 - 18)^\circ\text{C} = \mathbf{84.0 \text{ W}}$$

Next, we consider the natural convection between the two glass sheets separated by an air gap.

$$L_c = L = 2.2 \text{ cm}$$

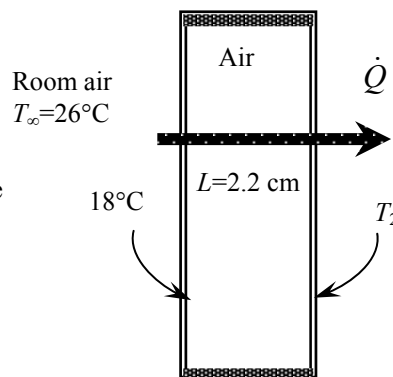
$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003546 \text{ K}^{-1})(18 - 0)\text{K}(0.022 \text{ m})^3}{(1.417 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7339) = 24,370$$

$$\text{Nu} = 0.42\text{Ra}^{1/4} \text{Pr}^{0.012} \left(\frac{H}{L} \right)^{-0.3} = 0.42(24,370)^{1/4} (0.7339)^{0.012} \left(\frac{1.3 \text{ m}}{0.022 \text{ m}} \right)^{-0.3} = 1.538$$

Under steady operation, the rate of heat transfer between the room air and the inner surface of the window is equal to the heat transfer through the air gap. Setting these two equal to each other we obtain the temperature of the outer glass sheet

$$\dot{Q} = k\text{Nu}A_s \frac{T_1 - T_2}{L} \longrightarrow 84 \text{ W} = (0.02431 \text{ W/m}\cdot^\circ\text{C})(1.538)(3.64 \text{ m}^2) \frac{(18 - T_2)^\circ\text{C}}{0.022 \text{ m}} \longrightarrow T_2 = \mathbf{4.4^\circ\text{C}}$$

which is sufficiently close to the assumed temperature 0°C . Therefore, there is no need to repeat the calculations.



14-76 The space between the two concentric cylinders is filled with water or air. The rate of heat transfer from the outer cylinder to the inner cylinder by natural convection is to be determined for both cases.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The air pressure in the enclosure is 1 atm. 4 Heat transfer by radiation is negligible.

Properties The properties of water air at the average temperature of $(T_i + T_o)/2 = (54 + 106)/2 = 80^\circ\text{C}$ are (Table A-15)

$$k = 0.670 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 3.653 \times 10^{-7} \text{ m}^2/\text{s}$$

$$\text{Pr} = 2.22$$

$$\beta = 0.653 \times 10^{-3} \text{ K}^{-1}$$

The properties of air at 1 atm and the average temperature of $(T_i + T_o)/2 = (54 + 106)/2 = 80^\circ\text{C}$ are (Table A-22)

$$k = 0.02953 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 2.097 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7154$$

$$\beta = \frac{1}{T_f} = \frac{1}{(80 + 273)\text{K}} = 0.002833 \text{ K}^{-1}$$

Analysis (a) The fluid is water:

$$L_c = \frac{D_o - D_i}{2} = \frac{65 - 55}{2} = 5 \text{ cm.}$$

$$\text{Ra} = \frac{g\beta(T_o - T_i)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.653 \times 10^{-3} \text{ K}^{-1})(106 - 54)\text{K}(0.05 \text{ m})^3}{(3.653 \times 10^{-7} \text{ m}^2/\text{s})^2} (2.22) = 6.927 \times 10^8$$

The effective thermal conductivity is

$$F_{\text{cyl}} = \frac{\left[\ln \frac{D_o}{D_i} \right]^4}{L_c^3 (D_i^{-3/5} + D_o^{-3/5})^5} = \frac{\left[\ln \frac{0.65 \text{ m}}{0.55 \text{ m}} \right]^4}{(0.05 \text{ m})^3 [(0.55 \text{ m})^{-7/5} + (0.65 \text{ m})^{-7/5}]^5} = 0.04136$$

$$k_{\text{eff}} = 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyl}} \text{Ra})^{1/4}$$

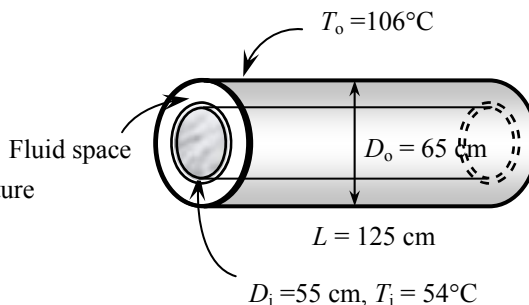
$$= 0.386(0.670 \text{ W/m}\cdot^\circ\text{C}) \left(\frac{2.22}{0.861 + 2.22} \right)^{1/4} [(0.04136)(6.927 \times 10^8)]^{1/4} = 17.43 \text{ W/m}\cdot^\circ\text{C}$$

Then the rate of heat transfer between the cylinders becomes

$$\dot{Q} = \frac{2\pi k_{\text{eff}}}{\ln \left(\frac{D_o}{D_i} \right)} (T_o - T_i) = \frac{2\pi(17.43 \text{ W/m}\cdot^\circ\text{C})}{\ln \left(\frac{0.65 \text{ m}}{0.55 \text{ m}} \right)} (106 - 54) = 34,090 \text{ W} = \mathbf{34.1 \text{ kW}}$$

(b) The fluid is air:

$$\text{Ra} = \frac{g\beta(T_o - T_i)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002833 \text{ K}^{-1})(106 - 54)\text{K}(0.05 \text{ m})^3}{(2.097 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7154) = 2.939 \times 10^5$$



The effective thermal conductivity is

$$F_{\text{cyl}} = \frac{\left[\ln \frac{D_o}{D_i} \right]^4}{L_c^3 (D_i^{-3/5} + D_o^{-3/5})^5} = \frac{\left[\ln \frac{0.65 \text{ m}}{0.55 \text{ m}} \right]^4}{(0.05 \text{ m})^3 \left[(0.55 \text{ m})^{-7/5} + (0.65 \text{ m})^{-7/5} \right]^5} = 0.04136$$

$$k_{\text{eff}} = 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyl}} \text{Ra})^{1/4}$$

$$= 0.386(0.02953 \text{ W/m} \cdot ^\circ\text{C}) \left(\frac{0.7154}{0.861 + 0.7154} \right)^{1/4} \left[(0.04136)(2.939 \times 10^5) \right]^{1/4} = 0.09824 \text{ W/m} \cdot ^\circ\text{C}$$

Then the rate of heat transfer between the cylinders becomes

$$\dot{Q} = \frac{2\pi k_{\text{eff}}}{\ln \left(\frac{D_o}{D_i} \right)} (T_o - T_i) = \frac{2\pi(0.09824 \text{ W/m} \cdot ^\circ\text{C})}{\ln \left(\frac{0.65 \text{ m}}{0.55 \text{ m}} \right)} (106 - 54) = \mathbf{192 \text{ W}}$$

Review Problems

14-77 A cold cylinder is placed horizontally in hot air. The rates of heat transfer from the stack with and without wind cases are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

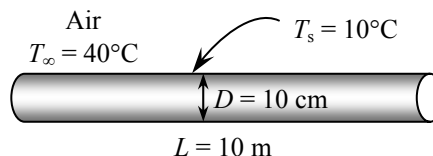
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (40 + 10)/2 = 25^\circ\text{C}$ are (Table A-22)

$$k = 0.02551 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7296$$

$$\beta = \frac{1}{T_f} = \frac{1}{(25 + 273)\text{K}} = 0.003356 \text{ K}^{-1}$$



Analysis (a) When the stack is exposed to 10 m/s winds, the heat transfer will be by forced convection. We have flow of air over a cylinder and the heat transfer rate is determined as follows:

$$\text{Re} = \frac{VD}{\nu} = \frac{(10 \text{ m/s})(0.1 \text{ m})}{1.562 \times 10^{-5} \text{ m}^2/\text{s}} = 64,020$$

$$\text{Nu} = 0.027 \text{Re}^{0.805} \text{Pr}^{1/3} = 0.027(64,020)^{0.805} (0.7296)^{1/3} = 179.8 \quad (\text{from Table 19-2})$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.02551 \text{ W/m}\cdot^\circ\text{C}}{0.1 \text{ m}} (179.8) = 45.87 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q}_{\text{forced conv}} = hA(T_\infty - T_s) = (45.87 \text{ W/m}^2\cdot^\circ\text{C})(\pi \times 0.1 \times 10 \text{ m}^2)(40 - 10)^\circ\text{C} = \mathbf{4323 \text{ W}}$$

(b) Without wind the heat transfer will be by natural convection. The characteristic length in this case is the outer diameter of the cylinder, $L_c = D = 0.1 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003356 \text{ K}^{-1})(40 - 10 \text{ K})(0.1 \text{ m})^3}{(1.562 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7296) = 2.953 \times 10^6$$

$$\text{Nu} = \left\{ 0.6 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387(2.953 \times 10^6)^{1/6}}{\left[1 + (0.559 / 0.7296)^{9/16} \right]^{8/27}} \right\}^2 = 19.86$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.02551 \text{ W/m}\cdot^\circ\text{C}}{0.1 \text{ m}} (19.86) = 5.066 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q}_{\text{nat. conv}} = hA(T_\infty - T_s) = (5.066 \text{ W/m}^2\cdot^\circ\text{C})(\pi \times 0.1 \times 10 \text{ m}^2)(40 - 10)^\circ\text{C} = \mathbf{477 \text{ W}}$$

14-78 A spherical vessel is completely submerged in a large water-filled tank. The rates of heat transfer from the vessel by natural convection, conduction, and forced convection are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature is constant.

Properties The properties of water at the film temperature of $(T_s + T_\infty)/2 = (30 + 20)/2 = 25^\circ\text{C}$ are (Table A-15)

$$\rho = 997 \text{ kg/m}^3$$

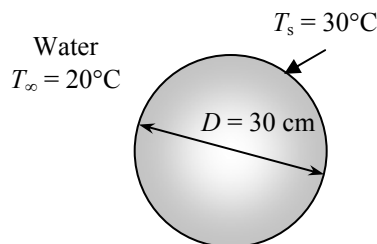
$$k = 0.607 \text{ W/m}\cdot^\circ\text{C}$$

$$\mu = 0.891 \times 10^{-3} \text{ kg/m}\cdot\text{s}$$

$$\nu = \mu / \rho = 8.937 \times 10^{-7} \text{ m}^2/\text{s}$$

$$\text{Pr} = 6.14$$

$$\beta = 0.247 \times 10^{-3} \text{ K}^{-1}$$



Analysis (a) Heat transfer in this case will be by natural convection.

The characteristic length in this case is $L_c = D = 0.3 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.247 \times 10^{-3} \text{ K}^{-1})(30 - 20 \text{ K})(0.3 \text{ m})^3}{(8.937 \times 10^{-7} \text{ m}^2/\text{s})^2} (6.14) = 5.029 \times 10^9$$

$$\text{Nu} = 2 + \frac{0.589 \text{Ra}^{1/4}}{\left[1 + (0.469 / \text{Pr})^{9/16}\right]^{4/9}} = 2 + \frac{0.589(5.029 \times 10^9)^{1/4}}{\left[1 + (0.469 / 6.14)^{9/16}\right]^{4/9}} = 144.8$$

Then

$$h = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m}\cdot^\circ\text{C}}{0.3 \text{ m}} (144.8) = 293.0 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi D^2 = \pi (0.3 \text{ m})^2 = 0.2827 \text{ m}^2$$

The rate of heat transfer is

$$\dot{Q}_{\text{nat. conv}} = hA(T_s - T_\infty) = (293.0 \text{ W/m}^2\cdot^\circ\text{C})(0.2827 \text{ m}^2)(30 - 20)^\circ\text{C} = \mathbf{828 \text{ W}}$$

(b) When buoyancy force is neglected, there will be no convection currents (since $\beta = 0$) and the heat transfer will be by conduction. Then Rayleigh number becomes zero ($\text{Ra} = 0$). The Nusselt number in this case is

$$\text{Nu} = 2$$

Then

$$h = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m}\cdot^\circ\text{C}}{0.3 \text{ m}} (2) = 4.047 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q}_{\text{cond}} = hA(T_s - T_\infty) = (4.047 \text{ W/m}^2\cdot^\circ\text{C})(0.2827 \text{ m}^2)(30 - 20)^\circ\text{C} = \mathbf{11.4 \text{ W}}$$

(c) In this case, the heat transfer from the vessel is by forced convection. The properties of water at the free stream temperature of 20°C are (Table A-15)

$$\rho = 998 \text{ kg/m}^3$$

$$k = 0.598 \text{ W/m}\cdot^\circ\text{C}$$

$$\mu_\infty = 1.002 \times 10^{-3} \text{ kg/m}\cdot\text{s}$$

$$\nu = \mu_\infty / \rho = 1.004 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\mu_{s, @ 30^\circ\text{C}} = 0.798 \times 10^{-3} \text{ kg/m}\cdot\text{s}$$

$$\text{Pr} = 7.01$$

The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(0.2 \text{ m/s})(0.3 \text{ m})}{1.004 \times 10^{-6} \text{ m}^2/\text{s}} = 59,760$$

The Nusselt number is

$$\begin{aligned} Nu = \frac{hD}{k} &= 2 + \left[0.4 \text{Re}^{0.5} + 0.06 \text{Re}^{2/3} \right] \text{Pr}^{0.4} \left(\frac{\mu_{\infty}}{\mu_s} \right)^{1/4} \\ &= 2 + \left[0.4(59,760)^{0.5} + 0.06(59,760)^{2/3} \right] (7.01)^{0.4} \left(\frac{1.002 \times 10^{-3}}{0.798 \times 10^{-3}} \right)^{1/4} = 439.1 \end{aligned}$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.598 \text{ W/m} \cdot ^\circ\text{C}}{0.3 \text{ m}} (439.1) = 875.3 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer is

$$\dot{Q}_{\text{forced conv}} = hA(T_s - T_{\infty}) = (875.3 \text{ W/m}^2 \cdot ^\circ\text{C})(0.2827 \text{ m}^2)(30 - 20)^\circ\text{C} = \mathbf{2474 \text{ W}}$$

14-79 A vertical cylindrical vessel loses heat to the surrounding air. The rates of heat transfer from the vessel with and without wind cases are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (60 + 0)/2 = 30^\circ\text{C}$ are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7282$$

$$\beta = \frac{1}{T_f} = \frac{1}{(30 + 273)\text{K}} = 0.003300 \text{ K}^{-1}$$

Analysis (a) When there is no wind heat transfer is by natural convection. The characteristic length in this case is the height of the vessel, $L_c = L = 3 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003300 \text{ K}^{-1})(60 - 0 \text{ K})(3 \text{ m})^3}{(1.608 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7282) = 1.477 \times 10^{11}$$

We can treat this vertical cylinder as a vertical plate since

$$\frac{35L}{Gr^{1/4}} = \frac{35(3)}{(1.477 \times 10^{11} / 0.7282)^{1/4}} = 0.156 < 1.0 \quad \text{and thus } D \geq \frac{35L}{Gr^{1/4}}$$

The Nusselt number is determined from

$$Nu = 0.1Ra^{1/3} = 0.1(1.477 \times 10^{11})^{1/3} = 528.6 \quad (\text{from Table 14-1})$$

Then

$$h = \frac{k}{L_c} Nu = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{3 \text{ m}} (528.6) = 4.560 \text{ W/m}^2\cdot^\circ\text{C}$$

and

$$\dot{Q} = hA(T_s - T_\infty) = (4.560 \text{ W/m}^2\cdot^\circ\text{C})(\pi \times 1 \times 3 \text{ m}^2)(60 - 0)^\circ\text{C} = \mathbf{2579 \text{ W}}$$

(b) When the vessel is exposed to 20 km/h winds, the heat transfer will be by forced convection. We have flow of air over a cylinder and the heat transfer rate is determined as follows:

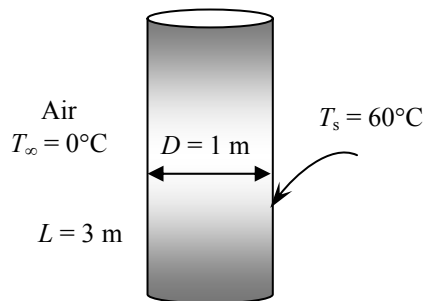
$$\text{Re} = \frac{VD}{\nu} = \frac{(20 \times 1000 / 3600 \text{ m/s})(1.0 \text{ m})}{1.608 \times 10^{-5} \text{ m}^2/\text{s}} = 3.455 \times 10^5$$

$$\text{Nu} = 0.027 \text{Re}^{0.805} \text{Pr}^{1/3} = 0.027(3.455 \times 10^5)^{0.805} (0.7282)^{1/3} = 698.1$$

$$h = \frac{k}{D} \text{Nu} = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{1.0 \text{ m}} (698.1) = 18.07 \text{ W/m}^2\cdot^\circ\text{C}$$

$$\dot{Q} = hA(T_s - T_\infty) = (18.07 \text{ W/m}^2\cdot^\circ\text{C})(\pi \times 1 \times 3 \text{ m}^2)(60 - 0)^\circ\text{C} = \mathbf{10,220 \text{ W}}$$

Discussion There is about four-fold increase in heat transfer due to winds.



14-80 A solid sphere is completely submerged in a large pool of oil. The rates of heat transfer from the sphere by natural convection, conduction, and forced convection are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The surface temperature is constant.

Properties The properties of oil are given in problem statement.

Analysis (a) For conduction heat transfer, $\beta = 0$ and $Ra = 0$. Then the Nusselt number is

$$Nu = 2$$

Then

$$h = \frac{k}{D} Nu = \frac{0.22 \text{ W/m} \cdot ^\circ\text{C}}{0.5 \text{ m}} (2) = 0.88 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q}_{\text{cond}} = hA(T_s - T_\infty) = (0.88 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi \times (0.5 \text{ m})^2](60 - 20)^\circ\text{C} = \mathbf{27.6 \text{ W}}$$

(b) In this case, the heat transfer from the vessel is by forced convection. Note that the fluid properties in this case are to be evaluated at the free stream temperature T_∞ except for μ_s , which is evaluated at the surface temperature, T_s . The Prandtl number and Reynolds number are

$$Pr = \frac{\mu c_p}{k} = \frac{(0.010 \text{ kg/m} \cdot \text{s})(1880 \text{ J/kg} \cdot ^\circ\text{C})}{0.22 \text{ W/m} \cdot ^\circ\text{C}} = 85.45$$

$$Re = \frac{VD\rho}{\mu} = \frac{(1.5 \text{ m/s})(0.5 \text{ m})(888 \text{ kg/m}^3)}{0.010 \text{ kg/m} \cdot \text{s}} = 66,600$$

The Nusselt number is

$$Nu = \frac{hD}{k} = 2 + \left[0.4 Re^{0.5} + 0.06 Re^{2/3} \right] Pr^{0.4} \left(\frac{\mu_\infty}{\mu_s} \right)^{1/4}$$

$$= 2 + \left[0.4(66,600)^{0.5} + 0.06(66,600)^{2/3} \right] (85.45)^{0.4} \left(\frac{0.010}{0.004} \right)^{1/4} = 1506$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.22 \text{ W/m} \cdot ^\circ\text{C}}{0.5 \text{ m}} (1506) = 662.4 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer is

$$\dot{Q}_{\text{forced conv}} = hA(T_s - T_\infty) = (662.4 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi \times (0.5 \text{ m})^2](60 - 20)^\circ\text{C} = \mathbf{20,810 \text{ W}}$$

(c) Heat transfer in this case will be by natural convection. Note that the fluid properties in this case are to be evaluated at the film temperature of 40°C . The characteristic length in this case is $L_c = D = 0.5 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} Pr = \frac{(9.81 \text{ m/s}^2)(0.00070 \text{ K}^{-1})(60 - 20 \text{ K})(0.5 \text{ m})^3}{(0.007 / 876 \text{ m}^2/\text{s})^2} (65.5) = 3.522 \times 10^{10}$$

where $Pr = \frac{\mu c_p}{k} = \frac{(0.007 \text{ kg/m} \cdot \text{s})(1965 \text{ J/kg} \cdot ^\circ\text{C})}{0.21 \text{ W/m} \cdot ^\circ\text{C}} = 65.5$

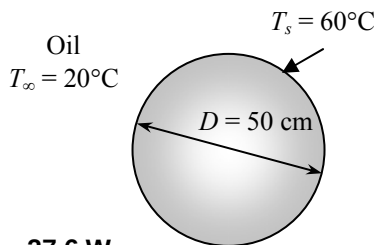
Then

$$Nu = 2 + \frac{0.589 Ra^{1/4}}{\left[1 + (0.469 / Pr)^{9/16} \right]^{4/9}} = 2 + \frac{0.589 (3.522 \times 10^{10})^{1/4}}{\left[1 + (0.469 / 65.5)^{9/16} \right]^{4/9}} = 250.4$$

$$h = \frac{k}{D} Nu = \frac{0.21 \text{ W/m} \cdot ^\circ\text{C}}{0.5 \text{ m}} (250.4) = 105.2 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$\dot{Q}_{\text{nat. conv}} = hA(T_s - T_\infty) = (105.2 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi \times (0.5 \text{ m})^2](60 - 20)^\circ\text{C} = \mathbf{3304 \text{ W}}$$

Discussion The heat transfer from the sphere by forced convection is much greater than that by natural convection and the heat transfer by natural convection is much greater than that by conduction.



14-81E A small cylindrical resistor mounted on the lower part of a vertical circuit board. The approximate surface temperature of the resistor is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Radiation effects are negligible. 5 Heat transfer through the connecting wires is negligible.

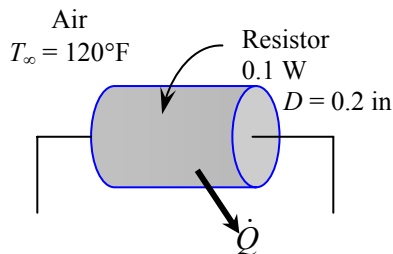
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (220 + 120)/2 = 170^\circ\text{F}$ are (Table A-22E)

$$k = 0.01692 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.222 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7161$$

$$\beta = \frac{1}{T_f} = \frac{1}{(170 + 460)\text{R}} = 0.001587 \text{ R}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 220°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the diameter of resistor, $L_c = D = 0.2 \text{ in}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001587 \text{ R}^{-1})(220 - 120 \text{ R})(0.2 / 12 \text{ ft})^3}{(0.222 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7161) = 343.8$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (343.8)^{1/6}}{\left[1 + (0.559 / 0.7161)^{9/16} \right]^{8/27}} \right\}^2 = 2.105$$

$$h = \frac{k}{D} Nu = \frac{0.01692 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.2 / 12 \text{ ft}} (2.105) = 2.138 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_s = \pi DL + 2\pi D^2 / 4 = \pi(0.2 / 12 \text{ ft})(0.3 / 12 \text{ ft}) + 2\pi(0.2 / 12 \text{ ft})^2 / 4 = 0.00175 \text{ ft}^2$$

$$\text{and } \dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 120^\circ\text{F} + \frac{(0.1 \times 3.412) \text{ Btu/h}}{(2.138 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(0.00175 \text{ ft}^2)} = 211^\circ\text{F}$$

which is sufficiently close to the assumed temperature for the evaluation of properties. Therefore, there is no need to repeat calculations.

14-82 An ice chest filled with ice at 0°C is exposed to ambient air. The time it will take for the ice in the chest to melt completely is to be determined for natural and forced convection cases.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Heat transfer from the base of the ice chest is disregarded. 4 Radiation effects are negligible. 5 Heat transfer coefficient is the same for all surfaces considered. 6 The local atmospheric pressure is 1 atm.

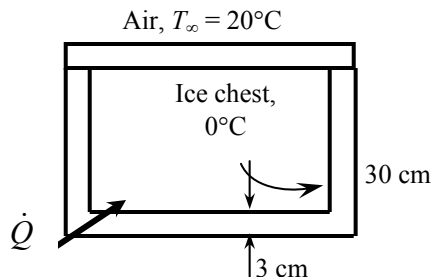
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (15 + 20)/2 = 17.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02495 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.493 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7316$$

$$\beta = \frac{1}{T_f} = \frac{1}{(17.5 + 273)\text{K}} = 0.003442 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by “guessing” the surface temperature to be 15°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length for the side surfaces is the height of the chest, $L_c = L = 0.3 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003442 \text{ K}^{-1})(20 - 15 \text{ K})(0.3 \text{ m})^3}{(1.493 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7316) = 1.495 \times 10^7$$

$$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387 (1.495 \times 10^7)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7316} \right)^{9/16} \right]^{8/27}} \right\}^2 = 35.15$$

$$h = \frac{k}{L} Nu = \frac{0.02495 \text{ W/m}\cdot^\circ\text{C}}{0.3 \text{ m}} (35.15) = 2.923 \text{ W/m}^2\cdot^\circ\text{C}$$

The heat transfer coefficient at the top surface can be determined similarly. However, the top surface constitutes only about one-fourth of the heat transfer area, and thus we can use the heat transfer coefficient for the side surfaces for the top surface also for simplicity. The heat transfer surface area is

$$A_s = 4(0.3 \text{ m})(0.4 \text{ m}) + (0.4 \text{ m})(0.4 \text{ m}) = 0.64 \text{ m}^2$$

Then the rate of heat transfer becomes

$$\dot{Q} = \frac{T_\infty - T_{s,i}}{R_{\text{wall}} + R_{\text{conv},o}} = \frac{T_\infty - T_{s,i}}{\frac{L}{kA_s} + \frac{1}{hA_s}} = \frac{(20 - 0)^\circ\text{C}}{\frac{0.03 \text{ m}}{(0.033 \text{ W/m}\cdot^\circ\text{C})(0.64 \text{ m}^2)} + \frac{1}{(2.923 \text{ W/m}^2\cdot^\circ\text{C})(0.64 \text{ m}^2)}} = 10.23 \text{ W}$$

The outer surface temperature of the ice chest is determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_\infty - T_s) \rightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 20^\circ\text{C} - \frac{10.23 \text{ W}}{(2.923 \text{ W/m}^2\cdot^\circ\text{C})(0.64 \text{ m}^2)} = 14.53^\circ\text{C}$$

which is almost identical to the assumed value of 15°C used in the evaluation of properties and h . Therefore, there is no need to repeat the calculations.

The rate at which the ice will melt is

$$\dot{Q} = \dot{m}h_{if} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{10.23 \times 10^{-3} \text{ kJ/s}}{333.7 \text{ kJ/kg}} = 3.066 \times 10^{-5} \text{ kg/s}$$

Therefore, the melting of the ice in the chest completely will take

$$m = \dot{m}\Delta t \rightarrow \Delta t = \frac{m}{\dot{m}} = \frac{30 \text{ kg}}{3.066 \times 10^{-5} \text{ kg/s}} = 9.786 \times 10^5 \text{ s} = \mathbf{271.8 \text{ h} = 11.3 \text{ days}}$$

(b) The temperature drop across the styrofoam will be much greater in this case than that across thermal boundary layer on the surface. Thus we assume outer surface temperature of the styrofoam to be 19°C . Radiation heat transfer will be neglected. The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (19 + 20)/2 = 19.5^\circ\text{C}$ are (Table A-22)

$$k = 0.0251 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.511 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7310$$

$$\beta = \frac{1}{T_f} = \frac{1}{(19.5 + 273)\text{K}} = 0.00342 \text{ K}^{-1}$$

The characteristic length in this case is the width of the chest, $L_c = W = 0.4 \text{ m}$. Then,

$$\text{Re} = \frac{VW}{\nu} = \frac{(50 \times 1000 / 3600 \text{ m/s})(0.4 \text{ m})}{1.511 \times 10^{-5} \text{ m}^2/\text{s}} = 367,700$$

which is less than critical Reynolds number (5×10^5). Therefore the flow is laminar, and the Nusselt number is determined from

$$\text{Nu} = \frac{hW}{k} = 0.664 \text{Re}^{0.5} \text{Pr}^{1/3} = 0.664(367,700)^{0.5} (0.7310)^{1/3} = 362.7$$

$$h = \frac{k}{W} \text{Nu} = \frac{0.0251 \text{ W/m}\cdot^\circ\text{C}}{0.4 \text{ m}} (362.7) = 22.76 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \frac{T_\infty - T_{s,i}}{R_{\text{wall}} + R_{\text{conv},o}} = \frac{T_\infty - T_{s,i}}{\frac{L}{kA_s} + \frac{1}{hA_s}} = \frac{(20 - 0)^\circ\text{C}}{\frac{0.03 \text{ m}}{(0.033 \text{ W/m}\cdot^\circ\text{C})(0.64 \text{ m}^2)} + \frac{1}{(22.76 \text{ W/m}^2 \cdot ^\circ\text{C})(0.64 \text{ m}^2)}} = 13.43 \text{ W}$$

The outer surface temperature of the ice chest is determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_\infty - T_s) \rightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 20^\circ\text{C} - \frac{13.43 \text{ W}}{(22.76 \text{ W/m}^2 \cdot ^\circ\text{C})(0.64 \text{ m}^2)} = 19.1^\circ\text{C}$$

which is almost identical to the assumed value of 19°C used in the evaluation of properties and h .

Therefore, there is no need to repeat the calculations. Then the rate at which the ice will melt becomes

$$\dot{Q} = \dot{m}h_{if} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{13.43 \times 10^{-3} \text{ kJ/s}}{333.7 \text{ kJ/kg}} = 4.025 \times 10^{-5} \text{ kg/s}$$

Therefore, the melting of the ice in the chest completely will take

$$m = \dot{m}\Delta t \rightarrow \Delta t = \frac{m}{\dot{m}} = \frac{30}{4.025 \times 10^{-5}} = 7.454 \times 10^5 \text{ s} = \mathbf{207.05 \text{ h} = 8.6 \text{ days}}$$

14-83 An electronic box is cooled internally by a fan blowing air into the enclosure. The fraction of the heat lost from the outer surfaces of the electronic box is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Heat transfer from the base surface is disregarded. 4 The pressure of air inside the enclosure is 1 atm.

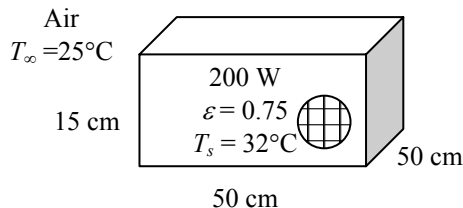
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (32 + 15)/2 = 28.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02577 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.594 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7286$$

$$\beta = \frac{1}{T_f} = \frac{1}{(28.5 + 273)\text{K}} = 0.003317 \text{ K}^{-1}$$



Analysis Heat loss from the horizontal top surface:

The characteristic length in this case is $\delta = \frac{A}{P} = \frac{(0.5 \text{ m})^2}{2[(0.5 \text{ m}) + (0.5 \text{ m})]} = 0.125 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003317 \text{ K}^{-1})(32 - 25 \text{ K})(0.125 \text{ m})^3}{(1.594 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7286) = 1.275 \times 10^6$$

$$\text{Nu} = 0.54 \text{Ra}^{1/4} = 0.54(1.275 \times 10^6)^{1/4} = 18.15$$

$$h = \frac{k}{L_c} \text{Nu} = \frac{0.02577 \text{ W/m}\cdot^\circ\text{C}}{0.125 \text{ m}} (18.15) = 3.741 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_{\text{top}} = (0.5 \text{ m})^2 = 0.25 \text{ m}^2$$

and $\dot{Q}_{\text{top}} = hA_{\text{top}}(T_s - T_\infty) = (3.741 \text{ W/m}^2\cdot^\circ\text{C})(0.25 \text{ m}^2)(32 - 25)^\circ\text{C} = 6.55 \text{ W}$

Heat loss from vertical side surfaces:

The characteristic length in this case is the height of the box $L_c = L = 0.15 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003317 \text{ K}^{-1})(32 - 25 \text{ K})(0.15 \text{ m})^3}{(1.594 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7286) = 2.204 \times 10^6$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(2.204 \times 10^6)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7286} \right)^{9/16} \right]^{8/27}} \right\}^2 = 20.55$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02577 \text{ W/m}\cdot^\circ\text{C}}{0.15 \text{ m}} (20.55) = 3.530 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_{\text{side}} = 4(0.15 \text{ m})(0.5 \text{ m}) = 0.3 \text{ m}^2$$

and $\dot{Q}_{\text{side}} = hA_{\text{side}}(T_s - T_\infty) = (3.530 \text{ W/m}^2\cdot^\circ\text{C})(0.3 \text{ m}^2)(32 - 25)^\circ\text{C} = 7.41 \text{ W}$

The radiation heat loss is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) \\ &= (0.75)(0.25 + 0.3 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(32 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] = 17.95 \text{ W} \end{aligned}$$

Then the fraction of the heat loss from the outer surfaces of the box is determined to be

$$f = \frac{(6.55 + 7.41 + 17.95) \text{ W}}{200 \text{ W}} = 0.160 = \mathbf{16.0\%}$$

14-84 A spherical tank made of stainless steel is used to store iced water. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Thermal resistance of the tank is negligible. 4 The local atmospheric pressure is 1 atm.

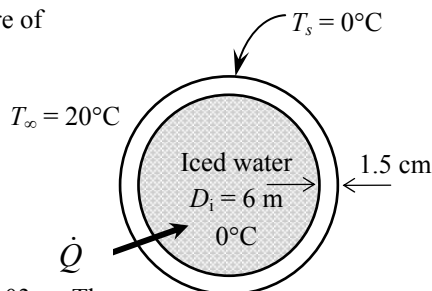
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (0 + 20)/2 = 10^\circ\text{C}$ are (Table A-22)

$$k = 0.02439 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7336$$

$$\beta = \frac{1}{T_f} = \frac{1}{(10 + 273)\text{K}} = 0.003534 \text{ K}^{-1}$$



Analysis (a) The characteristic length in this case is $L_c = D_o = 6.03 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_\infty - T_s)D_o^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003534 \text{ K}^{-1})(20 - 0 \text{ K})(6.03 \text{ m})^3}{(1.426 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7336) = 5.485 \times 10^{11}$$

$$Nu = 2 + \frac{0.589 Ra^{1/4}}{\left[1 + (0.469 / \text{Pr})^{9/16}\right]^{4/9}} = 2 + \frac{0.589(5.485 \times 10^{11})^{1/4}}{\left[1 + (0.469 / 0.7336)^{9/16}\right]^{4/9}} = 394.5$$

$$h = \frac{k}{D_o} Nu = \frac{0.02439 \text{ W/m}\cdot^\circ\text{C}}{6.03 \text{ m}} (394.5) = 1.596 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi D_o^2 = \pi (6.03 \text{ m})^2 = 114.2 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_\infty - T_s) = (1.596 \text{ W/m}^2\cdot^\circ\text{C})(114.2 \text{ m}^2)(20 - 0)^\circ\text{C} = 3646 \text{ W}$$

Heat transfer by radiation and the total rate of heat transfer are

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (1)(114.2 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(20 + 273 \text{ K})^4 - (0 + 273 \text{ K})^4] = 11,759 \text{ W} \end{aligned}$$

$$\dot{Q}_{total} = 3646 + 11,759 = 15,404 \text{ W} \cong \mathbf{15.4 \text{ kW}}$$

(b) The total amount of heat transfer during a 24-hour period is

$$Q = \dot{Q} \Delta t = (15.4 \text{ kJ/s})(24 \text{ h/day} \times 3600 \text{ s/h}) = 1.331 \times 10^6 \text{ kJ/day}$$

Then the amount of ice that melts during this period becomes

$$Q = m h_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{1.331 \times 10^6 \text{ kJ}}{333.7 \text{ kJ/kg}} = \mathbf{3988 \text{ kg}}$$

14-85 A double-pane window consisting of two layers of glass separated by an air space is considered. The rate of heat transfer through the window and the temperature of its inner surface are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Radiation effects are negligible. 4 The pressure of air inside the enclosure is 1 atm.

Properties We expect the average temperature of the air gap to be roughly the average of the indoor and outdoor temperatures, and evaluate The properties of air at 1 atm and the average temperature of $(T_{\infty 1} + T_{\infty 2})/2 = (20 + 0)/2 = 10^\circ\text{C}$ are (Table A-22)

$$k = 0.02439 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.426 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7336$$

$$\beta = \frac{1}{T_f} = \frac{1}{(10 + 273)\text{K}} = 0.003534 \text{ K}^{-1}$$

Analysis We “guess” the temperature difference across the air gap to be $15^\circ\text{C} = 15 \text{ K}$ for use in the Ra relation. The characteristic length in this case is the air gap thickness, $L_c = L = 0.03 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_1 - T_2)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003534 \text{ K}^{-1})(15 \text{ K})(0.03 \text{ m})^3}{(1.426 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7336) = 5.065 \times 10^4$$

Then the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = 0.42 Ra^{1/4} \text{Pr}^{0.012} \left(\frac{H}{L} \right)^{-0.3} = 0.42 (5.065 \times 10^4)^{1/4} (0.7336)^{0.012} \left(\frac{1.2 \text{ m}}{0.03 \text{ m}} \right)^{-0.3} = 2.076$$

$$h_{air} = \frac{k}{L} Nu = \frac{0.02439 \text{ W/m}\cdot^\circ\text{C}}{0.03 \text{ m}} (2.076) = 1.688 \text{ W/m}^2\cdot^\circ\text{C}$$

Then the rate of heat transfer through this double pane window is determined to be

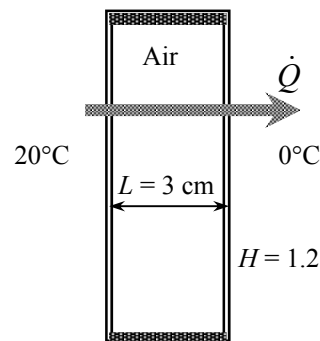
$$A_s = H \times W = (1.2 \text{ m})(2 \text{ m}) = 2.4 \text{ m}^2$$

$$\begin{aligned} \dot{Q} &= \frac{T_{\infty, i} - T_{\infty, o}}{R_{conv, i} + R_{cond, glasses} + R_{conv, air} + R_{conv, o}} = \frac{T_{\infty} - T_{s, i}}{\frac{1}{h_i A_s} + \frac{2t_{glass}}{k_{glass} A_s} + \frac{1}{h_{air} A_s} + \frac{1}{h_o A_s}} \\ &= \frac{20 - 0}{\frac{1}{(10)(2.4)} + \frac{2(0.003)}{(0.78)(2.4)} + \frac{1}{(1.688)(2.4)} + \frac{1}{(25)(2.4)}} = \mathbf{65 \text{ W}} \end{aligned}$$

Check: The temperature drop across the air gap is determined from

$$\dot{Q} = hA_s \Delta T \rightarrow \Delta T = \frac{\dot{Q}}{hA_s} = \frac{65 \text{ W}}{(1.688 \text{ W/m}^2\cdot^\circ\text{C})(2.4 \text{ m}^2)} = 16.0^\circ\text{C}$$

which is very close to the assumed value of 15°C used in the evaluation of the Ra number.



14-86 An electric resistance space heater filled with oil is placed against a wall. The power rating of the heater and the time it will take for the heater to reach steady operation when it is first turned on are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Heat transfer from the back, bottom, and top surfaces are disregarded. 4 The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75 + 25)/2 = 50^\circ\text{C}$ are (Table A-22)

$$k = 0.02735 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7228$$

$$\beta = \frac{1}{T_f} = \frac{1}{(50 + 273)\text{K}} = 0.003096 \text{ K}^{-1}$$

Analysis Heat transfer from the top and bottom surfaces are said to be negligible, and thus the heat transfer area in this case consists of the three exposed side surfaces. The characteristic length is the height of the box, $L_c = L = 0.5 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003096 \text{ K}^{-1})(75 - 25 \text{ K})(0.5 \text{ m})^3}{(1.798 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7228) = 4.244 \times 10^8$$

$$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387 (4.244 \times 10^8)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7228} \right)^{9/16} \right]^{8/27}} \right\}^2 = 94.68$$

$$h = \frac{k}{L} Nu = \frac{0.02735 \text{ W/m}\cdot^\circ\text{C}}{0.5 \text{ m}} (94.68) = 5.179 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (0.5 \text{ m})(0.8 \text{ m}) + 2(0.15 \text{ m})(0.5 \text{ m}) = 0.55 \text{ m}^2$$

and $\dot{Q} = hA_s(T_s - T_\infty) = (5.179 \text{ W/m}^2\cdot^\circ\text{C})(0.55 \text{ m}^2)(75 - 25)^\circ\text{C} = 142.4 \text{ W}$

The radiation heat loss is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.8)(0.55 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(75 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] = 169.1 \text{ W} \end{aligned}$$

Then the total rate of heat transfer, thus the power rating of the heater becomes

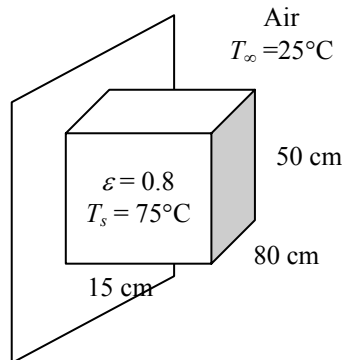
$$\dot{Q}_{total} = 142.4 + 169.1 = \mathbf{311.5 \text{ W}}$$

The specific heat of the oil at the average temperature of the oil is $2006 \text{ J/kg}\cdot^\circ\text{C}$ (Table A-19). Then the amount of heat transfer needed to raise the temperature of the oil to the steady operating temperature and the time it takes become

$$Q = mc_p(T_2 - T_1) = (45 \text{ kg})(2006 \text{ J/kg}\cdot^\circ\text{C})(75 - 25)^\circ\text{C} = 4.514 \times 10^6 \text{ J}$$

$$Q = \dot{Q}\Delta t \longrightarrow \Delta t = \frac{Q}{\dot{Q}} = \frac{4.514 \times 10^6 \text{ kJ}}{311.5 \text{ J/s}} = 14,490 \text{ s} = \mathbf{4.03 \text{ h}}$$

which is not practical. Therefore, the surface temperature of the heater must be allowed to be higher than 75°C .



14-87 A horizontal skylight made of a single layer of glass on the roof of a house is considered. The rate of heat loss through the skylight is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

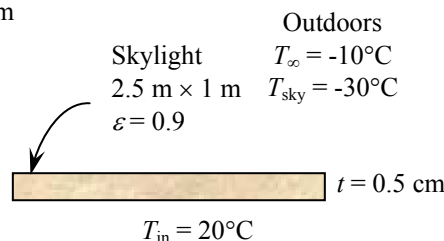
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (-4 - 10)/2 = -7^\circ\text{C}$ are (Table A-22)

$$k = 0.02311 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.278 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.738$$

$$\beta = \frac{1}{T_f} = \frac{1}{(-7 + 273)\text{K}} = 0.003759 \text{ K}^{-1}$$



Analysis We assume radiation heat transfer inside the house to be negligible. We start the calculations by “guessing” the glass temperature to be -4°C for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is

determined from $L_c = \frac{A_s}{p} = \frac{(1 \text{ m})(2.5 \text{ m})}{2(1 \text{ m} + 2.5 \text{ m})} = 0.357 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003759 \text{ K}^{-1})[-4 - (-10) \text{ K}](0.357 \text{ m})^3}{(1.278 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.738) = 4.553 \times 10^7$$

$$Nu = 0.15 Ra^{1/3} = 0.15(4.553 \times 10^7)^{1/3} = 53.56$$

$$h_o = \frac{k}{L_c} Nu = \frac{0.02311 \text{ W/m}\cdot^\circ\text{C}}{0.357 \text{ m}} (53.56) = 3.467 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (1 \text{ m})(2.5 \text{ m}) = 2.5 \text{ m}^2$$

Using the assumed value of glass temperature, the radiation heat transfer coefficient is determined to be

$$\begin{aligned} h_{rad} &= \varepsilon \sigma (T_s + T_{sky})(T_s^2 + T_{sky}^2) \\ &= 0.9(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(-4 + 273) + (-30 + 273)][(-4 + 273)^2 + (-30 + 273)^2] \text{K}^3 \\ &= 3.433 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

Then the combined convection and radiation heat transfer coefficient outside becomes

$$h_{o,combined} = h_o + h_{rad} = 3.467 + 3.433 = 6.90 \text{ W/m}^2$$

Again we take the glass temperature to be -4°C for the evaluation of the properties and h for the inner surface of the skylight. The properties of air at 1 atm and the film temperature of $T_f = (-4 + 20)/2 = 8^\circ\text{C}$ are (Table A-22)

$$k = 0.02424 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.408 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7342$$

$$\beta = \frac{1}{T_f} = \frac{1}{(8 + 273)\text{K}} = 0.003559 \text{ K}^{-1}$$

The characteristic length in this case is also 0.357 m. Then,

$$Ra = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003559 \text{ K}^{-1})[20 - (-4) \text{ K}](0.357 \text{ m})^3}{(1.408 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7342) = 1.412 \times 10^8$$

$$Nu = 0.27 Ra^{1/4} = 0.27(1.412 \times 10^8)^{1/4} = 29.43$$

$$h_i = \frac{k}{L_c} Nu = \frac{0.02424 \text{ W/m}\cdot^\circ\text{C}}{0.357 \text{ m}} (29.43) = 1.998 \text{ W/m}^2\cdot^\circ\text{C}$$

Using the thermal resistance network, the rate of heat loss through the skylight is determined to be

$$\begin{aligned}\dot{Q}_{\text{skylight}} &= \frac{T_{s,i} - T_{\infty,o}}{R_{\text{conv},i} + R_{\text{cond,glas}} + R_{\text{combined},o}} \\ &= \frac{A_s (T_{\text{room}} - T_{\text{out}})}{\frac{1}{h_i} + \frac{t_{\text{glass}}}{k_{\text{glass}}} + \frac{1}{h}} = \frac{(2.5 \text{ m}^2)[20 - (-10)]^\circ\text{C}}{\frac{1}{1.998 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{0.005 \text{ m}}{0.78 \text{ W/m} \cdot ^\circ\text{C}} + \frac{1}{6.90 \text{ W/m}^2 \cdot ^\circ\text{C}}} = 115 \text{ W}\end{aligned}$$

Using the same heat transfer coefficients for simplicity, the rate of heat loss through the roof in the case of R-5.34 construction is determined to be

$$\begin{aligned}\dot{Q}_{\text{roof}} &= \frac{T_{s,i} - T_{\infty,o}}{R_{\text{conv},i} + R_{\text{cond}} + R_{\text{combined},o}} \\ &= \frac{A_s (T_{\text{room}} - T_{\text{out}})}{\frac{1}{h_i} + R_{\text{glass}} + \frac{1}{h}} = \frac{(2.5 \text{ m}^2)[20 - (-10)]^\circ\text{C}}{\frac{1}{1.998 \text{ W/m}^2 \cdot ^\circ\text{C}} + 5.34 \text{ m}^2 \cdot ^\circ\text{C/W} + \frac{1}{6.90 \text{ W/m}^2 \cdot ^\circ\text{C}}} = 12.5 \text{ W}\end{aligned}$$

Therefore, a house loses $115/12.5 \cong 9$ times more heat through the skylights than it does through an insulated wall of the same size.

Using Newton's law of cooling, the glass temperature corresponding to a heat transfer rate of 115 W is calculated to be -3.3°C , which is sufficiently close to the assumed value of -4°C . Therefore, there is no need to repeat the calculations.

14-88 A solar collector consists of a horizontal copper tube enclosed in a concentric thin glass tube. Water is heated in the tube, and the annular space between the copper and glass tube is filled with air. The rate of heat loss from the collector by natural convection is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 Radiation effects are negligible. 3 The pressure of air in the enclosure is 1 atm.

Properties The properties of air at 1 atm and the average temperature of $(T_i + T_o)/2 = (60 + 32)/2 = 46^\circ\text{C}$ are (Table A-22)

$$k = 0.02706 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.760 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7238$$

$$\beta = \frac{1}{T_f} = \frac{1}{(46 + 273)\text{K}} = 0.003135 \text{ K}^{-1}$$

Analysis The characteristic length in this case is the distance between the two cylinders

$$L_c = \frac{D_o - D_i}{2} = \frac{(9 - 5) \text{ cm}}{2} = 2 \text{ cm}$$

and,

$$\text{Ra} = \frac{g\beta(T_i - T_o)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003135 \text{ K}^{-1})(60 - 32 \text{ K})(0.02 \text{ m})^3}{(1.760 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7238) = 16,100$$

The effective thermal conductivity is

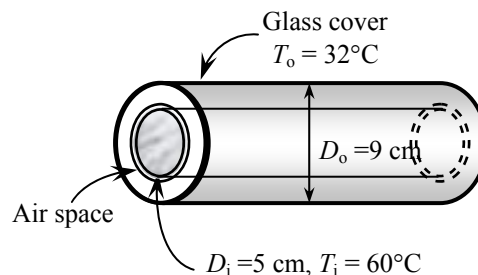
$$F_{\text{cyl}} = \frac{\left[\ln \frac{D_o}{D_i} \right]^4}{L_c^3 (D_i^{-3/5} + D_o^{-3/5})^5} = \frac{\left[\ln \frac{0.09 \text{ m}}{0.05 \text{ m}} \right]^4}{(0.02 \text{ m})^3 [(0.05 \text{ m})^{-7/5} + (0.09 \text{ m})^{-7/5}]^5} = 0.1303$$

$$k_{\text{eff}} = 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyl}} \text{Ra})^{1/4}$$

$$= 0.386(0.02706 \text{ W/m}\cdot^\circ\text{C}) \left(\frac{0.7238}{0.861 + 0.7238} \right)^{1/4} [(0.1303)(16,100)]^{1/4} = 0.05811 \text{ W/m}\cdot^\circ\text{C}$$

Then the heat loss from the collector per meter length of the tube becomes

$$\dot{Q} = \frac{2\pi k_{\text{eff}}}{\ln \left(\frac{D_o}{D_i} \right)} (T_i - T_o) = \frac{2\pi(0.05811 \text{ W/m}\cdot^\circ\text{C})}{\ln \left(\frac{0.09 \text{ m}}{0.05 \text{ m}} \right)} (60 - 32)^\circ\text{C} = 17.4 \text{ W}$$



14-89 A solar collector consists of a horizontal tube enclosed in a concentric thin glass tube is considered. The pump circulating the water fails. The temperature of the aluminum tube when equilibrium is established is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

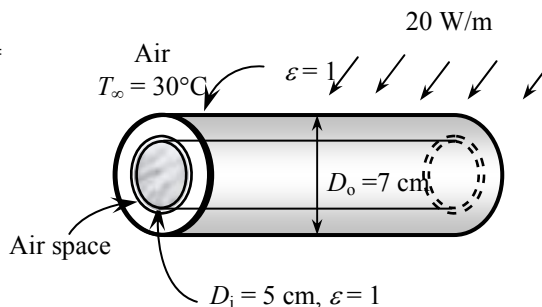
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (33 + 30)/2 = 31.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02599 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.622 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7278$$

$$\beta = \frac{1}{T_f} = \frac{1}{(31.5 + 273)\text{K}} = 0.003284 \text{ K}^{-1}$$



Analysis This problem involves heat transfer from the aluminum tube to the glass cover, and from the outer surface of the glass cover to the surrounding ambient air. When steady operation is reached, these two heat transfers will be equal to the rate of heat gain. That is,

$$\dot{Q}_{\text{tube-glass}} = \dot{Q}_{\text{glass-ambient}} = \dot{Q}_{\text{solar gain}} = 20 \text{ W (per meter length)}$$

Now we assume the surface temperature of the glass cover to be 33°C . We will check this assumption later on, and repeat calculations with a better assumption, if necessary.

The characteristic length for the outer diameter of the glass cover $L_c = D_o = 0.07 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D_o^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003284 \text{ K}^{-1})(33 - 30 \text{ K})(0.07 \text{ m})^3}{(1.622 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7278) = 91,700$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (91,700)^{1/6}}{\left[1 + (0.559 / 0.7278)^{9/16} \right]^{8/27}} \right\}^2 = 7.626$$

$$A_s = \pi D_o L = \pi (0.07 \text{ m})(1 \text{ m}) = 0.2199 \text{ m}^2$$

$$h = \frac{k}{D_o} Nu = \frac{0.02599 \text{ W/m}\cdot^\circ\text{C}}{0.07 \text{ m}} (7.626) = 2.832 \text{ W/m}^2\cdot^\circ\text{C}$$

and,

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) = (2.832 \text{ W/m}^2\cdot^\circ\text{C})(0.2199 \text{ m}^2)(T_{\text{glass}} - 30)^\circ\text{C}$$

The radiation heat loss is

$$\dot{Q}_{\text{rad}} = \epsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4) = (1)(0.2199 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(T_{\text{glass}} + 273 \text{ K})^4 - (20 + 273 \text{ K})^4]$$

The expression for the total rate of heat transfer is

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}}$$

$$20 \text{ W} = (2.832 \text{ W/m}^2\cdot^\circ\text{C})(0.2199 \text{ m}^2)(T_{\text{glass}} - 30)^\circ\text{C} + (1)(0.2199 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4) [(T_{\text{glass}} + 273 \text{ K})^4 - (20 + 273 \text{ K})^4]$$

Its solution is $T_{\text{glass}} = 33.34^\circ\text{C}$, which is sufficiently close to the assumed value of 33°C .

Now we will calculate heat transfer through the air layer between aluminum tube and glass cover. We will assume the aluminum tube temperature to be 45°C and evaluate properties at the average temperature of $(T_i + T_o)/2 = (45 + 33.34)/2 = 39.17^\circ\text{C}$ are (Table A-22)

$$k = 0.02656 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.694 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7257$$

$$\beta = \frac{1}{T_f} = \frac{1}{(39.17 + 273)\text{K}} = 0.003203 \text{ K}^{-1}$$

The characteristic length in this case is the distance between the two cylinders,

$$L_c = (D_o - D_i) / 2 = (7 - 5) / 2 \text{ cm} = 1 \text{ cm}$$

Then,

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003203 \text{ K}^{-1})(45 - 33.34 \text{ K})(0.01 \text{ m})^3}{(1.694 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7257) = 926.5$$

The effective thermal conductivity is

$$F_{\text{cyl}} = \frac{\left[\ln \frac{D_o}{D_i} \right]^4}{L_c^3 (D_i^{-3/5} + D_o^{-3/5})^5} = \frac{\left[\ln \frac{0.07 \text{ m}}{0.05 \text{ m}} \right]^4}{(0.01 \text{ m})^3 [(0.05 \text{ m})^{-3/5} + (0.07 \text{ m})^{-3/5}]^5} = 0.08085$$

$$k_{\text{eff}} = 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyl}} \text{Ra})^{1/4}$$

$$= 0.386(0.02656 \text{ W/m}\cdot^\circ\text{C}) \left(\frac{0.7257}{0.861 + 0.7257} \right)^{1/4} [(0.08085)(926.5)]^{1/4} = 0.02480 \text{ W/m}\cdot^\circ\text{C}$$

The heat transfer expression is

$$\dot{Q} = \frac{2\pi k_{\text{eff}}}{\ln \left(\frac{D_o}{D_i} \right)} (T_1 - T_2) = \frac{2\pi(0.02480 \text{ W/m}\cdot^\circ\text{C})}{\ln \left(\frac{0.07 \text{ m}}{0.05 \text{ m}} \right)} (T_{\text{tube}} - 33.34)^\circ\text{C}$$

The radiation heat loss is

$$A_s = \pi D_i L = \pi (0.05 \text{ m})^2 (1 \text{ m}) = 0.1571 \text{ m}^2$$

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma (T_s^4 - T_{\text{surr}}^4)$$

$$= (1)(0.1571 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(T_{\text{tube}} + 273 \text{ K})^4 - (33.34 + 273 \text{ K})^4]$$

The expression for the total rate of heat transfer is

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}}$$

$$20 \text{ W} = \frac{2\pi(0.02480 \text{ W/m}\cdot^\circ\text{C})}{\ln \left(\frac{0.07 \text{ m}}{0.05 \text{ m}} \right)} (T_{\text{tube}} - 33.34)^\circ\text{C}$$

$$+ (1)(0.1571 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(T_{\text{tube}} + 273 \text{ K})^4 - (33.34 + 273 \text{ K})^4]$$

Its solution is $T_{\text{tube}} = 46.3^\circ\text{C}$,

which is sufficiently close to the assumed value of 45°C. Therefore, there is no need to repeat the calculations.

14-90E The components of an electronic device located in a horizontal duct of rectangular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct by natural convection and the average temperature of the duct are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Radiation effects are negligible. 5 The thermal resistance of the duct is negligible.

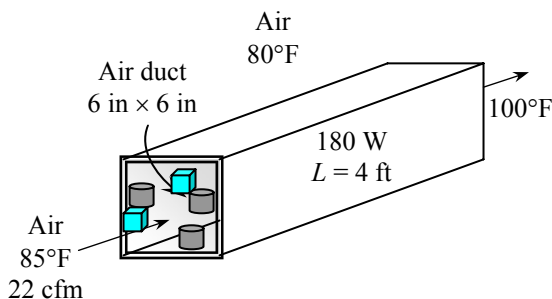
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (120 + 80)/2 = 100^\circ\text{F}$ are (Table A-22E)

$$k = 0.01529 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1809 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.726$$

$$\beta = 1/T_f = 1/(100 + 460)\text{R} = 0.001786 \text{ R}^{-1}$$



Analysis (a) Using air density at the inlet temperature of 85°F and the specific heat at the average temperature of $(85 + 100)/2 = 92.5^\circ\text{F}$ and 1 atm for the forced air, the mass flow rate of air and the heat transfer rate by forced convection are determined to be

$$\dot{m} = \rho \dot{V} = (0.07284 \text{ lbm/ft}^3)(22 \text{ ft}^3/\text{min}) = 1.602 \text{ lbm/min}$$

$$\dot{Q}_{\text{forced}} = \dot{m} c_p (T_{\text{out}} - T_{\text{in}}) = (1.602 \times 60 \text{ lbm/h})(0.2404 \text{ Btu/lbm}\cdot^\circ\text{F})(100 - 85)^\circ\text{F} = 346.6 \text{ Btu/h}$$

Noting that radiation heat transfer is negligible, the rest of the 180 W heat generated must be dissipated by natural convection,

$$\dot{Q}_{\text{natural}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{forced}} = (180 \times 3.412) - 346.6 = 268 \text{ Btu/h}$$

(b) We start the calculations by “guessing” the surface temperature to be 120°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary.

Horizontal top surface: The characteristic length is $L_c = \frac{A_s}{P} = \frac{(4 \text{ ft})(6/12 \text{ ft})}{2(4 \text{ ft} + 6/12 \text{ ft})} = 0.2222 \text{ ft}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001786 \text{ R}^{-1})(120 - 80 \text{ R})(0.2222 \text{ ft})^3}{(0.1809 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.726) = 5.599 \times 10^5$$

$$Nu = 0.54 Ra^{1/4} = 0.54(5.599 \times 10^5)^{1/4} = 14.77$$

$$h_{\text{top}} = \frac{k}{L_c} Nu = \frac{0.01529 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.2222 \text{ ft}} (14.77) = 1.016 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_{\text{top}} = (4 \text{ ft})(6/12 \text{ ft}) = 2 \text{ ft}^2 = A_{\text{bottom}}$$

Horizontal bottom surface: The Nusselt number for this geometry and orientation can be determined from

$$Nu = 0.27 Ra^{1/4} = 0.27(5.599 \times 10^5)^{1/4} = 7.386$$

$$h_{\text{bottom}} = \frac{k}{L_c} Nu = \frac{0.01529 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.2222 \text{ ft}} (7.386) = 0.5082 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

Vertical side surfaces: The characteristic length in this case is the height of the duct, $L_c = L = 6 \text{ in}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001786 \text{ R}^{-1})(120 - 80 \text{ R})(0.5 \text{ ft})^3}{(0.1809 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.726) = 6.379 \times 10^6$$

$$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{\left[1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387 (6.379 \times 10^6)^{1/6}}{\left[1 + \left(\frac{0.492}{0.726} \right)^{9/16} \right]^{8/27}} \right\}^2 = 27.57$$

$$h_{side} = \frac{k}{L} Nu = \frac{0.01529 \text{ Btu/h.ft.}^\circ\text{F}}{0.5 \text{ ft}} (27.57) = 0.843 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

$$A_{side} = 2(4 \text{ ft})(0.5 \text{ ft}) = 4 \text{ ft}^2$$

Then the total heat loss from the duct can be expressed as

$$\dot{Q}_{total} = \dot{Q}_{top} + \dot{Q}_{bottom} + \dot{Q}_{side} = [(hA)_{top} + (hA)_{bottom} + (hA)_{side}](T_s - T_\infty)$$

Substituting and solving for the surface temperature,

$$268 \text{ Btu/h} = [(1.016 \times 2 + 0.5082 \times 2 + 0.843 \times 4) \text{ Btu/h.}^\circ\text{F}](T_s - 80)^\circ\text{F}$$

$$T_s = \mathbf{121.7^\circ\text{F}}$$

which is sufficiently close to the assumed value of 120°F used in the evaluation of properties and h . Therefore, there is no need to repeat the calculations.

14-91E The components of an electronic system located in a horizontal duct of circular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct by natural convection and the average temperature of the duct are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Radiation effects are negligible. 5 The thermal resistance of the duct is negligible.

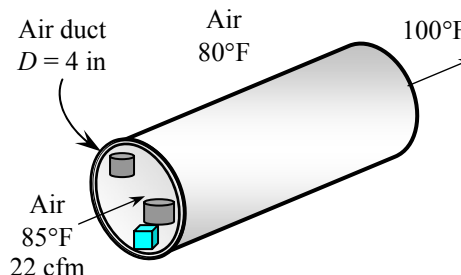
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (150 + 80)/2 = 115^\circ\text{F}$ are (Table A-22E)

$$k = 0.01564 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1895 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7238$$

$$\beta = \frac{1}{T_f} = \frac{1}{(115 + 460) \text{ R}} = 0.001739 \text{ R}^{-1}$$



Analysis (a) Using air density at the inlet temperature of 85°F and the specific heat at the average temperature of $(85 + 100)/2 = 92.5^\circ\text{F}$ and 1 atm for the forced air, the mass flow rate of air and the heat transfer rate by forced convection are determined to be

$$\dot{m} = \rho \dot{V} = (0.07284 \text{ lbm/ft}^3)(22 \text{ ft}^3/\text{min}) = 1.602 \text{ lbm/min}$$

$$\dot{Q}_{\text{forced}} = \dot{m} c_p (T_{\text{out}} - T_{\text{in}}) = (1.602 \times 60 \text{ lbm/h})(0.2404 \text{ Btu/lbm}\cdot^\circ\text{F})(100 - 85)^\circ\text{F} = 346.6 \text{ Btu/h}$$

Noting that radiation heat transfer is negligible, the rest of the 180 W heat generated must be dissipated by natural convection,

$$\dot{Q}_{\text{natural}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{forced}} = (180 \times 3.412) - 346.6 = 268 \text{ Btu/h}$$

(b) We start the calculations by “guessing” the surface temperature to be 150°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the duct, $L_c = D = 4 \text{ in}$. Then,

$$Ra = \frac{g\beta(T_1 - T_2)D^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001739 \text{ R}^{-1})(150 - 80 \text{ R})(4/12 \text{ ft})^3}{(0.1895 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7238) = 2.926 \times 10^6$$

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (2.926 \times 10^6)^{1/6}}{\left[1 + (0.559 / 0.7238)^{9/16} \right]^{8/27}} \right\}^2 = 19.79$$

$$h = \frac{k}{D} Nu = \frac{0.01564 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{4/12 \text{ ft}} (19.79) = 0.9285 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_s = \pi DL = \pi (4/12 \text{ ft})(4 \text{ ft}) = 4.19 \text{ ft}^2$$

Then the surface temperature is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 80^\circ\text{F} + \frac{268 \text{ Btu/h}}{(0.9285 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F})(4.19 \text{ ft}^2)} = 148.9^\circ\text{F}$$

which is very close to the assumed value of 150°F used in the evaluation of properties and h . Therefore, there is no need to repeat the calculations.

14-92E The components of an electronic system located in a horizontal duct of rectangular cross section is cooled by natural convection. The heat transfer from the outer surfaces of the duct by natural convection and the average temperature of the duct are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Radiation effects are negligible. 5 The thermal resistance of the duct is negligible.

Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (160 + 80)/2 = 120^\circ\text{F}$ are (Table A-22E)

$$k = 0.01576 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1923 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.723$$

$$\beta = 1/T_f = 1/(120 + 460 \text{ R}) = 0.001724 \text{ R}^{-1}$$

Analysis (a) Noting that radiation heat transfer is negligible and no heat is removed by forced convection because of the failure of the fan, the entire 180 W heat generated must be dissipated by natural convection,

$$\dot{Q}_{\text{natural}} = \dot{Q}_{\text{total}} = 180 \text{ W}$$

(b) We start the calculations by “guessing” the surface temperature to be 160°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary.

Horizontal top surface: The characteristic length is $L_c = \frac{A_s}{p} = \frac{(4 \text{ ft})(6/12 \text{ ft})}{2(4 \text{ ft} + 6/12 \text{ ft})} = 0.2222 \text{ ft}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001724 \text{ R}^{-1})(160 - 80 \text{ R})(0.2222 \text{ ft})^3}{(0.1923 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.723) = 9.534 \times 10^5$$

$$\text{Nu} = 0.54 \text{Ra}^{1/4} = 0.54(9.534 \times 10^5)^{1/4} = 16.87$$

$$h_{\text{top}} = \frac{k}{L_c} \text{Nu} = \frac{0.01576 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.2222 \text{ ft}} (16.87) = 1.197 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_{\text{top}} = (4 \text{ ft})(6/12 \text{ ft}) = 2 \text{ ft}^2 = A_{\text{bottom}}$$

Horizontal bottom surface: The Nusselt number for this geometry and orientation can be determined from

$$\text{Nu} = 0.27 \text{Ra}^{1/4} = 0.27(9.534 \times 10^5)^{1/4} = 8.437$$

$$h_{\text{bottom}} = \frac{k}{L_c} \text{Nu} = \frac{0.01576 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.2222 \text{ ft}} (8.437) = 0.5983 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

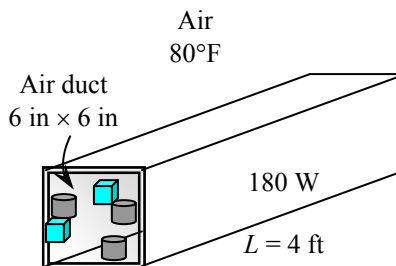
Vertical side surfaces: The characteristic length in this case is the height of the duct, $L_c = L = 6 \text{ in}$. Then,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.001724 \text{ R}^{-1})(160 - 80 \text{ R})(0.5 \text{ ft})^3}{(0.1923 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.723) = 1.086 \times 10^7$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.086 \times 10^7)^{1/6}}{\left[1 + \left(\frac{0.492}{0.723} \right)^{9/16} \right]^{8/27}} \right\}^2 = 32.03$$

$$h_{\text{side}} = \frac{k}{L} \text{Nu} = \frac{0.01576 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{0.5 \text{ ft}} (32.03) = 1.009 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_{\text{side}} = 2(4 \text{ ft})(0.5 \text{ ft}) = 4 \text{ ft}^2$$



Then the total heat loss from the duct can be expressed as

$$\dot{Q}_{total} = \dot{Q}_{top} + \dot{Q}_{bottom} + \dot{Q}_{side} = [(hA)_{top} + (hA)_{bottom} + (hA)_{side}](T_s - T_\infty)$$

Substituting and solving for the surface temperature,

$$180 \text{ W} \left(\frac{3.41214 \text{ Btu/h}}{1 \text{ W}} \right) = [(1.197 \times 2 + 0.5983 \times 2 + 1.009 \times 4) \text{ Btu/h} \cdot ^\circ\text{F}](T_s - 80)^\circ\text{F}$$

$$T_s = \mathbf{160.5^\circ\text{F}}$$

which is sufficiently close to the assumed value of 160°F used in the evaluation of properties and h . Therefore, there is no need to repeat the calculations.

14-93 A cold aluminum canned drink is exposed to ambient air. The time it will take for the average temperature to rise to a specified value is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Any heat transfer from the bottom surface of the can is disregarded. 5 The thermal resistance of the can is negligible.

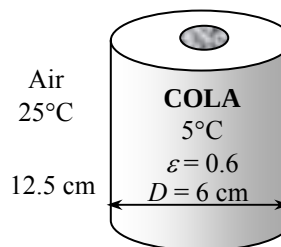
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (6 + 25)/2 = 15.5^\circ\text{C}$ are (Table A-22)

$$k = 0.0248 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.475 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7322$$

$$\beta = 1/T_f = 1/(15.5 + 273 \text{ K}) = 0.003466 \text{ K}^{-1}$$



Analysis We assume the surface temperature of aluminum can to be equal to the temperature of the drink in the can since the can is made of a very thin layer of aluminum. Noting that the temperature of the drink rises from 5°C to 7°C , we take the average surface temperature to be 6°C . The characteristic length in this case is the height of the box $L_c = L = 0.125 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003466 \text{ K}^{-1})(25 - 6 \text{ K})(0.125 \text{ m})^3}{(1.475 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7322) = 4.246 \times 10^6$$

At this point we should check if we can treat this aluminum can as a vertical plate. The criteria is

$$D \geq \frac{35L}{\text{Gr}^{1/4}} \longrightarrow \frac{35(12.5 \text{ cm})}{(4.246 \times 10^6 / 0.7322)^{1/4}} = 8.92 \text{ cm}$$

which is not smaller than the diameter of the can (6 cm), but close to it. Therefore, we can still use vertical plate relation approximately (besides, we do not have another relation available). Then

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(4.246 \times 10^6)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7322} \right)^{9/16} \right]^{8/27}} \right\}^2 = 24.63$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.0248 \text{ W/m}\cdot^\circ\text{C}}{0.125 \text{ m}} (24.63) = 4.887 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$A_s = \pi DL + \frac{\pi D^2}{4} = \pi(0.06 \text{ m})(0.125 \text{ m}) + \frac{\pi(0.06 \text{ m})^2}{4} = 0.02639 \text{ m}^2$$

Note that we also include top surface area of the can to the total surface area, and assume the heat transfer coefficient for that area to be the same for simplicity (actually, it will be a little lower). Then heat transfer rate from outer surfaces of the can by natural convection becomes

$$\dot{Q} = hA_s(T_\infty - T_s) = (4.887 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02639 \text{ m}^2)(25 - 6)^\circ\text{C} = 2.45 \text{ W}$$

The radiation heat loss is

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) = (0.6)(0.02639 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(298 \text{ K})^4 - (6 + 273 \text{ K})^4] = 1.64 \text{ W}$$

and $\dot{Q}_{\text{total}} = 2.41 + 1.64 = 4.09 \text{ W}$

Using the properties of water for the cold drink at 6°C , the amount of heat transfer to the drink is

$$m = \rho V = \rho \frac{\pi D^2}{4} L = (999.9 \text{ kg/m}^3) \frac{\pi(0.06 \text{ m})^2}{4} (0.125 \text{ m}) = 0.3534 \text{ kg}$$

$$Q = mc_p(T_2 - T_1) = (0.3534 \text{ kg})(4203 \text{ J/kg}\cdot^\circ\text{C})(7 - 5)^\circ\text{C} = 2971 \text{ J}$$

Then the time required for the temperature of the cold drink to rise to 7°C becomes

$$Q = \dot{Q}\Delta t \longrightarrow \Delta t = \frac{Q}{\dot{Q}} = \frac{2971 \text{ J}}{4.09 \text{ J/s}} = 726 \text{ s} = \mathbf{12.1 \text{ min}}$$

14-94 An electric hot water heater is located in a small room. A hot water tank insulation kit is available for \$60. The payback period of this insulation to pay for itself from the energy it saves is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Any heat transfer from the top and bottom surfaces of the tank is disregarded. 5 The thermal resistance of the metal sheet is negligible.

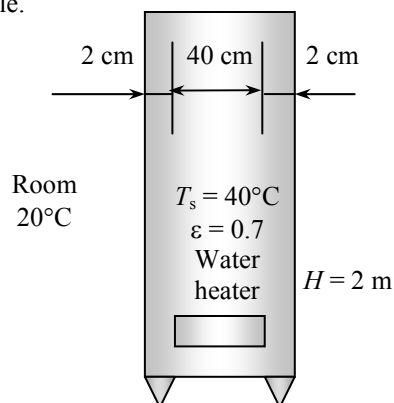
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (40 + 20)/2 = 30^\circ\text{C}$ are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7282$$

$$\beta = \frac{1}{T_f} = \frac{1}{(30 + 273)\text{K}} = 0.0033 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the height of the heater, $L_c = L = 2 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.0033 \text{ K}^{-1})(40 - 20 \text{ K})(2 \text{ m})^3}{(1.608 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7282) = 1.459 \times 10^{10}$$

$$\text{Nu} = \left\{ 0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387(1.459 \times 10^{10})^{1/6}}{\left[1 + \left(\frac{0.492}{0.7282} \right)^{9/16} \right]^{8/27}} \right\}^2 = 285.4$$

$$h = \frac{k}{L} \text{Nu} = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{2 \text{ m}} (285.4) = 3.693 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.44 \text{ m})(2 \text{ m}) = 2.765 \text{ m}^2$$

and $\dot{Q} = hA_s(T_\infty - T_s) = (3.693 \text{ W/m}^2\cdot^\circ\text{C})(2.765 \text{ m}^2)(40 - 20)^\circ\text{C} = 204.2 \text{ W}$

The radiation heat loss is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) \\ &= (0.7)(2.765 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(40 + 273 \text{ K})^4 - (20 + 273 \text{ K})^4] = 244.5 \text{ W} \end{aligned}$$

and $\dot{Q}_{\text{total}} = 204.2 + 244.5 = 448.7 \text{ W}$

The reduction in heat loss after adding insulation is

$$\dot{Q} = (0.80)(448.7) = 359.0 \text{ W}$$

The amount of heat and money saved per hour is

$$Q_{\text{saved}} = \dot{Q}_{\text{saved}} \Delta t = (0.3590 \text{ kW})(1 \text{ h}) = 0.3590 \text{ kWh}$$

$$\text{Money saved} = (0.3590 \text{ kWh})(\$0.08/\text{kWh}) = \$0.02872$$

Then it will take $\Delta t = \frac{\$60}{\$0.02872} = 2089 \text{ h} = \mathbf{87.0 \text{ days}}$

for the additional insulation to pay for itself from the energy it saves.

14-95 A hot part of the vertical front section of a natural gas furnace in a plant is considered. The rate of heat loss from this section and the annual cost of this heat loss are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Any heat transfer from other surfaces of the tank is disregarded.

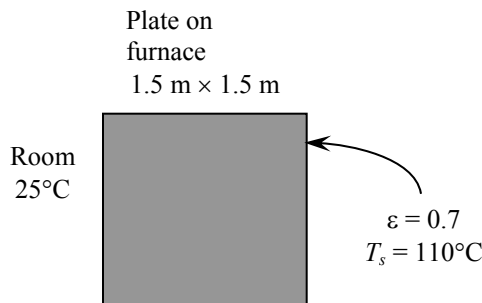
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (110 + 25)/2 = 67.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02863 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.97 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7183$$

$$\beta = \frac{1}{T_f} = \frac{1}{(67.5 + 273)\text{K}} = 0.002937 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the height of that section of furnace, $L_c = L = 1.5 \text{ m}$. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002937 \text{ K}^{-1})(110 - 25 \text{ K})(1.5 \text{ m})^3}{(1.97 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7183) = 1.530 \times 10^{10}$$

$$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387 (1.530 \times 10^{10})^{1/6}}{\left[1 + \left(\frac{0.492}{0.7183} \right)^{9/16} \right]^{8/27}} \right\}^2 = 289.1$$

$$h = \frac{k}{L} Nu = \frac{0.02863 \text{ W/m}\cdot^\circ\text{C}}{1.5 \text{ m}} (289.1) = 5.518 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (1 \text{ m})(1.5 \text{ m}) = 1.5 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.518 \text{ W/m}^2\cdot^\circ\text{C})(1.5 \text{ m}^2)(110 - 25)^\circ\text{C} = 703.5 \text{ W}$$

The radiation heat loss is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_{surr}^4 - T_s^4) \\ &= (0.7)(1.5 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(110 + 273 \text{ K})^4 - (25 + 273 \text{ K})^4] = 812 \text{ W} \end{aligned}$$

$$\dot{Q}_{total} = 703.5 + 812 = \mathbf{1515 \text{ W}}$$

The amount and cost of natural gas used to overcome this heat loss per year is

$$Q_{gas} = \dot{Q}_{gas} \Delta t = \frac{\dot{Q}_{total}}{0.79} \Delta t = \frac{(1.515 \text{ kJ/s})}{0.79} (310 \text{ days/yr} \times 10 \text{ hr/day} \times 3600 \text{ s/hr}) = 2.14 \times 10^7 \text{ kJ}$$

$$\text{Cost} = (2.14 \times 10^7 / 105,500 \text{ therm})(\$1.20/\text{therm}) = \mathbf{\$243}$$

14-96 A group of 25 transistors are cooled by attaching them to a square aluminum plate and mounting the plate on the wall of a room. The required size of the plate to limit the surface temperature to 50°C is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Any heat transfer from the back side of the plate is negligible.

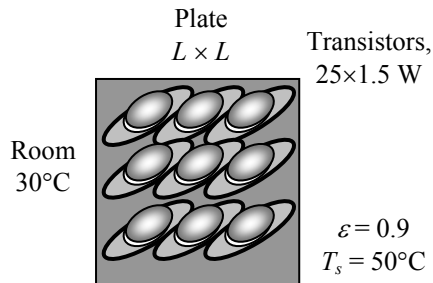
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (50 + 30)/2 = 40^\circ\text{C}$ are (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7255$$

$$\beta = \frac{1}{T_f} = \frac{1}{(40 + 273)\text{K}} = 0.003195 \text{ K}^{-1}$$



Analysis The Rayleigh number can be determined in terms of the characteristic length (length of the plate) to be

$$Ra = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003195 \text{ K}^{-1})(50 - 30 \text{ K})(L)^3}{(1.702 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7255) = 1.571 \times 10^9 L^3$$

The Nusselt number relation is

$$Nu = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.825 + \frac{0.387 (1.571 \times 10^9 L^3)^{1/6}}{\left[1 + \left(\frac{0.492}{0.7255} \right)^{9/16} \right]^{8/27}} \right\}^2$$

The heat transfer coefficient is

$$h = \frac{k}{L} Nu = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{L} Nu$$

$$A_s = L^2$$

Noting that both the surface and surrounding temperatures are known, the rate of convection and radiation heat transfer are expressed as

$$\dot{Q}_{\text{conv}} = h A_s (T_s - T_\infty) = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{L} Nu L^2 (50 - 30)^\circ\text{C}$$

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma (T_s^4 - T_{\text{sky}}^4) = (0.9) L^2 (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(50 + 273)^4 - (30 + 273)^4] \text{K}^4 = 125.3 L^2$$

The rate of total heat transfer is expressed as

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}}$$

$$25 \times (1.5 \text{ W}) = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{L} Nu L^2 (50 - 30)^\circ\text{C} + 125.3 L^2$$

Substituting Nusselt number expression above into this equation and solving for L , the length of the plate is determined to be

$$L = 0.426 \text{ m}$$

14-97 A group of 25 transistors are cooled by attaching them to a square aluminum plate and positioning the plate horizontally in a room. The required size of the plate to limit the surface temperature to 50°C is to be determined for two cases.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 Any heat transfer from the back side of the plate is negligible.

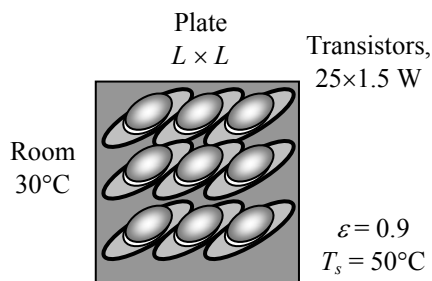
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (50 + 30)/2 = 40^\circ\text{C}$ are (Table A-22)

$$k = 0.02662 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.702 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7255$$

$$\beta = \frac{1}{T_f} = \frac{1}{(40 + 273)\text{K}} = 0.003195 \text{ K}^{-1}$$



Analysis The characteristic length and the Rayleigh number for the horizontal case are determined to be

$$L_c = \frac{A_s}{p} = \frac{L^2}{4L} = \frac{L}{4}$$

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003195 \text{ K}^{-1})(50 - 30 \text{ K})(L/4)^3}{(1.702 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7255) = 2.454 \times 10^7 L^3$$

Noting that both the surface and surrounding temperatures are known, the rate of radiation heat transfer is determined to be

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma (T_s^4 - T_{\text{sky}}^4) = (0.9)L^2 (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(50 + 273)^4 - (30 + 273)^4] \text{K}^4 = 125.3 L^2$$

(a) **Hot surface facing up:** We assume $\text{Ra} < 10^7$ and thus $L < 0.74 \text{ m}$ so that we can determine the Nu number from Eq. 14-22. Then the Nusselt number and the convection heat transfer coefficient become

$$\text{Nu} = 0.54 \text{Ra}^{1/4} = 0.54(2.454 \times 10^7 L^3)^{1/4} = 38.0 L^{3/4}$$

Then,

$$h = \frac{k}{L} \text{Nu} = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{L/4} (38.0 L^{3/4}) = 4.047 L^{-1/4} \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$A_s = L^2$$

The rate of convection heat transfer is

$$\dot{Q}_{\text{conv}} = h A_s (T_s - T_\infty) = (4.047 L^{-1/4}) L^2 (50 - 30) = 80.94 L^{7/4} \text{ W}$$

Then,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} \longrightarrow 25 \times (1.5 \text{ W}) = 80.94 L^{7/4} + 125.3 L^2 \text{ W}$$

Solving for L , the length of the plate is determined to be

$$L = \mathbf{0.407 \text{ m}}$$

Note that $L < 0.75 \text{ m}$, and therefore the assumption of $\text{Ra} < 10^7$ is verified. That is,

(b) **Hot surface facing down:** The Nusselt number in this case is determined from

$$\text{Nu} = 0.27 \text{Ra}^{1/4} = 0.27(2.454 \times 10^7 L^3)^{1/4} = 19.0 L^{3/4}$$

$$\text{Then, } h = \frac{k}{L_c} \text{Nu} = \frac{0.02662 \text{ W/m}\cdot^\circ\text{C}}{L/4} (19.0 L^{3/4}) = 2.023 L^{-1/4}$$

The rate of convection heat transfer is

$$\dot{Q}_{\text{conv}} = h A_s (T_s - T_\infty) = (2.023 L^{-1/4}) L^2 (50 - 30) = 40.47 L^{7/4} \text{ W}$$

Then,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} \longrightarrow 25 \times (1.5 \text{ W}) = 40.47 L^{7/4} + 125.3 L^2 \text{ W}$$

Solving for L , the length of the plate is determined to be

$$L = \mathbf{0.464 \text{ m}}$$

14-98E A hot water pipe passes through a basement. The temperature drop of water in the basement due to heat loss from the pipe is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

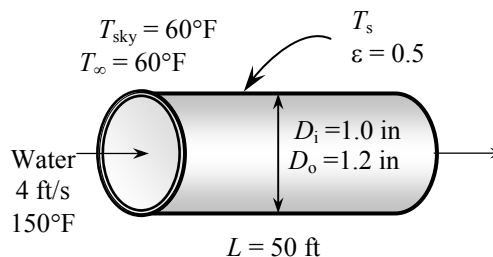
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s + T_\infty)/2 = (150 + 60)/2 = 105^\circ\text{F}$ are (Table A-22E)

$$k = 0.01541 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

$$\nu = 0.1838 \times 10^{-3} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7253$$

$$\beta = \frac{1}{T_f} = \frac{1}{(105 + 460)\text{R}} = 0.00177 \text{ R}^{-1}$$



Analysis We expect the pipe temperature to be very close to the water temperature, and start the calculations by “guessing” the average outer surface temperature of the pipe to be 150°F for the evaluation of the properties and h . We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the pipe, $L_c = D_o = 1.2$ in. Then,

$$Ra = \frac{g\beta(T_\infty - T_s)D_o^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)(0.00177 \text{ R}^{-1})(150 - 60 \text{ R})(1.2/12 \text{ ft})^3}{(0.1838 \times 10^{-3} \text{ ft}^2/\text{s})^2} (0.7253) = 1.101 \times 10^5$$

The natural convection Nusselt number can be determined from

$$Nu = \left\{ 0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559/\text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (1.101 \times 10^5)^{1/6}}{\left[1 + (0.559/0.7253)^{9/16} \right]^{8/27}} \right\}^2 = 7.998$$

$$h_o = \frac{k}{D_o} Nu = \frac{0.01541 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}}{(1.2/12) \text{ ft}} (7.998) = 1.232 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$$

$$A_i = \pi D_i L = \pi (1/12 \text{ ft})(50 \text{ ft}) = 13.09 \text{ ft}^2$$

$$A_o = \pi D_o L = \pi (1.2/12 \text{ ft})(50 \text{ ft}) = 15.708 \text{ ft}^2$$

Using the assumed value of glass temperature, the radiation heat transfer coefficient is determined to be

$$\begin{aligned} h_{rad} &= \varepsilon \sigma (T_s + T_{surr})(T_s^2 + T_{surr}^2) \\ &= (0.5)(0.1714 \times 10^{-8} \text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)[(150 + 460) + (60 + 460)][(150 + 460)^2 + (60 + 460)^2] \text{R}^3 \\ &= 0.6222 \text{ Btu/h}\cdot\text{ft}^2\cdot\text{R} \end{aligned}$$

Then the combined convection and radiation heat transfer coefficient outside becomes

$$h_{o,combined} = h_o + h_{rad} = 1.232 + 0.6222 = 1.854 \text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

$$\text{and } \dot{Q} = \frac{T_{\text{water}} - T_\infty}{\frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{4\pi k L} + \frac{1}{h_o A_o}} = \frac{150 - 60}{\frac{1}{(30)(13.09)} + \frac{\ln(1.2/1)}{4\pi(30)(50)} + \frac{1}{(1.854)(15.708)}} = 2440 \text{ Btu/h}$$

The mass flow rate of water

$$\dot{m} = \rho A_c V = (61.2 \text{ lbm/ft}^3) \left[\pi (1/12 \text{ ft})^2 / 4 \right] (4 \text{ ft/s}) = 1.335 \text{ lbm/s} = 4807 \text{ lbm/h}$$

Then the temperature drop of water as it flows through the pipe becomes

$$\dot{Q} = \dot{m} c_p \Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m} c_p} = \frac{2440 \text{ Btu/h}}{(4807 \text{ lbm/h})(1.0 \text{ Btu/lbm}\cdot^\circ\text{F})} = 0.51^\circ\text{F}$$

14-99 A flat-plate solar collector placed horizontally on the flat roof of a house is exposed to the calm ambient air. The rate of heat loss from the collector by natural convection and radiation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm.

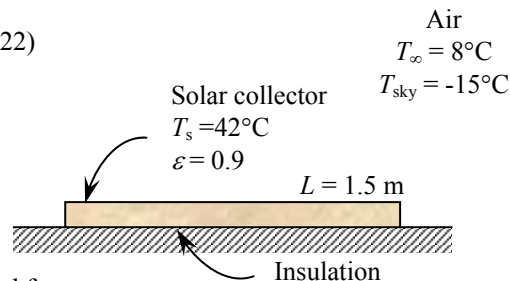
Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (42 + 8)/2 = 25^\circ\text{C}$ are (Table A-22)

$$k = 0.02551 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7296$$

$$\beta = \frac{1}{T_f} = \frac{1}{(25 + 273)\text{K}} = 0.003356 \text{ K}^{-1}$$



Analysis The characteristic length in this case is determined from

$$L_c = \frac{A_s}{p} = \frac{(1.5 \text{ m})(6 \text{ m})}{2(1.5 \text{ m} + 6 \text{ m})} = 0.6 \text{ m}$$

Then,

$$Ra = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003356 \text{ K}^{-1})(42 - 8 \text{ K})(0.6 \text{ m})^3}{(1.562 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7296) = 7.230 \times 10^8$$

$$Nu = 0.15 Ra^{1/3} = 0.15(7.230 \times 10^8)^{1/3} = 134.6$$

$$h = \frac{k}{L_c} Nu = \frac{0.02551 \text{ W/m}\cdot^\circ\text{C}}{0.6 \text{ m}} (134.6) = 5.72 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (1.5 \text{ m})(6 \text{ m}) = 9 \text{ m}^2$$

and

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (5.72 \text{ W/m}^2\cdot^\circ\text{C})(9 \text{ m}^2)(42 - 8)^\circ\text{C} = \mathbf{1750 \text{ W}}$$

Heat transfer rate by radiation is

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_{surr}^4 - T_s^4) \\ &= (0.9)(9 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(42 + 273 \text{ K})^4 - (-15 + 273 \text{ K})^4] = \mathbf{2490 \text{ W}} \end{aligned}$$

14-100 A flat-plate solar collector tilted 40° from the horizontal is exposed to the calm ambient air. The total rate of heat loss from the collector, the collector efficiency, and the temperature rise of water in the collector are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant properties. 3 The local atmospheric pressure is 1 atm. 4 There is no heat loss from the back surface of the absorber plate.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (40 + 20)/2 = 30^\circ\text{C}$ are (Table A-22)

$$k = 0.02588 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.608 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7282$$

$$\beta = \frac{1}{T_f} = \frac{1}{(30 + 273)\text{K}} = 0.0033 \text{ K}^{-1}$$

Analysis (a) The characteristic length in this case is determined from

$$L_c = \frac{A_s}{p} = \frac{(1.5 \text{ m})(2 \text{ m})}{2(1.5 \text{ m} + 2 \text{ m})} = 0.429 \text{ m}^2$$

Then,

$$\begin{aligned} Ra &= \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2} \text{Pr} \\ &= \frac{(9.81 \text{ m/s}^2)(\cos 40^\circ)(0.0033 \text{ K}^{-1})(40 - 20 \text{ K})(0.429 \text{ m})^3}{(1.608 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7282) = 1.103 \times 10^8 \end{aligned}$$

$$Nu = 0.15 Ra^{1/3} = 0.15(1.103 \times 10^8)^{1/3} = 71.94$$

$$h = \frac{k}{L_s} Nu = \frac{0.02588 \text{ W/m}\cdot^\circ\text{C}}{0.429 \text{ m}} (71.94) = 4.340 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_s = (1.5 \text{ m})(2 \text{ m}) = 3 \text{ m}^2$$

and $\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) = (4.340 \text{ W/m}^2\cdot^\circ\text{C})(3 \text{ m}^2)(40 - 20)^\circ\text{C} = 260.4 \text{ W}$

Heat transfer rate by radiation is

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) \\ &= (0.9)(3 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(40 + 273 \text{ K})^4 - (-40 + 273 \text{ K})^4] = 1018 \text{ W} \end{aligned}$$

and $\dot{Q}_{\text{total}} = 260.4 + 1018 = \mathbf{1278 \text{ W}}$

(b) The solar energy incident on the collector is

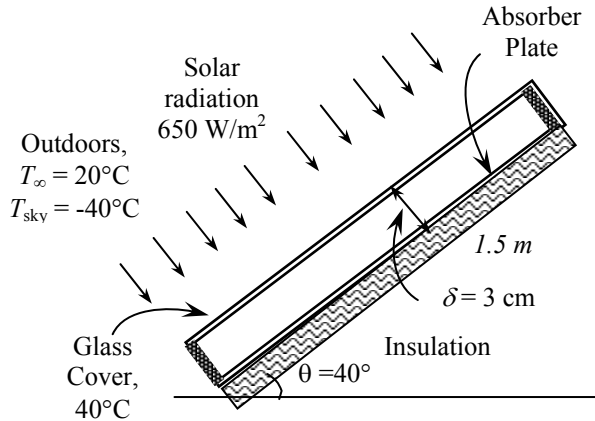
$$\dot{Q}_{\text{incident}} = \alpha \dot{q}_s A_s = (0.88)(650 \text{ W/m}^2)(3 \text{ m}^2) = 1716 \text{ W}$$

Then the collector efficiency becomes

$$\text{efficiency} = \frac{\dot{Q}_{\text{incident}} - \dot{Q}_{\text{lost}}}{\dot{Q}_{\text{incident}}} = \frac{1716 - 1278}{1716} = 0.255 = \mathbf{25.5\%}$$

(c) The temperature rise of the water as it passes through the collector is

$$\dot{Q} = \dot{m} c_p \Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m} c_p} = \frac{(1716 - 1278) \text{ W}}{(1/60 \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})} = \mathbf{6.3^\circ\text{C}}$$



14-101 14-103 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 15
RADIATION HEAT TRANSFER

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Electromagnetic and Thermal Radiation

15-1C Electromagnetic waves are caused by accelerated charges or changing electric currents giving rise to electric and magnetic fields. Sound waves are caused by disturbances. Electromagnetic waves can travel in vacuum, sound waves cannot.

15-2C Electromagnetic waves are characterized by their frequency ν and wavelength λ . These two properties in a medium are related by $\lambda = c / \nu$ where c is the speed of light in that medium.

15-3C Visible light is a kind of electromagnetic wave whose wavelength is between 0.40 and 0.76 μm . It differs from the other forms of electromagnetic radiation in that it triggers the sensation of seeing in the human eye.

15-4C Infrared radiation lies between 0.76 and 100 μm whereas ultraviolet radiation lies between the wavelengths 0.01 and 0.40 μm . The human body does not emit any radiation in the ultraviolet region since bodies at room temperature emit radiation in the infrared region only.

15-5C Thermal radiation is the radiation emitted as a result of vibrational and rotational motions of molecules, atoms and electrons of a substance, and it extends from about 0.1 to 100 μm in wavelength. Unlike the other forms of electromagnetic radiation, thermal radiation is emitted by bodies because of their temperature.

15-6C Light (or visible) radiation consists of narrow bands of colors from violet to red. The color of a surface depends on its ability to reflect certain wavelength. For example, a surface that reflects radiation in the wavelength range 0.63-0.76 μm while absorbing the rest appears red to the eye. A surface that reflects all the light appears white while a surface that absorbs the entire light incident on it appears black. The color of a surface at room temperature is not related to the radiation it emits.

15-7C Radiation in opaque solids is considered surface phenomena since only radiation emitted by the molecules in a very thin layer of a body at the surface can escape the solid.

15-8C Because the snow reflects almost all of the visible and ultraviolet radiation, and the skin is exposed to radiation both from the sun and from the snow.

15-9C Microwaves in the range of 10^2 to 10^5 μm are very suitable for use in cooking as they are reflected by metals, transmitted by glass and plastics and absorbed by food (especially water) molecules. Thus the electric energy converted to radiation in a microwave oven eventually becomes part of the internal energy of the food with no conduction and convection thermal resistances involved. In conventional cooking, on the other hand, conduction and convection thermal resistances slow down the heat transfer, and thus the heating process.

15-10 The speeds of light in air, water, and glass are to be determined.

Analysis The speeds of light in air, water and glass are

$$\text{Air: } c = \frac{c_0}{n} = \frac{3.0 \times 10^8 \text{ m/s}}{1} = \mathbf{3.0 \times 10^8 \text{ m/s}}$$

$$\text{Water: } c = \frac{c_0}{n} = \frac{3.0 \times 10^8 \text{ m/s}}{1.33} = \mathbf{2.26 \times 10^8 \text{ m/s}}$$

$$\text{Glass: } c = \frac{c_0}{n} = \frac{3.0 \times 10^8 \text{ m/s}}{1.5} = \mathbf{2.0 \times 10^8 \text{ m/s}}$$

15-11 Electricity is generated and transmitted in power lines at a frequency of 60 Hz. The wavelength of the electromagnetic waves is to be determined.

Analysis The wavelength of the electromagnetic waves is

$$\lambda = \frac{c}{\nu} = \frac{2.998 \times 10^8 \text{ m/s}}{60 \text{ Hz}(1/\text{s})} = \mathbf{4.997 \times 10^6 \text{ m}}$$

Power lines



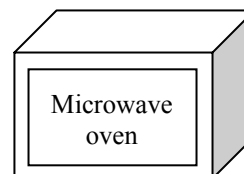
15-12 A microwave oven operates at a frequency of 2.2×10^9 Hz. The wavelength of these microwaves and the energy of each microwave are to be determined.

Analysis The wavelength of these microwaves is

$$\lambda = \frac{c}{\nu} = \frac{2.998 \times 10^8 \text{ m/s}}{2.2 \times 10^9 \text{ Hz}(1/\text{s})} = 0.136 \text{ m} = \mathbf{136 \text{ mm}}$$

Then the energy of each microwave becomes

$$e = h\nu = \frac{hc}{\lambda} = \frac{(6.625 \times 10^{-34} \text{ Js})(2.998 \times 10^8 \text{ m/s})}{0.136 \text{ m}} = \mathbf{1.46 \times 10^{-24} \text{ J}}$$



15-13 A radio station is broadcasting radiowaves at a wavelength of 200 m. The frequency of these waves is to be determined.

Analysis The frequency of the waves is determined from

$$\lambda = \frac{c}{\nu} \longrightarrow \nu = \frac{c}{\lambda} = \frac{2.998 \times 10^8 \text{ m/s}}{200 \text{ m}} = \mathbf{1.5 \times 10^6 \text{ Hz}}$$



15-14 A cordless telephone operates at a frequency of 8.5×10^8 Hz. The wavelength of these telephone waves is to be determined.

Analysis The wavelength of the telephone waves is

$$\lambda = \frac{c}{\nu} = \frac{2.998 \times 10^8 \text{ m/s}}{8.5 \times 10^8 \text{ Hz(1/s)}} = 0.353 \text{ m} = \mathbf{353 \text{ mm}}$$



Blackbody Radiation

15-15C A blackbody is a perfect emitter and absorber of radiation. A blackbody does not actually exist. It is an idealized body that emits the maximum amount of radiation that can be emitted by a surface at a given temperature.

15-16C *Spectral blackbody emissive power* is the amount of radiation energy emitted by a blackbody at an absolute temperature T per unit time, per unit surface area and per unit wavelength about wavelength λ . The integration of the spectral blackbody emissive power over the entire wavelength spectrum gives the *total blackbody emissive power*,

$$E_b(T) = \int_0^{\infty} E_{b\lambda}(T) d\lambda = \sigma T^4$$

The spectral blackbody emissive power varies with wavelength, the total blackbody emissive power does not.

15-17C We defined the blackbody radiation function f_λ because the integration $\int_0^{\infty} E_{b\lambda}(T) d\lambda$ cannot be performed. The blackbody radiation function f_λ represents the fraction of radiation emitted from a blackbody at temperature T in the wavelength range from $\lambda = 0$ to λ . This function is used to determine the fraction of radiation in a wavelength range between λ_1 and λ_2 .

15-18C The larger the temperature of a body, the larger the fraction of the radiation emitted in shorter wavelengths. Therefore, the body at 1500 K will emit more radiation in the shorter wavelength region. The body at 1000 K emits more radiation at $20 \mu\text{m}$ than the body at 1500 K since $\lambda T = \text{constant}$.

15-19 The maximum thermal radiation that can be emitted by a surface is to be determined.

Analysis The maximum thermal radiation that can be emitted by a surface is determined from Stefan-Boltzman law to be

$$E_b(T) = \sigma T^4 = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(800 \text{ K})^4 = \mathbf{2.32 \times 10^4 \text{ W/m}^2}$$

15-20 An isothermal cubical body is suspended in the air. The rate at which the cube emits radiation energy and the spectral blackbody emissive power are to be determined.

Assumptions The body behaves as a black body.

Analysis (a) The total blackbody emissive power is determined from Stefan-Boltzman Law to be

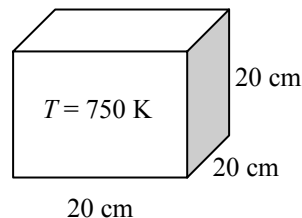
$$A_s = 6a^2 = 6(0.2^2) = 0.24 \text{ m}^2$$

$$E_b(T) = \sigma T^4 A_s = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(750 \text{ K})^4 (0.24 \text{ m}^2) = \mathbf{4306 \text{ W}}$$

(b) The spectral blackbody emissive power at a wavelength of $4 \mu\text{m}$ is determined from Planck's distribution law,

$$E_{b\lambda} = \frac{C_1}{\lambda^5 \left[\exp\left(\frac{C_2}{\lambda T}\right) - 1 \right]} = \frac{3.74177 \times 10^8 \text{ W} \cdot \mu\text{m}^4/\text{m}^2}{(4 \mu\text{m})^5 \left[\exp\left(\frac{1.43878 \times 10^4 \mu\text{m} \cdot \text{K}}{(4 \mu\text{m})(750 \text{ K})}\right) - 1 \right]}$$

$$= 3045 \text{ W/m}^2 \cdot \mu\text{m} = \mathbf{3.05 \text{ kW/m}^2 \cdot \mu\text{m}}$$



15-21E The sun is at an effective surface temperature of 10,400 R. The rate of infrared radiation energy emitted by the sun is to be determined.

Assumptions The sun behaves as a black body.

Analysis Noting that $T = 10,400 \text{ R} = 5778 \text{ K}$, the blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined from Table 15-2 to be

$$\lambda_1 T = (0.76 \mu\text{m})(5778 \text{ K}) = 4391.3 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.547370$$

$$\lambda_2 T = (100 \mu\text{m})(5778 \text{ K}) = 577,800 \mu\text{mK} \longrightarrow f_{\lambda_2} = 1.0$$

Then the fraction of radiation emitted between these two wavelengths becomes

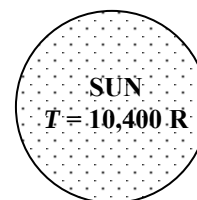
$$f_{\lambda_2} - f_{\lambda_1} = 1.0 - 0.547 = 0.453 \quad (\text{or } 45.3\%)$$

The total blackbody emissive power of the sun is determined from Stefan-Boltzman Law to be

$$E_b = \sigma T^4 = (0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)(10,400 \text{ R})^4 = 2.005 \times 10^7 \text{ Btu/h} \cdot \text{ft}^2$$

Then,

$$E_{\text{infrared}} = (0.453)E_b = (0.453)(2.005 \times 10^7 \text{ Btu/h} \cdot \text{ft}^2) = \mathbf{9.08 \times 10^6 \text{ Btu/h} \cdot \text{ft}^2}$$



15-22 EES The spectral blackbody emissive power of the sun versus wavelength in the range of 0.01 μm to 1000 μm is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

T=5780 [K]

lambda=0.01[micrometer]

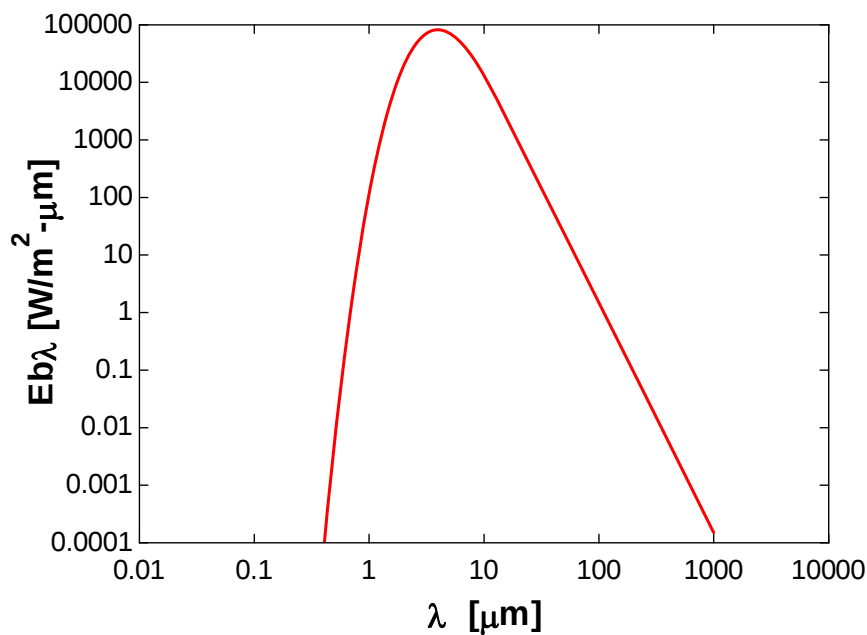
"ANALYSIS"

$E_{b,\lambda} = C_1 / (\lambda^5 (\exp(C_2 / (\lambda T)) - 1))$

C_1=3.742E8 [W-micrometer⁴/m²]

C_2=1.439E4 [micrometer-K]

λ [μm]	$E_{b,\lambda}$ [W/m ² - μm]
0.01	0
10.11	12684
20.21	846.3
30.31	170.8
40.41	54.63
50.51	22.52
60.62	10.91
70.72	5.905
80.82	3.469
90.92	2.17
...	...
...	...
909.1	0.0002198
919.2	0.0002103
929.3	0.0002013
939.4	0.0001928
949.5	0.0001847
959.6	0.000177
969.7	0.0001698
979.8	0.0001629
989.9	0.0001563
1000	0.0001501



15-23 The temperature of the filament of an incandescent light bulb is given. The fraction of visible radiation emitted by the filament and the wavelength at which the emission peaks are to be determined.

Assumptions The filament behaves as a black body.

Analysis The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.40 \mu\text{m}$ to $\lambda_2 = 0.76 \mu\text{m}$. Noting that $T = 3200 \text{ K}$, the blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined from Table 15-2 to be

$$\lambda_1 T = (0.40 \mu\text{m})(3200 \text{ K}) = 1280 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0043964$$

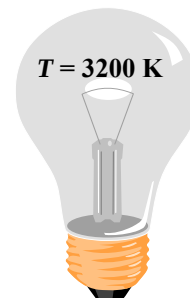
$$\lambda_2 T = (0.76 \mu\text{m})(3200 \text{ K}) = 2432 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.147114$$

Then the fraction of radiation emitted between these two wavelengths becomes

$$f_{\lambda_2} - f_{\lambda_1} = 0.147114 - 0.004396 = \mathbf{0.142718} \quad (\text{or } 14.3\%)$$

The wavelength at which the emission of radiation from the filament is maximum is

$$(\lambda T)_{\text{max power}} = 2897.8 \mu\text{m} \cdot \text{K} \longrightarrow \lambda_{\text{max power}} = \frac{2897.8 \mu\text{m} \cdot \text{K}}{3200 \text{ K}} = \mathbf{0.906 \text{ mm}}$$



15-24 EES Prob. 15-23 is reconsidered. The effect of temperature on the fraction of radiation emitted in the visible range is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$$T = 3200 \text{ [K]}$$

$$\lambda_1 = 0.40 \text{ [micrometer]}$$

$$\lambda_2 = 0.76 \text{ [micrometer]}$$

"ANALYSIS"

$$E_{b,\lambda} = C_1 / (\lambda^5 (\exp(C_2 / (\lambda T)) - 1))$$

$$C_1 = 3.742 \times 10^8 \text{ [W-micrometer}^4\text{/m}^2\text{]}$$

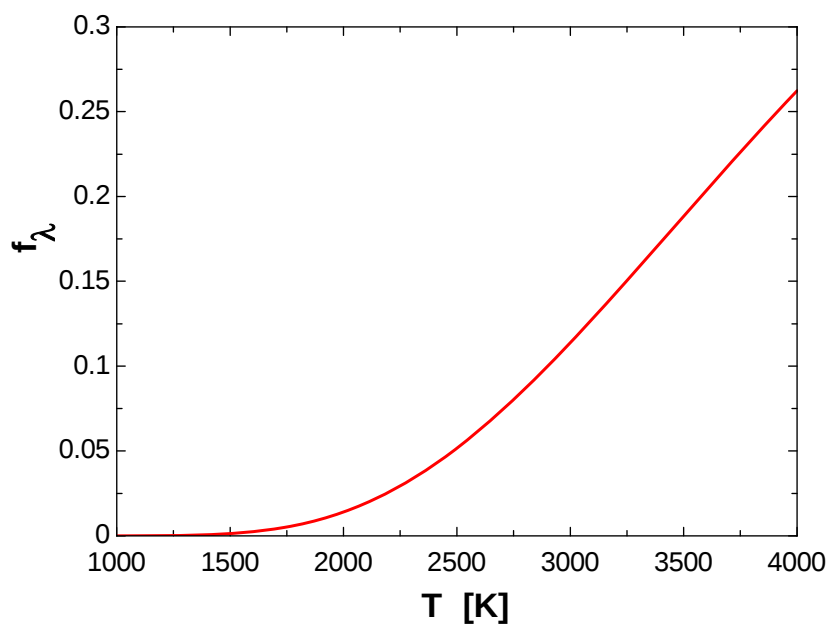
$$C_2 = 1.439 \times 10^4 \text{ [micrometer-K]}$$

$$f_{\lambda} = \text{integral}(E_{b,\lambda}, \lambda, \lambda_1, \lambda_2) / E_b$$

$$E_b = \sigma T^4$$

$$\sigma = 5.67 \times 10^{-8} \text{ [W/m}^2\text{-K}^4\text{]} \quad \text{"Stefan-Boltzmann constant"}$$

T [K]	f_{λ}
1000	0.000007353
1200	0.0001032
1400	0.0006403
1600	0.002405
1800	0.006505
2000	0.01404
2200	0.02576
2400	0.04198
2600	0.06248
2800	0.08671
3000	0.1139
3200	0.143
3400	0.1732
3600	0.2036
3800	0.2336
4000	0.2623



15-25 An incandescent light bulb emits 15% of its energy at wavelengths shorter than $0.8\ \mu\text{m}$. The temperature of the filament is to be determined.

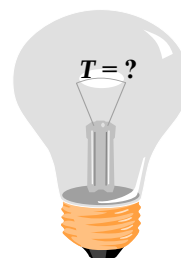
Assumptions The filament behaves as a black body.

Analysis From the Table 15-2 for the fraction of the radiation, we read

$$f_{\lambda} = 0.15 \longrightarrow \lambda T = 2445\ \mu\text{mK}$$

For the wavelength range of $\lambda_1 = 0.0\ \mu\text{m}$ to $\lambda_2 = 0.8\ \mu\text{m}$

$$\lambda = 0.8\ \mu\text{m} \longrightarrow \lambda T = 2445\ \mu\text{mK} \longrightarrow T = \mathbf{3056\ K}$$



15-26 Radiation emitted by a light source is maximum in the blue range. The temperature of this light source and the fraction of radiation it emits in the visible range are to be determined.

Assumptions The light source behaves as a black body.

Analysis The temperature of this light source is

$$(\lambda T)_{\text{max power}} = 2897.8\ \mu\text{m} \cdot \text{K} \longrightarrow T = \frac{2897.8\ \mu\text{m} \cdot \text{K}}{0.47\ \mu\text{m}} = \mathbf{6166\ K}$$

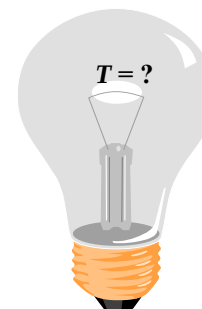
The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.40\ \mu\text{m}$ to $\lambda_2 = 0.76\ \mu\text{m}$. Noting that $T = 6166\ \text{K}$, the blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined from Table 15-2 to be

$$\lambda_1 T = (0.40\ \mu\text{m})(6166\ \text{K}) = 2466\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.15440$$

$$\lambda_2 T = (0.76\ \mu\text{m})(6166\ \text{K}) = 4686\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.59144$$

Then the fraction of radiation emitted between these two wavelengths becomes

$$f_{\lambda_2} - f_{\lambda_1} = 0.59144 - 0.15440 \cong \mathbf{0.437} \quad (\text{or } 43.7\%)$$



15-27 A glass window transmits 90% of the radiation in a specified wavelength range and is opaque for radiation at other wavelengths. The rate of radiation transmitted through this window is to be determined for two cases.

Assumptions The sources behave as a black body.

Analysis The surface area of the glass window is

$$A_s = 4 \text{ m}^2$$

(a) For a blackbody source at 5800 K, the total blackbody radiation emission is

$$E_b(T) = \sigma T^4 A_s = (5.67 \times 10^{-8} \text{ kW/m}^2 \cdot \text{K}^4)(5800 \text{ K})^4 (4 \text{ m}^2) = 2.567 \times 10^5 \text{ kW}$$

The fraction of radiation in the range of 0.3 to 3.0 μm is

$$\lambda_1 T = (0.30 \mu\text{m})(5800 \text{ K}) = 1740 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.03345$$

$$\lambda_2 T = (3.0 \mu\text{m})(5800 \text{ K}) = 17,400 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.97875$$

$$\Delta f = f_{\lambda_2} - f_{\lambda_1} = 0.97875 - 0.03345 = 0.9453$$

Noting that 90% of the total radiation is transmitted through the window,

$$\begin{aligned} E_{\text{transmit}} &= 0.90 \Delta f E_b(T) \\ &= (0.90)(0.9453)(2.567 \times 10^5 \text{ kW}) = \mathbf{2.184 \times 10^5 \text{ kW}} \end{aligned}$$

(b) For a blackbody source at 1000 K, the total blackbody emissive power is

$$E_b(T) = \sigma T^4 A_s = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1000 \text{ K})^4 (4 \text{ m}^2) = 226.8 \text{ kW}$$

The fraction of radiation in the visible range of 0.3 to 3.0 μm is

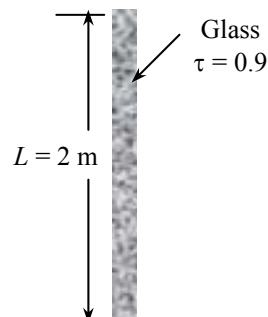
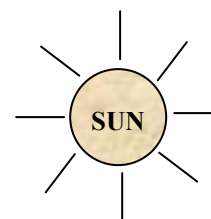
$$\lambda_1 T = (0.30 \mu\text{m})(1000 \text{ K}) = 300 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0000$$

$$\lambda_2 T = (3.0 \mu\text{m})(1000 \text{ K}) = 3000 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.273232$$

$$\Delta f = f_{\lambda_2} - f_{\lambda_1} = 0.273232 - 0$$

and

$$E_{\text{transmit}} = 0.90 \Delta f E_b(T) = (0.90)(0.273232)(226.8 \text{ kW}) = \mathbf{55.8 \text{ kW}}$$



Radiation Properties

15-28C The emissivity ε is the ratio of the radiation emitted by the surface to the radiation emitted by a blackbody at the same temperature. The fraction of radiation absorbed by the surface is called the absorptivity α ,

$$\varepsilon(T) = \frac{E(T)}{E_b(T)} \quad \text{and} \quad \alpha = \frac{\text{absorbed radiation}}{\text{incident radiation}} = \frac{G_{abs}}{G}$$

When the surface temperature is equal to the temperature of the source of radiation, the total hemispherical emissivity of a surface at temperature T is equal to its total hemispherical absorptivity for radiation coming from a blackbody at the same temperature $\varepsilon_\lambda(T) = \alpha_\lambda(T)$.

15-29C The fraction of irradiation reflected by the surface is called reflectivity ρ and the fraction transmitted is called the transmissivity τ

$$\rho = \frac{G_{ref}}{G} \quad \text{and} \quad \tau = \frac{G_{tr}}{G}$$

Surfaces are assumed to reflect in a perfectly spectral or diffuse manner for simplicity. In spectral (or mirror like) reflection, the angle of reflection equals the angle of incidence of the radiation beam. In diffuse reflection, radiation is reflected equally in all directions.

15-30C A body whose surface properties are independent of wavelength is said to be a graybody. The emissivity of a blackbody is one for all wavelengths, the emissivity of a graybody is between zero and one.

15-31C The heating effect which is due to the non-gray characteristic of glass, clear plastic, or atmospheric gases is known as the greenhouse effect since this effect is utilized primarily in greenhouses. The combustion gases such as CO_2 and water vapor in the atmosphere transmit the bulk of the solar radiation but absorb the infrared radiation emitted by the surface of the earth, acting like a heat trap. There is a concern that the energy trapped on earth will eventually cause global warming and thus drastic changes in weather patterns.

15-32C Glass has a transparent window in the wavelength range 0.3 to $3 \mu\text{m}$ and it is not transparent to the radiation which has wavelength range greater than $3 \mu\text{m}$. Therefore, because the microwaves are in the range of 10^2 to $10^5 \mu\text{m}$, the harmful microwave radiation cannot escape from the glass door.

15-33 The variation of emissivity of a surface at a specified temperature with wavelength is given. The average emissivity of the surface and its emissive power are to be determined.

Analysis The average emissivity of the surface can be determined from

$$\begin{aligned}\varepsilon(T) &= \frac{\varepsilon_1 \int_0^{\lambda_1} E_{b_\lambda}(T) d\lambda}{\sigma T^4} + \frac{\varepsilon_2 \int_{\lambda_1}^{\lambda_2} E_{b_\lambda}(T) d\lambda}{\sigma T^4} + \frac{\varepsilon_3 \int_{\lambda_2}^{\infty} E_{b_\lambda}(T) d\lambda}{\sigma T^4} \\ &= \varepsilon_1 f_{0-\lambda_1} + \varepsilon_2 f_{\lambda_1-\lambda_2} + \varepsilon_3 f_{\lambda_2-\infty} \\ &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (f_{\lambda_2} - f_{\lambda_1}) + \varepsilon_3 (1 - f_{\lambda_2})\end{aligned}$$

where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$, determined from

$$\lambda_1 T = (2 \mu\text{m})(1000 \text{ K}) = 2000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.066728$$

$$\lambda_2 T = (6 \mu\text{m})(1000 \text{ K}) = 6000 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.737818$$

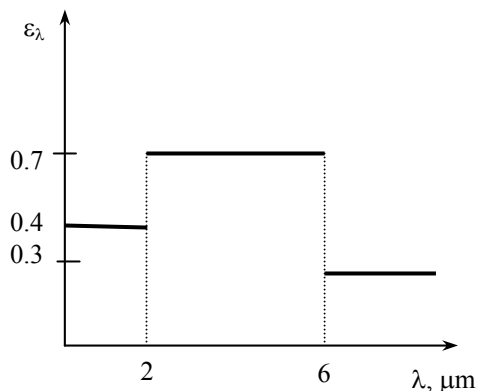
$$f_{0-\lambda_1} = f_{\lambda_1} - f_0 = f_{\lambda_1} \text{ since } f_0 = 0 \text{ and } f_{\lambda_2-\infty} = f_\infty - f_{\lambda_2} \text{ since } f_\infty = 1.$$

and,

$$\varepsilon = (0.4)0.066728 + (0.7)(0.737818 - 0.066728) + (0.3)(1 - 0.737818) = \mathbf{0.575}$$

Then the emissive power of the surface becomes

$$E = \varepsilon \sigma T^4 = 0.575(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1000 \text{ K})^4 = \mathbf{32.6 \text{ kW/m}^2}$$



15-34 The variation of reflectivity of a surface with wavelength is given. The average reflectivity, emissivity, and absorptivity of the surface are to be determined for two source temperatures.

Analysis The average reflectivity of this surface for solar radiation ($T = 5800$ K) is determined to be

$$\lambda T = (3 \mu\text{m})(5800 \text{ K}) = 17400 \mu\text{mK} \rightarrow f_{\lambda} = 0.978746$$

$$\begin{aligned} \rho(T) &= \rho_1 f_{0-\lambda_1}(T) + \rho_2 f_{\lambda_1-\infty}(T) \\ &= \rho_1 f_{\lambda_1} + \rho_2 (1 - f_{\lambda_1}) \\ &= (0.35)(0.978746) + (0.95)(1 - 0.978746) \\ &= \mathbf{0.362} \end{aligned}$$

Noting that this is an opaque surface, $\tau = 0$

$$\text{At } T = 5800 \text{ K: } \alpha + \rho = 1 \rightarrow \alpha = 1 - \rho = 1 - 0.362 = \mathbf{0.638}$$

Repeating calculations for radiation coming from surfaces at $T = 300$ K,

$$\lambda T = (3 \mu\text{m})(300 \text{ K}) = 900 \mu\text{mK} \rightarrow f_{\lambda_1} = 0.0001685$$

$$\rho(T) = (0.35)(0.0001685) + (0.95)(1 - 0.0001685) = \mathbf{0.95}$$

$$\text{At } T = 300 \text{ K: } \alpha + \rho = 1 \rightarrow \alpha = 1 - \rho = 1 - 0.95 = \mathbf{0.05}$$

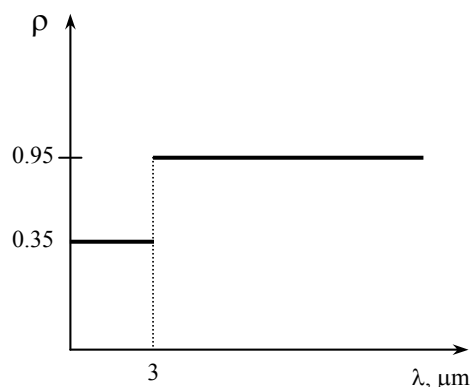
and $\varepsilon = \alpha = \mathbf{0.05}$

The temperature of the aluminum plate is close to room temperature, and thus emissivity of the plate will be equal to its absorptivity at room temperature. That is,

$$\varepsilon = \varepsilon_{\text{room}} = 0.05$$

$$\alpha = \alpha_s = 0.638$$

which makes it suitable as a solar collector. ($\alpha_s = 1$ and $\varepsilon_{\text{room}} = 0$ for an ideal solar collector)



15-35 The variation of transmissivity of the glass window of a furnace at a specified temperature with wavelength is given. The fraction and the rate of radiation coming from the furnace and transmitted through the window are to be determined.

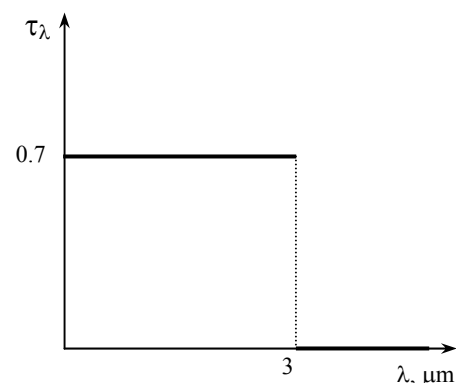
Assumptions The window glass behaves as a black body.

Analysis The fraction of radiation at wavelengths smaller than $3 \mu\text{m}$ is

$$\lambda T = (3 \mu\text{m})(1200 \text{ K}) = 3600 \mu\text{mK} \rightarrow f_{\lambda} = 0.403607$$

The fraction of radiation coming from the furnace and transmitted through the window is

$$\begin{aligned} \tau(T) &= \tau_1 f_{\lambda} + \tau_2 (1 - f_{\lambda}) \\ &= (0.7)(0.403607) + (0)(1 - 0.403607) \\ &= \mathbf{0.2825} \end{aligned}$$



Then the rate of radiation coming from the furnace and transmitted through the window becomes

$$G_{tr} = \tau A \sigma T^4 = 0.2825(0.40 \times 0.40 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1200 \text{ K})^4 = \mathbf{5315 \text{ W}}$$

15-36 The variation of emissivity of a tungsten filament with wavelength is given. The average emissivity, absorptivity, and reflectivity of the filament are to be determined for two temperatures.

Analysis (a) $T = 2000 \text{ K}$

$$\lambda_1 T = (1 \mu\text{m})(2000 \text{ K}) = 2000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.066728$$

The average emissivity of this surface is

$$\begin{aligned}\varepsilon(T) &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) \\ &= (0.5)(0.066728) + (0.15)(1 - 0.066728) \\ &= \mathbf{0.173}\end{aligned}$$

From Kirchhoff's law,

$$\varepsilon = \alpha = \mathbf{0.173} \quad (\text{at } 2000 \text{ K})$$

and $\alpha + \rho = 1 \longrightarrow \rho = 1 - \alpha = 1 - 0.173 = \mathbf{0.827}$

(b) $T = 3000 \text{ K}$

$$\lambda_1 T = (1 \mu\text{m})(3000 \text{ K}) = 3000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.273232$$

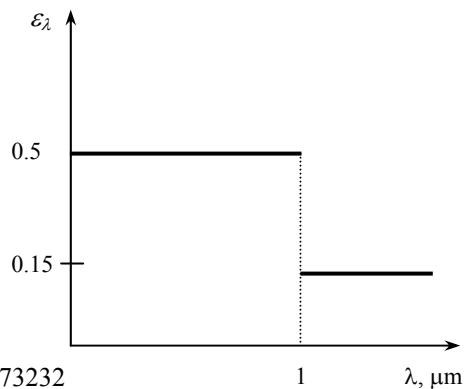
Then

$$\varepsilon(T) = \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) = (0.5)(0.273232) + (0.15)(1 - 0.273232) = \mathbf{0.246}$$

From Kirchhoff's law,

$$\varepsilon = \alpha = \mathbf{0.246} \quad (\text{at } 3000 \text{ K})$$

and $\alpha + \rho = 1 \longrightarrow \rho = 1 - \alpha = 1 - 0.246 = \mathbf{0.754}$



15-37 The variations of emissivity of two surfaces are given. The average emissivity, absorptivity, and reflectivity of each surface are to be determined at the given temperature.

Analysis For the first surface:

$$\lambda_1 T = (3 \mu\text{m})(3000 \text{ K}) = 9000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.890029$$

The average emissivity of this surface is

$$\begin{aligned}\varepsilon(T) &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) \\ &= (0.2)(0.890029) + (0.9)(1 - 0.890029) \\ &= \mathbf{0.28}\end{aligned}$$

The absorptivity and reflectivity are determined from Kirchhoff's law

$$\begin{aligned}\varepsilon &= \alpha = \mathbf{0.28} \quad (\text{at } 3000 \text{ K}) \\ \alpha + \rho &= 1 \longrightarrow \rho = 1 - \alpha = 1 - 0.28 = \mathbf{0.72}\end{aligned}$$

For the second surface:

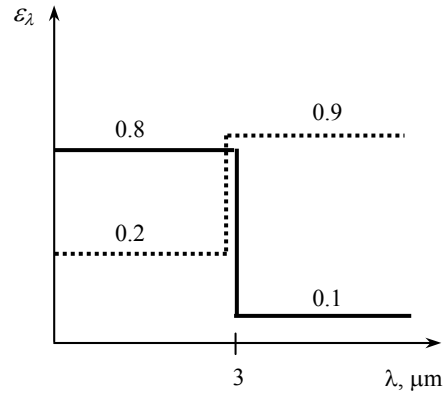
$$\lambda_1 T = (3 \mu\text{m})(3000 \text{ K}) = 9000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.890029$$

The average emissivity of this surface is

$$\begin{aligned}\varepsilon(T) &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) \\ &= (0.8)(0.890029) + (0.1)(1 - 0.890029) \\ &= \mathbf{0.72}\end{aligned}$$

Then,

$$\begin{aligned}\varepsilon &= \alpha = \mathbf{0.72} \quad (\text{at } 3000 \text{ K}) \\ \alpha + \rho &= 1 \rightarrow \rho = 1 - \alpha = 1 - 0.72 = \mathbf{0.28}\end{aligned}$$



Discussion The second surface is more suitable to serve as a solar absorber since its absorptivity for short wavelength radiation (typical of radiation emitted by a high-temperature source such as the sun) is high, and its emissivity for long wavelength radiation (typical of emitted radiation from the absorber plate) is low.

15-38 The variation of emissivity of a surface with wavelength is given. The average emissivity and absorptivity of the surface are to be determined for two temperatures.

Analysis (a) For $T = 5800 \text{ K}$:

$$\lambda_1 T = (5 \mu\text{m})(5800 \text{ K}) = 29,000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.994715$$

The average emissivity of this surface is

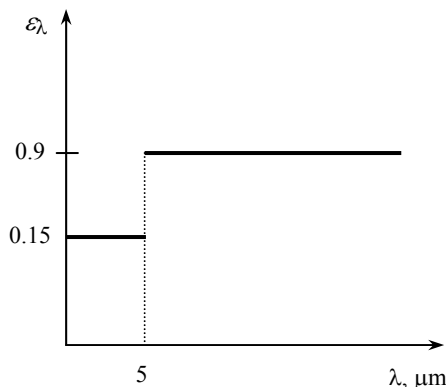
$$\begin{aligned}\varepsilon(T) &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) \\ &= (0.15)(0.994715) + (0.9)(1 - 0.994715) \\ &= \mathbf{0.154}\end{aligned}$$

(b) For $T = 300 \text{ K}$:

$$\lambda_1 T = (5 \mu\text{m})(300 \text{ K}) = 1500 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.013754$$

and

$$\begin{aligned}\varepsilon(T) &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) \\ &= (0.15)(0.013754) + (0.9)(1 - 0.013754) \\ &= \mathbf{0.89}\end{aligned}$$



The absorptivities of this surface for radiation coming from sources at 5800 K and 300 K are, from Kirchhoff's law,

$$\alpha = \varepsilon = \mathbf{0.154} \quad (\text{at } 5800 \text{ K})$$

$$\alpha = \varepsilon = \mathbf{0.89} \quad (\text{at } 300 \text{ K})$$

15-39 The variation of absorptivity of a surface with wavelength is given. The average absorptivity, reflectivity, and emissivity of the surface are to be determined at given temperatures.

Analysis For $T = 2500 \text{ K}$:

$$\lambda_1 T = (2 \mu\text{m})(2500 \text{ K}) = 5000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.633747$$

The average absorptivity of this surface is

$$\begin{aligned}\alpha(T) &= \alpha_1 f_{\lambda_1} + \alpha_2 (1 - f_{\lambda_1}) \\ &= (0.2)(0.633747) + (0.7)(1 - 0.633747) \\ &= \mathbf{0.38}\end{aligned}$$

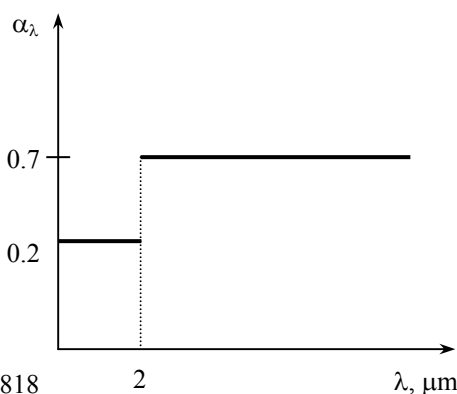
Then the reflectivity of this surface becomes

$$\alpha + \rho = 1 \longrightarrow \rho = 1 - \alpha = 1 - 0.38 = \mathbf{0.62}$$

Using Kirchhoff's law, $\alpha = \varepsilon$, the average emissivity of this surface at $T = 3000 \text{ K}$ is determined to be

$$\lambda_1 T = (2 \mu\text{m})(3000 \text{ K}) = 6000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.737818$$

$$\begin{aligned}\varepsilon(T) &= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (1 - f_{\lambda_1}) \\ &= (0.2)(0.737818) + (0.7)(1 - 0.737818) \\ &= \mathbf{0.33}\end{aligned}$$



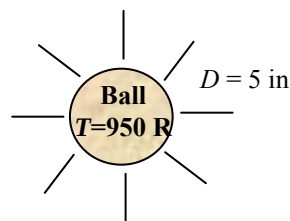
15-40E A spherical ball emits radiation at a certain rate. The average emissivity of the ball is to be determined at the given temperature.

Analysis The surface area of the ball is

$$A = \pi D^2 = \pi(5/12 \text{ ft})^2 = 0.5454 \text{ ft}^2$$

Then the average emissivity of the ball at this temperature is determined to be

$$E = \varepsilon A \sigma T^4 \longrightarrow \varepsilon = \frac{E}{A \sigma T^4} = \frac{550 \text{ Btu/h}}{(0.5454 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)(950 \text{ R})^4} = \mathbf{0.722}$$



15-41 The variation of transmissivity of a glass is given. The average transmissivity of the pane at two temperatures and the amount of solar radiation transmitted through the pane are to be determined.

Analysis For $T=5800 \text{ K}$:

$$\lambda_1 T_1 = (0.3 \mu\text{m})(5800 \text{ K}) = 1740 \mu\text{mK}$$

$$\longrightarrow f_{\lambda_1} = 0.033454$$

$$\lambda_2 T_1 = (3 \mu\text{m})(5800 \text{ K}) = 17,400 \mu\text{mK}$$

$$\longrightarrow f_{\lambda_2} = 0.978746$$

The average transmissivity of this surface is

$$\begin{aligned} \tau(T) &= \tau_1 (f_{\lambda_2} - f_{\lambda_1}) \\ &= (0.92)(0.978746 - 0.033454) = \mathbf{0.870} \end{aligned}$$

For $T=300 \text{ K}$:

$$\lambda_1 T_2 = (0.3 \mu\text{m})(300 \text{ K}) = 90 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0$$

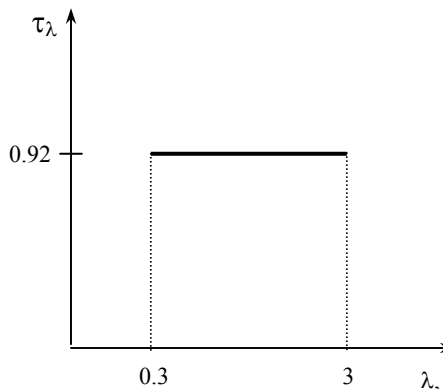
$$\lambda_2 T_2 = (3 \mu\text{m})(300 \text{ K}) = 900 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.0001685$$

Then,

$$\tau(T) = \tau_1 (f_{\lambda_2} - f_{\lambda_1}) = (0.92)(0.0001685 - 0.0) = \mathbf{0.00016} \approx \mathbf{0}$$

The amount of solar radiation transmitted through this glass is

$$G_{\text{tr}} = \tau G_{\text{incident}} = 0.870(650 \text{ W/m}^2) = \mathbf{566 \text{ W/m}^2}$$



View Factors

15-42C The view factor $F_{i \rightarrow j}$ represents the fraction of the radiation leaving surface i that strikes surface j directly. The view factor from a surface to itself is non-zero for concave surfaces.

15-43C The pair of view factors $F_{i \rightarrow j}$ and $F_{j \rightarrow i}$ are related to each other by the reciprocity rule $A_i F_{ij} = A_j F_{ji}$ where A_i is the area of the surface i and A_j is the area of the surface j . Therefore,

$$A_1 F_{12} = A_2 F_{21} \longrightarrow F_{12} = \frac{A_2}{A_1} F_{21}$$

15-44C The summation rule for an enclosure and is expressed as $\sum_{j=1}^N F_{i \rightarrow j} = 1$ where N is the number of surfaces of the enclosure. It states that the sum of the view factors from surface i of an enclosure to all surfaces of the enclosure, including to itself must be equal to unity.

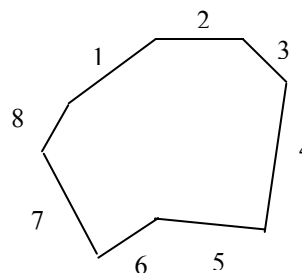
The superposition rule is stated as the view factor from a surface i to a surface j is equal to the sum of the view factors from surface i to the parts of surface j , $F_{1 \rightarrow (2,3)} = F_{1 \rightarrow 2} + F_{1 \rightarrow 3}$.

15-45C The cross-string method is applicable to geometries which are very long in one direction relative to the other directions. By attaching strings between corners the Crossed-Strings Method is expressed as

$$F_{i \rightarrow j} = \frac{\sum \text{Crossed strings} - \sum \text{Uncrossed strings}}{2 \times \text{string on surface } i}$$

15-46 An enclosure consisting of eight surfaces is considered. The number of view factors this geometry involves and the number of these view factors that can be determined by the application of the reciprocity and summation rules are to be determined.

Analysis An eight surface enclosure ($N = 8$) involves $N^2 = 8^2 = 64$ view factors and we need to determine $\frac{N(N-1)}{2} = \frac{8(8-1)}{2} = 28$ view factors directly. The remaining $64 - 28 = 36$ of the view factors can be determined by the application of the reciprocity and summation rules.

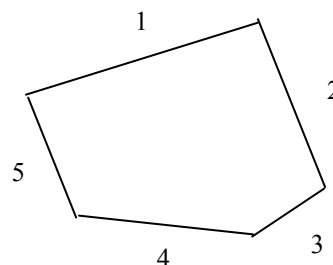


15-47 An enclosure consisting of five surfaces is considered. The number of view factors this geometry involves and the number of these view factors that can be determined by the application of the reciprocity and summation rules are to be determined.

Analysis A five surface enclosure ($N=5$) involves $N^2 = 5^2 = 25$

view factors and we need to determine $\frac{N(N-1)}{2} = \frac{5(5-1)}{2} = 10$

view factors directly. The remaining $25-10 = 15$ of the view factors can be determined by the application of the reciprocity and summation rules.



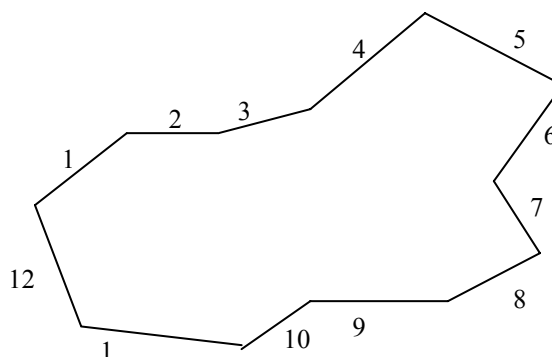
15-48 An enclosure consisting of twelve surfaces is considered. The number of view factors this geometry involves and the number of these view factors that can be determined by the application of the reciprocity and summation rules are to be determined.

Analysis A twelve surface enclosure ($N=12$)

involves $N^2 = 12^2 = 144$ view factors and we

need to determine $\frac{N(N-1)}{2} = \frac{12(12-1)}{2} = 66$

view factors directly. The remaining $144-66 = 78$ of the view factors can be determined by the application of the reciprocity and summation rules.



15-49 The view factors between the rectangular surfaces shown in the figure are to be determined.

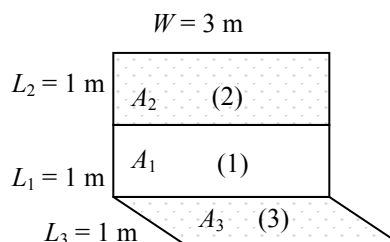
Assumptions The surfaces are diffuse emitters and reflectors.

Analysis From Fig. 15-6,

$$\left. \begin{aligned} \frac{L_3}{W} &= \frac{1}{3} = 0.33 \\ \frac{L_1}{W} &= \frac{1}{3} = 0.33 \end{aligned} \right\} F_{31} = 0.27$$

and

$$\left. \begin{aligned} \frac{L_3}{W} &= \frac{1}{3} = 0.33 \\ \frac{L_1 + L_2}{W} &= \frac{2}{3} = 0.67 \end{aligned} \right\} F_{3 \rightarrow (1+2)} = 0.32$$



We note that $A_1 = A_3$. Then the reciprocity and superposition rules gives

$$A_1 F_{13} = A_3 F_{31} \longrightarrow F_{13} = F_{31} = \mathbf{0.27}$$

$$F_{3 \rightarrow (1+2)} = F_{31} + F_{32} \longrightarrow 0.32 = 0.27 + F_{32} \longrightarrow F_{32} = 0.05$$

Finally, $A_2 = A_3 \longrightarrow F_{23} = F_{32} = \mathbf{0.05}$

15-50 A cylindrical enclosure is considered. The view factor from the side surface of this cylindrical enclosure to its base surface is to be determined.

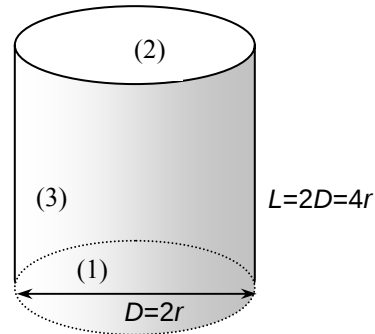
Assumptions The surfaces are diffuse emitters and reflectors.

Analysis We designate the surfaces as follows:

- Base surface by (1),
- top surface by (2), and
- side surface by (3).

Then from Fig. 15-7

$$\left. \begin{aligned} \frac{L}{r_1} &= \frac{4r_1}{r_1} = 4 \\ \frac{r_2}{L} &= \frac{r_2}{4r_2} = 0.25 \end{aligned} \right\} F_{12} = F_{21} = 0.05$$



summation rule : $F_{11} + F_{12} + F_{13} = 1$

$$0 + 0.05 + F_{13} = 1 \longrightarrow F_{13} = 0.95$$

$$\text{reciprocity rule : } A_1 F_{13} = A_3 F_{31} \longrightarrow F_{31} = \frac{A_1}{A_3} F_{13} = \frac{\pi r_1^2}{2\pi r_1 L} F_{13} = \frac{\pi r_1^2}{8\pi r_1^2} F_{13} = \frac{1}{8} (0.95) = \mathbf{0.119}$$

Discussion This problem can be solved more accurately by using the view factor relation from Table 15-3 to be

$$R_1 = \frac{r_1}{L} = \frac{r_1}{4r_1} = 0.25$$

$$R_2 = \frac{r_2}{L} = \frac{r_2}{4r_2} = 0.25$$

$$S = 1 + \frac{1 + R_2^2}{R_1^2} = 1 + \frac{1 + 0.25^2}{0.25^2} = 18$$

$$F_{12} = \frac{1}{2} \left\{ S - \left[S^2 - 4 \left(\frac{R_2}{R_1} \right)^2 \right]^{0.5} \right\} = \frac{1}{2} \left\{ 18 - \left[18^2 - 4 \left(\frac{1}{1} \right)^2 \right]^{0.5} \right\} = 0.056$$

$$F_{13} = 1 - F_{12} = 1 - 0.056 = 0.944$$

$$\text{reciprocity rule : } A_1 F_{13} = A_3 F_{31} \longrightarrow F_{31} = \frac{A_1}{A_3} F_{13} = \frac{\pi r_1^2}{2\pi r_1 L} F_{13} = \frac{\pi r_1^2}{8\pi r_1^2} F_{13} = \frac{1}{8} (0.944) = \mathbf{0.118}$$

15-51 A semispherical furnace is considered. The view factor from the dome of this furnace to its flat base is to be determined.

Assumptions The surfaces are diffuse emitters and reflectors.

Analysis We number the surfaces as follows:

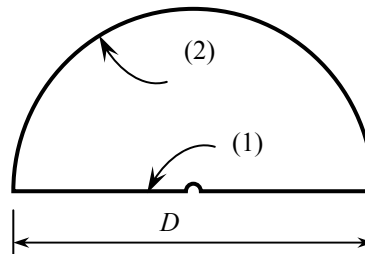
(1): circular base surface

(2): dome surface

Surface (1) is flat, and thus $F_{11} = 0$.

Summation rule: $F_{11} + F_{12} = 1 \rightarrow F_{12} = 1$

$$\text{reciprocity rule: } A_1 F_{12} = A_2 F_{21} \longrightarrow F_{21} = \frac{A_1}{A_2} F_{12} = \frac{A_1}{A_2} (1) = \frac{\frac{\pi D^2}{4}}{\frac{\pi D^2}{2}} = \frac{1}{2} = \mathbf{0.5}$$



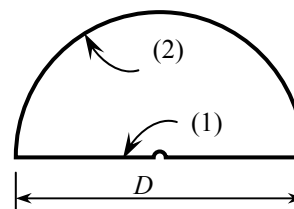
15-52 Two view factors associated with three very long ducts with different geometries are to be determined.

Assumptions 1 The surfaces are diffuse emitters and reflectors. **2** End effects are neglected.

Analysis (a) Surface (1) is flat, and thus $F_{11} = 0$.

summation rule: $F_{11} + F_{12} = 1 \rightarrow F_{12} = \mathbf{1}$

$$\text{reciprocity rule: } A_1 F_{12} = A_2 F_{21} \longrightarrow F_{21} = \frac{A_1}{A_2} F_{12} = \frac{Ds}{\left(\frac{\pi D}{2}\right)s} (1) = \frac{2}{\pi} = \mathbf{0.64}$$



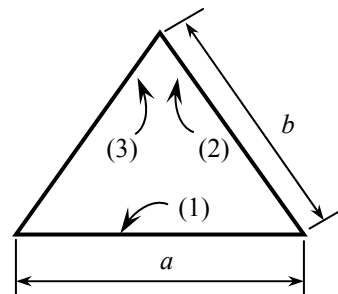
(b) Noting that surfaces 2 and 3 are symmetrical and thus $F_{12} = F_{13}$, the summation rule gives

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow 0 + F_{12} + F_{13} = 1 \longrightarrow F_{12} = \mathbf{0.5}$$

Also by using the equation obtained in Example 15-4,

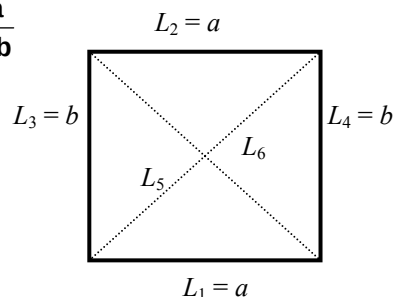
$$F_{12} = \frac{L_1 + L_2 - L_3}{2L_1} = \frac{a + b - b}{2a} = \frac{a}{2a} = \frac{1}{2} = \mathbf{0.5}$$

$$\text{reciprocity rule: } A_1 F_{12} = A_2 F_{21} \longrightarrow F_{21} = \frac{A_1}{A_2} F_{12} = \frac{a}{b} \left(\frac{1}{2}\right) = \mathbf{\frac{a}{2b}}$$



(c) Applying the crossed-string method gives

$$\begin{aligned} F_{12} = F_{21} &= \frac{(L_5 + L_6) - (L_3 + L_4)}{2L_1} \\ &= \frac{2\sqrt{a^2 + b^2} - 2b}{2a} = \frac{\sqrt{a^2 + b^2} - b}{a} \end{aligned}$$



15-53 View factors from the very long grooves shown in the figure to the surroundings are to be determined.

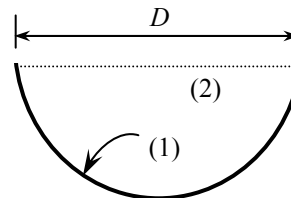
Assumptions 1 The surfaces are diffuse emitters and reflectors. 2 End effects are neglected.

Analysis (a) We designate the circular dome surface by (1) and the imaginary flat top surface by (2). Noting that (2) is flat,

$$F_{22} = 0$$

$$\text{summation rule: } F_{21} + F_{22} = 1 \longrightarrow F_{21} = 1$$

$$\text{reciprocity rule: } A_1 F_{12} = A_2 F_{21} \longrightarrow F_{12} = \frac{A_2}{A_1} F_{21} = \frac{D}{\frac{\pi D^2}{4}} (1) = \frac{4}{\pi} = \mathbf{0.64}$$



(b) We designate the two identical surfaces of length b by (1) and (3), and the imaginary flat top surface by (2). Noting that (2) is flat,

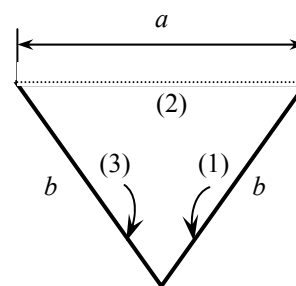
$$F_{22} = 0$$

$$\text{summation rule: } F_{21} + F_{22} + F_{23} = 1 \longrightarrow F_{21} = F_{23} = 0.5 \quad (\text{symmetry})$$

$$\text{summation rule: } F_{22} + F_{2 \rightarrow (1+3)} = 1 \longrightarrow F_{2 \rightarrow (1+3)} = 1$$

$$\text{reciprocity rule: } A_2 F_{2 \rightarrow (1+3)} = A_{(1+3)} F_{(1+3) \rightarrow 2}$$

$$\longrightarrow F_{(1+3) \rightarrow 2} = F_{(1+3) \rightarrow \text{surr}} = \frac{A_2}{A_{(1+3)}} (1) = \frac{\mathbf{a}}{\mathbf{2b}}$$

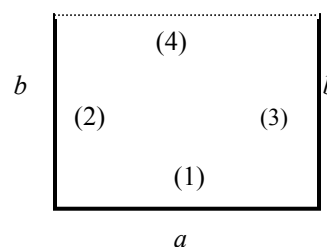


(c) We designate the bottom surface by (1), the side surfaces by (2) and (3), and the imaginary top surface by (4). Surface 4 is flat and is completely surrounded by other surfaces.

Therefore, $F_{44} = 0$ and $F_{4 \rightarrow (1+2+3)} = 1$.

$$\text{reciprocity rule: } A_4 F_{4 \rightarrow (1+2+3)} = A_{(1+2+3)} F_{(1+2+3) \rightarrow 4}$$

$$\longrightarrow F_{(1+2+3) \rightarrow 4} = F_{(1+2+3) \rightarrow \text{surr}} = \frac{A_4}{A_{(1+2+3)}} (1) = \frac{\mathbf{a}}{\mathbf{a + 2b}}$$



15-54 The view factors from the base of a cube to each of the other five surfaces are to be determined.

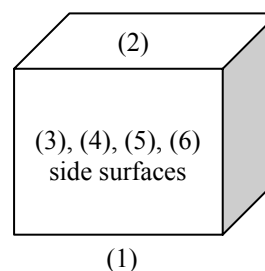
Assumptions The surfaces are diffuse emitters and reflectors.

Analysis Noting that $L_1 / D = L_2 / D = 1$, from Fig. 15-6 we read

$$F_{12} = 0.2$$

Because of symmetry, we have

$$F_{12} = F_{13} = F_{14} = F_{15} = F_{16} = \mathbf{0.2}$$



15-55 The view factor from the conical side surface to a hole located at the center of the base of a conical enclosure is to be determined.

Assumptions The conical side surface is diffuse emitter and reflector.

Analysis We number different surfaces as

the hole located at the center of the base (1)

the base of conical enclosure (2)

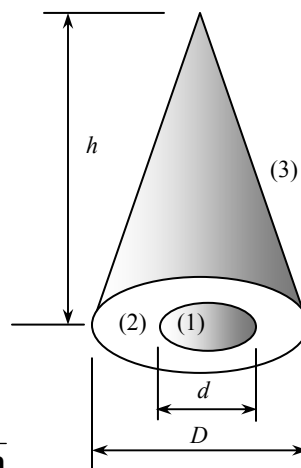
conical side surface (3)

Surfaces 1 and 2 are flat, and they have no direct view of each other. Therefore,

$$F_{11} = F_{22} = F_{12} = F_{21} = 0$$

summation rule: $F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1$

reciprocity rule: $A_1 F_{13} = A_3 F_{31} \longrightarrow \frac{\pi d^2}{4} (1) = \frac{\pi D h}{2} F_{31} \longrightarrow F_{31} = \frac{d^2}{2 D h}$



15-56 The four view factors associated with an enclosure formed by two very long concentric cylinders are to be determined.

Assumptions 1 The surfaces are diffuse emitters and reflectors. 2 End effects are neglected.

Analysis We number different surfaces as

the outer surface of the inner cylinder (1)

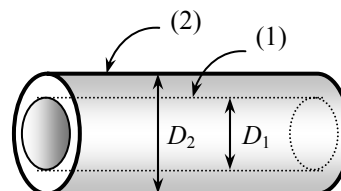
the inner surface of the outer cylinder (2)

No radiation leaving surface 1 strikes itself and thus $F_{11} = 0$

All radiation leaving surface 1 strikes surface 2 and thus $F_{12} = 1$

reciprocity rule: $A_1 F_{12} = A_2 F_{21} \longrightarrow F_{21} = \frac{A_1}{A_2} F_{12} = \frac{\pi D_1 h}{\pi D_2 h} (1) = \frac{D_1}{D_2}$

summation rule: $F_{21} + F_{22} = 1 \longrightarrow F_{22} = 1 - F_{21} = 1 - \frac{D_1}{D_2}$

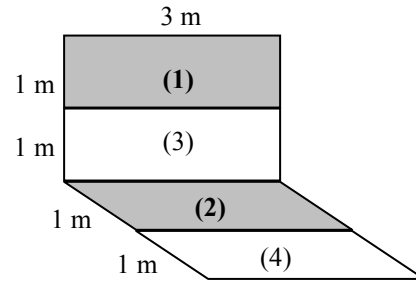


15-57 The view factors between the rectangular surfaces shown in the figure are to be determined.

Assumptions The surfaces are diffuse emitters and reflectors.

Analysis We designate the different surfaces as follows:

- shaded part of perpendicular surface by (1),
- bottom part of perpendicular surface by (3),
- shaded part of horizontal surface by (2), and
- front part of horizontal surface by (4).



(a) From Fig. 15-6

$$\left. \begin{aligned} \frac{L_2}{W} &= \frac{1}{3} \\ \frac{L_1}{W} &= \frac{1}{3} \end{aligned} \right\} F_{23} = 0.25 \quad \text{and} \quad \left. \begin{aligned} \frac{L_2}{W} &= \frac{2}{3} \\ \frac{L_1}{W} &= \frac{1}{3} \end{aligned} \right\} F_{2 \rightarrow (1+3)} = 0.32$$

$$\text{superposition rule: } F_{2 \rightarrow (1+3)} = F_{21} + F_{23} \longrightarrow F_{21} = F_{2 \rightarrow (1+3)} - F_{23} = 0.32 - 0.25 = 0.07$$

$$\text{reciprocity rule: } A_1 = A_2 \longrightarrow A_1 F_{12} = A_2 F_{21} \longrightarrow F_{12} = F_{21} = \mathbf{0.07}$$

(b) From Fig. 15-6,

$$\left. \begin{aligned} \frac{L_2}{W} &= \frac{1}{3} \\ \frac{L_1}{W} &= \frac{2}{3} \end{aligned} \right\} F_{(4+2) \rightarrow 3} = 0.15 \quad \text{and} \quad \left. \begin{aligned} \frac{L_2}{W} &= \frac{2}{3} \\ \frac{L_1}{W} &= \frac{2}{3} \end{aligned} \right\} F_{(4+2) \rightarrow (1+3)} = 0.22$$

$$\text{superposition rule: } F_{(4+2) \rightarrow (1+3)} = F_{(4+2) \rightarrow 1} + F_{(4+2) \rightarrow 3} \longrightarrow F_{(4+2) \rightarrow 1} = 0.22 - 0.15 = 0.07$$

$$\text{reciprocity rule: } A_{(4+2)} F_{(4+2) \rightarrow 1} = A_1 F_{1 \rightarrow (4+2)}$$

$$\longrightarrow F_{1 \rightarrow (4+2)} = \frac{A_{(4+2)}}{A_1} F_{(4+2) \rightarrow 1} = \frac{6}{3} (0.07) = 0.14$$

$$\text{superposition rule: } F_{1 \rightarrow (4+2)} = F_{14} + F_{12}$$

$$\longrightarrow F_{14} = 0.14 - 0.07 = \mathbf{0.07}$$

since $F_{12} = 0.07$ (from part a). Note that F_{14} in part (b) is equivalent to F_{12} in part (a).

(c) We designate

- shaded part of top surface by (1),
- remaining part of top surface by (3),
- remaining part of bottom surface by (4), and
- shaded part of bottom surface by (2).

From Fig. 15-5,

$$\left. \begin{aligned} \frac{L_2}{D} &= \frac{2}{2} \\ \frac{L_1}{D} &= \frac{2}{2} \end{aligned} \right\} F_{(2+4) \rightarrow (1+3)} = 0.20 \quad \text{and} \quad \left. \begin{aligned} \frac{L_2}{D} &= \frac{2}{2} \\ \frac{L_1}{D} &= \frac{1}{2} \end{aligned} \right\} F_{14} = 0.12$$

$$\text{superposition rule: } F_{(2+4) \rightarrow (1+3)} = F_{(2+4) \rightarrow 1} + F_{(2+4) \rightarrow 3}$$

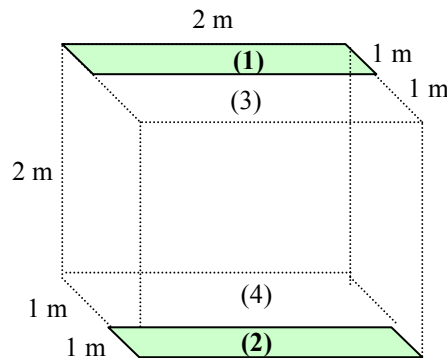
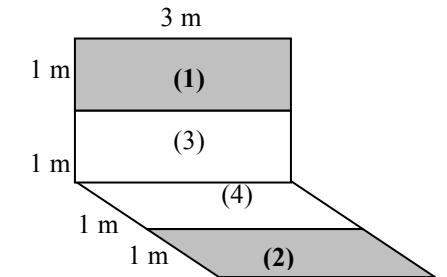
$$\text{symmetry rule: } F_{(2+4) \rightarrow 1} = F_{(2+4) \rightarrow 3}$$

Substituting symmetry rule gives

$$F_{(2+4) \rightarrow 1} = F_{(2+4) \rightarrow 3} = \frac{F_{(2+4) \rightarrow (1+3)}}{2} = \frac{0.20}{2} = 0.10$$

$$\text{reciprocity rule: } A_1 F_{1 \rightarrow (2+4)} = A_{(2+4)} F_{(2+4) \rightarrow 1} \longrightarrow (2) F_{1 \rightarrow (2+4)} = (4)(0.10) \longrightarrow F_{1 \rightarrow (2+4)} = 0.20$$

$$\text{superposition rule: } F_{1 \rightarrow (2+4)} = F_{12} + F_{14} \longrightarrow 0.20 = F_{12} + 0.12 \longrightarrow F_{12} = 0.20 - 0.12 = \mathbf{0.08}$$



15-58 The view factor between the two infinitely long parallel cylinders located a distance s apart from each other is to be determined.

Assumptions The surfaces are diffuse emitters and reflectors.

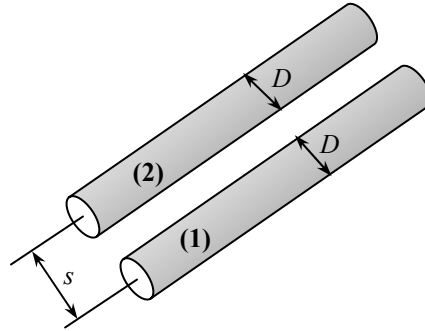
Analysis Using the crossed-strings method, the view factor between two cylinders facing each other for $s/D > 3$ is determined to be

$$F_{1-2} = \frac{\sum \text{Crossed strings} - \sum \text{Uncrossed strings}}{2 \times \text{String on surface 1}}$$

$$= \frac{2\sqrt{s^2 + D^2} - 2s}{2(\pi D / 2)}$$

or

$$F_{1-2} = \frac{2\left(\sqrt{s^2 + D^2} - s\right)}{\pi D}$$



15-59 Three infinitely long cylinders are located parallel to each other. The view factor between the cylinder in the middle and the surroundings is to be determined.

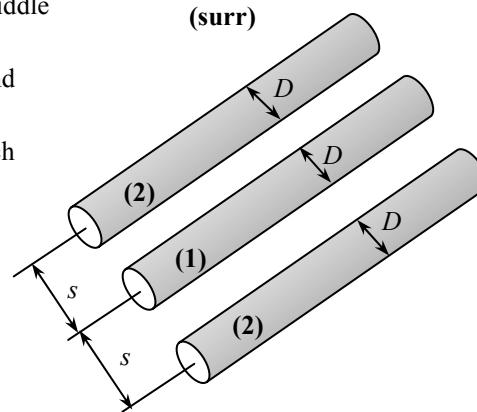
Assumptions The cylinder surfaces are diffuse emitters and reflectors.

Analysis The view factor between two cylinders facing each other is, from Prob. 15-17,

$$F_{1-2} = \frac{2\left(\sqrt{s^2 + D^2} - s\right)}{\pi D}$$

Noting that the radiation leaving cylinder 1 that does not strike the cylinder will strike the surroundings, and this is also the case for the other half of the cylinder, the view factor between the cylinder in the middle and the surroundings becomes

$$F_{1-surr} = 1 - 2F_{1-2} = 1 - \frac{4\left(\sqrt{s^2 + D^2} - s\right)}{\pi D}$$



Radiation Heat Transfer between Surfaces

15-60C The analysis of radiation exchange between black surfaces is relatively easy because of the absence of reflection. The rate of radiation heat transfer between two surfaces in this case is expressed as $\dot{Q} = A_1 F_{12} \sigma (T_1^4 - T_2^4)$ where A_1 is the surface area, F_{12} is the view factor, and T_1 and T_2 are the temperatures of two surfaces.

15-61C Radiosity is the total radiation energy leaving a surface per unit time and per unit area. Radiosity includes the emitted radiation energy as well as reflected energy. Radiosity and emitted energy are equal for blackbodies since a blackbody does not reflect any radiation.

15-62C Radiation surface resistance is given as $R_i = \frac{1 - \epsilon_i}{A_i \epsilon_i}$ and it represents the resistance of a surface to the emission of radiation. It is zero for black surfaces. The space resistance is the radiation resistance between two surfaces and is expressed as $R_{ij} = \frac{1}{A_i F_{ij}}$

15-63C The two methods used in radiation analysis are the matrix and network methods. In matrix method, equations 15-34 and 15-35 give N linear algebraic equations for the determination of the N unknown radiosities for an N-surface enclosure. Once the radiosities are available, the unknown surface temperatures and heat transfer rates can be determined from these equations respectively. This method involves the use of matrices especially when there are a large number of surfaces. Therefore this method requires some knowledge of linear algebra.

The network method involves drawing a surface resistance associated with each surface of an enclosure and connecting them with space resistances. Then the radiation problem is solved by treating it as an electrical network problem where the radiation heat transfer replaces the current and the radiosity replaces the potential. The network method is not practical for enclosures with more than three or four surfaces due to the increased complexity of the network.

15-64C Some surfaces encountered in numerous practical heat transfer applications are modeled as being adiabatic as the back sides of these surfaces are well insulated and net heat transfer through these surfaces is zero. When the convection effects on the front (heat transfer) side of such a surface is negligible and steady-state conditions are reached, the surface must lose as much radiation energy as it receives. Such a surface is called reradiating surface. In radiation analysis, the surface resistance of a reradiating surface is taken to be zero since there is no heat transfer through it.

15-65 A solid sphere is placed in an evacuated equilateral triangular enclosure. The view factor from the enclosure to the sphere and the emissivity of the enclosure are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivity of sphere is given to be $\varepsilon_1 = 0.45$.

Analysis (a) We take the sphere to be surface 1 and the surrounding enclosure to be surface 2. The view factor from surface 2 to surface 1 is determined from reciprocity relation:

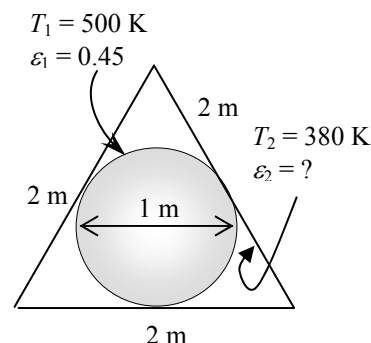
$$A_1 = \pi D^2 = \pi (1 \text{ m})^2 = 3.142 \text{ m}^2$$

$$A_2 = 3\sqrt{L^2 - D^2} \frac{L}{2} = 3\sqrt{(2 \text{ m})^2 - (1 \text{ m})^2} \frac{2 \text{ m}}{2} = 5.196 \text{ m}^2$$

$$A_1 F_{12} = A_2 F_{21}$$

$$(3.142)(1) = (5.196)F_{21}$$

$$F_{21} = \mathbf{0.605}$$



(b) The net rate of radiation heat transfer can be expressed for this two-surface enclosure to yield the emissivity of the enclosure:

$$\dot{Q} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1 - \varepsilon_1}{A_1 \varepsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1 - \varepsilon_2}{A_2 \varepsilon_2}}$$

$$3100 \text{ W} = \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(500 \text{ K})^4 - (380 \text{ K})^4]}{\frac{1 - 0.45}{(3.142 \text{ m}^2)(0.45)} + \frac{1}{(3.142 \text{ m}^2)(1)} + \frac{1 - \varepsilon_2}{(5.196 \text{ m}^2)\varepsilon_2}}$$

$$\varepsilon_2 = \mathbf{0.78}$$

15-66 Radiation heat transfer occurs between a sphere and a circular disk. The view factors and the net rate of radiation heat transfer for the existing and modified cases are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of sphere and disk are given to be $\varepsilon_1 = 0.9$ and $\varepsilon_2 = 0.5$, respectively.

Analysis (a) We take the sphere to be surface 1 and the disk to be surface 2. The view factor from surface 1 to surface 2 is determined from

$$F_{12} = 0.5 \left\{ 1 - \left[1 + \left(\frac{r_2}{h} \right)^2 \right]^{-0.5} \right\} = 0.5 \left\{ 1 - \left[1 + \left(\frac{1.2 \text{ m}}{0.60 \text{ m}} \right)^2 \right]^{-0.5} \right\} = \mathbf{0.2764}$$

The view factor from surface 2 to surface 1 is determined from reciprocity relation:

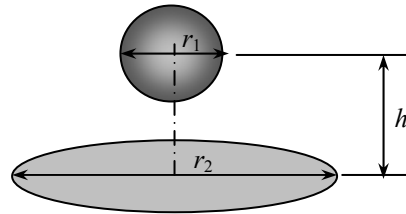
$$A_1 = 4\pi r_1^2 = 4\pi(0.3 \text{ m})^2 = 1.131 \text{ m}^2$$

$$A_2 = \pi r_2^2 = \pi(1.2 \text{ m})^2 = 4.524 \text{ m}^2$$

$$A_1 F_{12} = A_2 F_{21}$$

$$(1.131)(0.2764) = (4.524)F_{21}$$

$$F_{21} = \mathbf{0.0691}$$



(b) The net rate of radiation heat transfer between the surfaces can be determined from

$$\dot{Q} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}} = \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(873 \text{ K})^4 - (473 \text{ K})^4]}{\frac{1-0.9}{(1.131 \text{ m}^2)(0.9)} + \frac{1}{(1.131 \text{ m}^2)(0.2764)} + \frac{1-0.5}{(4.524 \text{ m}^2)(0.5)}} = \mathbf{8550 \text{ W}}$$

(c) The best values are $\varepsilon_1 = \varepsilon_2 = 1$ and $h = r_1 = 0.3 \text{ m}$. Then the view factor becomes

$$F_{12} = 0.5 \left\{ 1 - \left[1 + \left(\frac{r_2}{h} \right)^2 \right]^{-0.5} \right\} = 0.5 \left\{ 1 - \left[1 + \left(\frac{1.2 \text{ m}}{0.30 \text{ m}} \right)^2 \right]^{-0.5} \right\} = 0.3787$$

The net rate of radiation heat transfer in this case is

$$\dot{Q} = A_1 F_{12} \sigma(T_1^4 - T_2^4) = (1.131 \text{ m}^2)(0.3787)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(873 \text{ K})^4 - (473 \text{ K})^4] = \mathbf{12,890 \text{ W}}$$

15-67E Top and side surfaces of a cubical furnace are black, and are maintained at uniform temperatures. Net radiation heat transfer rate to the base from the top and side surfaces are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities are given to be $\varepsilon = 0.7$ for the bottom surface and 1 for other surfaces.

Analysis We consider the base surface to be surface 1, the top surface to be surface 2 and the side surfaces to be surface 3. The cubical furnace can be considered to be three-surface enclosure. The areas and blackbody emissive powers of surfaces are

$$A_1 = A_2 = (10 \text{ ft})^2 = 100 \text{ ft}^2 \quad A_3 = 4(10 \text{ ft})^2 = 400 \text{ ft}^2$$

$$E_{b1} = \sigma T_1^4 = (0.1714 \times 10^{-8} \text{ Btu/h.ft}^2 \cdot \text{R}^4)(800 \text{ R})^4 = 702 \text{ Btu/h.ft}^2$$

$$E_{b2} = \sigma T_2^4 = (0.1714 \times 10^{-8} \text{ Btu/h.ft}^2 \cdot \text{R}^4)(1600 \text{ R})^4 = 11,233 \text{ Btu/h.ft}^2$$

$$E_{b3} = \sigma T_3^4 = (0.1714 \times 10^{-8} \text{ Btu/h.ft}^2 \cdot \text{R}^4)(2400 \text{ R})^4 = 56,866 \text{ Btu/h.ft}^2$$

The view factor from the base to the top surface of the cube is $F_{12} = 0.2$. From the summation rule, the view factor from the base or top to the side surfaces is

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1 - F_{12} = 1 - 0.2 = 0.8$$

since the base surface is flat and thus $F_{11} = 0$. Then the radiation resistances become

$$R_1 = \frac{1 - \varepsilon_1}{A_1 \varepsilon_1} = \frac{1 - 0.7}{(100 \text{ ft}^2)(0.7)} = 0.0043 \text{ ft}^{-2} \quad R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{(100 \text{ ft}^2)(0.2)} = 0.0500 \text{ ft}^{-2}$$

$$R_{13} = \frac{1}{A_1 F_{13}} = \frac{1}{(100 \text{ ft}^2)(0.8)} = 0.0125 \text{ ft}^{-2}$$

Note that the side and the top surfaces are black, and thus their radiosities are equal to their emissive powers. The radiosity of the base surface is determined

$$\frac{E_{b1} - J_1}{R_1} + \frac{E_{b2} - J_1}{R_{12}} + \frac{E_{b3} - J_1}{R_{13}} = 0$$

$$\text{Substituting,} \quad \frac{702 - J_1}{0.0043} + \frac{11,233 - J_1}{0.05} + \frac{56,866 - J_1}{0.0125} = 0 \longrightarrow J_1 = 15,054 \text{ W/m}^2$$

(a) The net rate of radiation heat transfer between the base and the side surfaces is

$$\dot{Q}_{31} = \frac{E_{b3} - J_1}{R_{13}} = \frac{(56,866 - 15,054) \text{ Btu/h.ft}^2}{0.0125 \text{ ft}^{-2}} = \mathbf{3.345 \times 10^6 \text{ Btu/h}}$$

(b) The net rate of radiation heat transfer between the base and the top surfaces is

$$\dot{Q}_{12} = \frac{J_1 - E_{b2}}{R_{12}} = \frac{(15,054 - 11,233) \text{ Btu/h.ft}^2}{0.05 \text{ ft}^{-2}} = \mathbf{7.642 \times 10^4 \text{ Btu/h}}$$

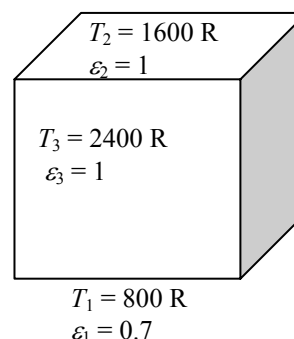
The net rate of radiation heat transfer to the base surface is finally determined from

$$\dot{Q}_1 = \dot{Q}_{21} + \dot{Q}_{31} = -76,420 + 3,344,960 = \mathbf{3.269 \times 10^6 \text{ Btu/h}}$$

Discussion The same result can be found from

$$\dot{Q}_1 = \frac{J_1 - E_{b1}}{R_1} = \frac{(15,054 - 702) \text{ Btu/h.ft}^2}{0.0043 \text{ ft}^{-2}} = 3.338 \times 10^6 \text{ Btu/h}$$

The small difference is due to round-off error.



15-68E EES Prob. 15-67E is reconsidered. The effect of base surface emissivity on the net rates of radiation heat transfer between the base and the side surfaces, between the base and top surfaces, and to the base surface is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

a=10 [ft]
epsilon_1=0.7
T_1=800 [R]
T_2=1600 [R]
T_3=2400 [R]

"ANALYSIS"

sigma=0.1714E-8 [Btu/h-ft^2-R^4] "Stefan-Boltzmann constant"

"Consider the base surface 1, the top surface 2, and the side surface 3"

E_b1=sigma*T_1^4

E_b2=sigma*T_2^4

E_b3=sigma*T_3^4

A_1=a^2

A_2=A_1

A_3=4*a^2

F_12=0.2 "view factor from the base to the top of a cube"

F_11+F_12+F_13=1 "summation rule"

F_11=0 "since the base surface is flat"

R_1=(1-epsilon_1)/(A_1*epsilon_1) "surface resistance"

R_12=1/(A_1*F_12) "space resistance"

R_13=1/(A_1*F_13) "space resistance"

(E_b1-J_1)/R_1+(E_b2-J_1)/R_12+(E_b3-J_1)/R_13=0 "J_1 : radiosity of base surface"

"(a)"

Q_dot_31=(E_b3-J_1)/R_13

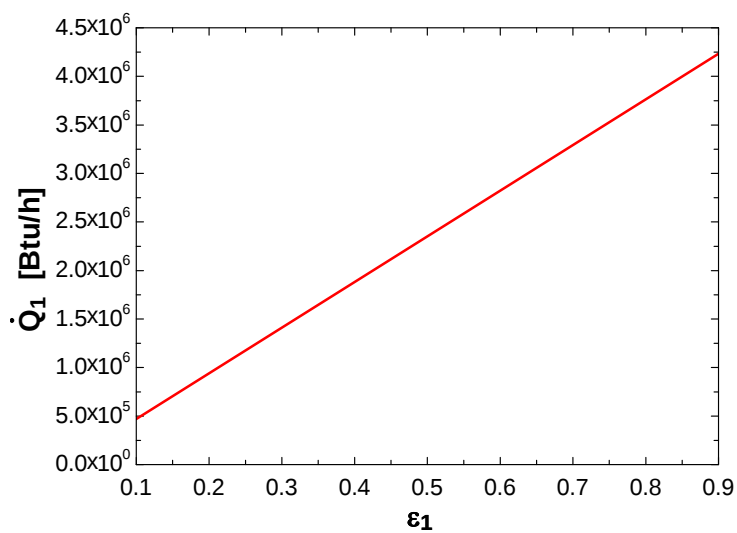
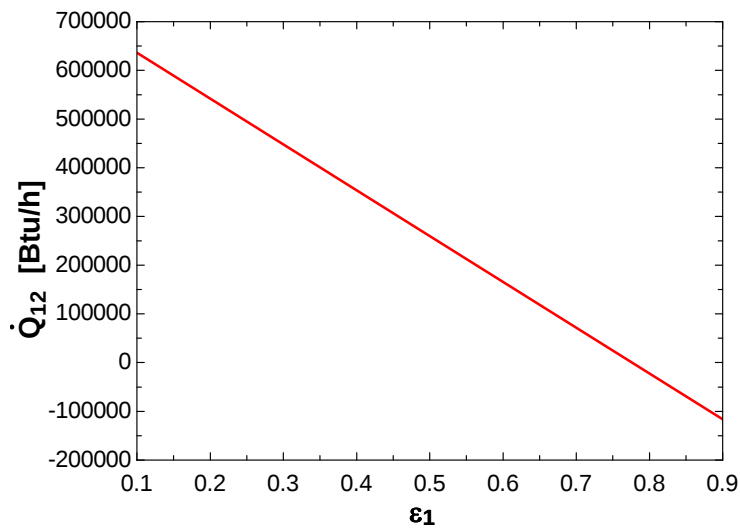
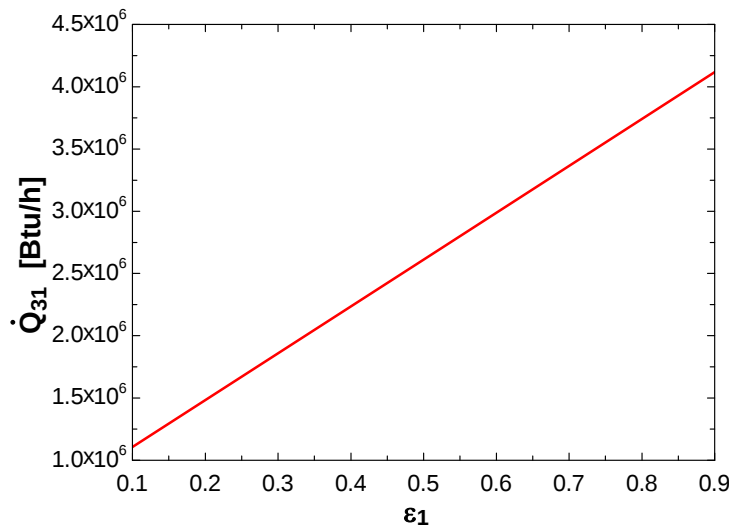
"(b)"

Q_dot_12=(J_1-E_b2)/R_12

Q_dot_21=-Q_dot_12

Q_dot_1=Q_dot_21+Q_dot_31

ϵ_1	Q_{31} [Btu/h]	Q_{12} [Btu/h]	Q_1 [Btu/h]
0.1	1.106E+06	636061	470376
0.15	1.295E+06	589024	705565
0.2	1.483E+06	541986	940753
0.25	1.671E+06	494948	1.176E+06
0.3	1.859E+06	447911	1.411E+06
0.35	2.047E+06	400873	1.646E+06
0.4	2.235E+06	353835	1.882E+06
0.45	2.423E+06	306798	2.117E+06
0.5	2.612E+06	259760	2.352E+06
0.55	2.800E+06	212722	2.587E+06
0.6	2.988E+06	165685	2.822E+06
0.65	3.176E+06	118647	3.057E+06
0.7	3.364E+06	71610	3.293E+06
0.75	3.552E+06	24572	3.528E+06
0.8	3.741E+06	-22466	3.763E+06
0.85	3.929E+06	-69503	3.998E+06
0.9	4.117E+06	-116541	4.233E+06

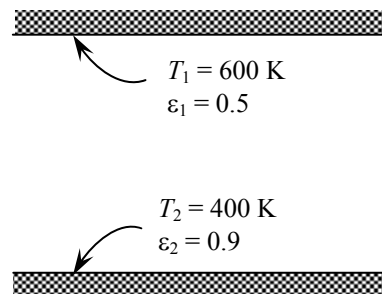


15-69 Two very large parallel plates are maintained at uniform temperatures. The net rate of radiation heat transfer between the two plates is to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivities ε of the plates are given to be 0.5 and 0.9.

Analysis The net rate of radiation heat transfer between the two surfaces per unit area of the plates is determined directly from



$$\frac{\dot{Q}_{12}}{A_s} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} = \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(600 \text{ K})^4 - (400 \text{ K})^4]}{\frac{1}{0.5} + \frac{1}{0.9} - 1} = \mathbf{2793 \text{ W/m}^2}$$

15-70 EES Prob. 15-69 is reconsidered. The effects of the temperature and the emissivity of the hot plate on the net rate of radiation heat transfer between the plates are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$T_1=600$ [K]

$T_2=400$ [K]

$\epsilon_1=0.5$

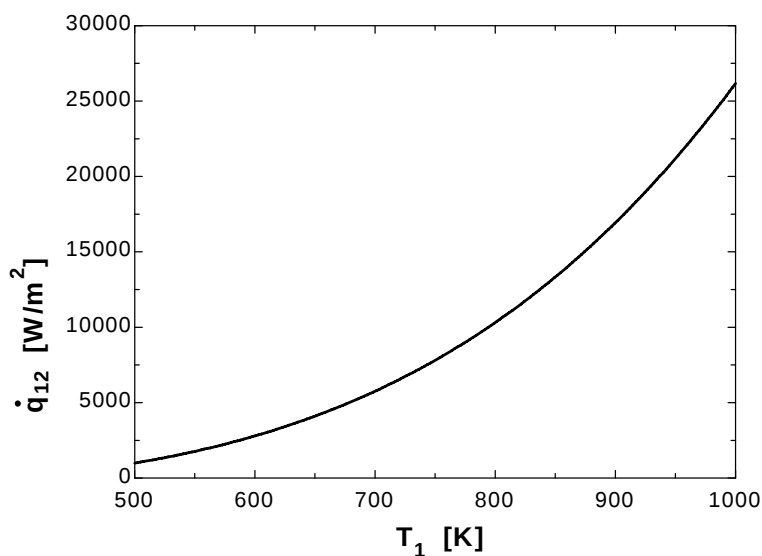
$\epsilon_2=0.9$

$\sigma=5.67E-8$ [W/m²·K⁴] "Stefan-Boltzmann constant"

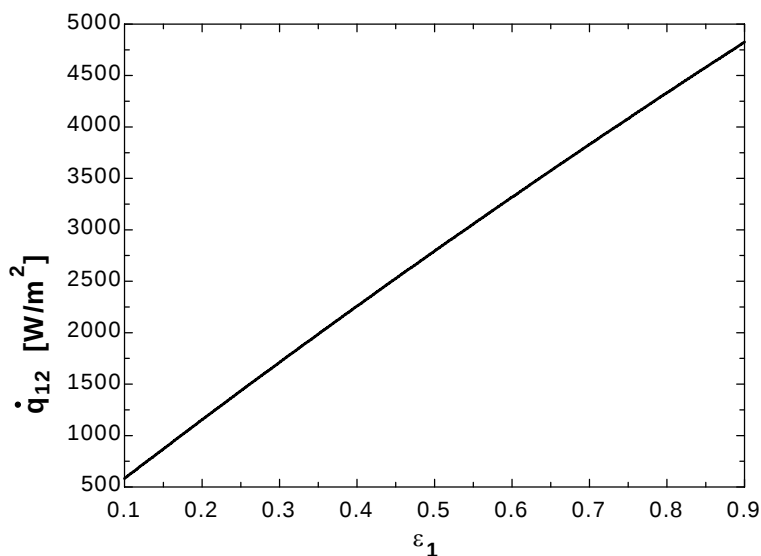
"ANALYSIS"

$\dot{q}_{12}=(\sigma*(T_1^4-T_2^4))/(1/\epsilon_1+1/\epsilon_2-1)$

T_1 [K]	$\dot{q}_{12}$ [W/m ²]
500	991.1
525	1353
550	1770
575	2248
600	2793
625	3411
650	4107
675	4888
700	5761
725	6733
750	7810
775	9001
800	10313
825	11754
850	13332
875	15056
900	16934
925	18975
950	21188
975	23584
1000	26170



ϵ_1	$\dot{q}_{12}$ [W/m ²]
0.1	583.2
0.15	870
0.2	1154
0.25	1434
0.3	1712
0.35	1987
0.4	2258
0.45	2527
0.5	2793
0.55	3056
0.6	3317
0.65	3575
0.7	3830
0.75	4082
0.8	4332
0.85	4580
0.9	4825



15-71 The base, top, and side surfaces of a furnace of cylindrical shape are black, and are maintained at uniform temperatures. The net rate of radiation heat transfer to or from the top surface is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are black. 3 Convection heat transfer is not considered.

Properties The emissivity of all surfaces are $\varepsilon = 1$ since they are black.

Analysis We consider the top surface to be surface 1, the base surface to be surface 2 and the side surfaces to be surface 3. The cylindrical furnace can be considered to be three-surface enclosure. We assume that steady-state conditions exist. Since all surfaces are black, the radiosities are equal to the emissive power of surfaces, and the net rate of radiation heat transfer from the top surface can be determined from

$$\dot{Q} = A_1 F_{12} \sigma (T_1^4 - T_2^4) + A_1 F_{13} \sigma (T_1^4 - T_3^4)$$

and $A_1 = \pi r^2 = \pi (2 \text{ m})^2 = 12.57 \text{ m}^2$

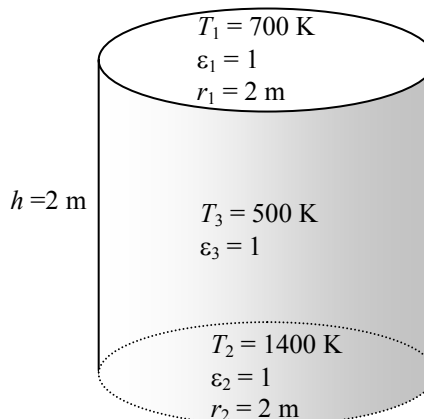
The view factor from the base to the top surface of the cylinder is $F_{12} = 0.38$ (From Figure 15-7). The view factor from the base to the side surfaces is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1 - F_{12} = 1 - 0.38 = 0.62$$

Substituting,

$$\begin{aligned} \dot{Q} &= A_1 F_{12} \sigma (T_1^4 - T_2^4) + A_1 F_{13} \sigma (T_1^4 - T_3^4) \\ &= (12.57 \text{ m}^2)(0.38)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(700 \text{ K}^4 - 500 \text{ K}^4) \\ &\quad + (12.57 \text{ m}^2)(0.62)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(700 \text{ K}^4 - 1400 \text{ K}^4) \\ &= -1.543 \times 10^6 \text{ W} = \mathbf{-1543 \text{ kW}} \end{aligned}$$

Discussion The negative sign indicates that net heat transfer is to the top surface.



15-72 The base and the dome of a hemispherical furnace are maintained at uniform temperatures. The net rate of radiation heat transfer from the dome to the base surface is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

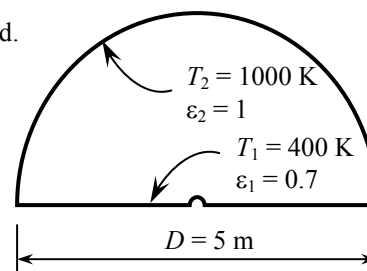
Analysis The view factor is first determined from

$$F_{11} = 0 \text{ (flat surface)}$$

$$F_{11} + F_{12} = 1 \rightarrow F_{12} = 1 \text{ (summation rule)}$$

Noting that the dome is black, net rate of radiation heat transfer from dome to the base surface can be determined from

$$\begin{aligned} \dot{Q}_{21} &= -\dot{Q}_{12} = -\varepsilon A_1 F_{12} \sigma (T_1^4 - T_2^4) \\ &= -(0.7)[\pi (5 \text{ m})^2 / 4](1)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(400 \text{ K})^4 - (1000 \text{ K})^4] \\ &= 7.594 \times 10^5 \text{ W} \\ &= \mathbf{759 \text{ kW}} \end{aligned}$$



The positive sign indicates that the net heat transfer is from the dome to the base surface, as expected.

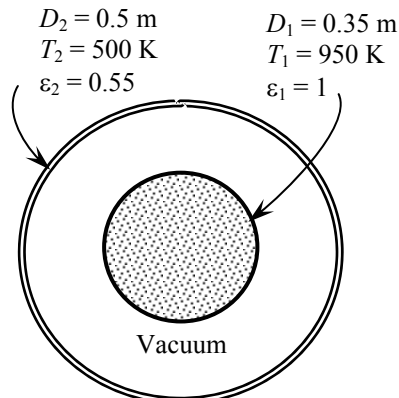
15-73 Two very long concentric cylinders are maintained at uniform temperatures. The net rate of radiation heat transfer between the two cylinders is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 1$ and $\varepsilon_2 = 0.55$.

Analysis The net rate of radiation heat transfer between the two cylinders per unit length of the cylinders is determined from

$$\begin{aligned}\dot{Q}_{12} &= \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1}{r_2} \right)} \\ &= \frac{[\pi(0.35 \text{ m})(1 \text{ m})](5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(950 \text{ K})^4 - (500 \text{ K})^4]}{\frac{1}{1} + \frac{1 - 0.55}{0.55} \left(\frac{3.5}{5} \right)} \\ &= 29,810 \text{ W} = \mathbf{29.81 \text{ kW}}\end{aligned}$$



15-74 A long cylindrical rod coated with a new material is placed in an evacuated long cylindrical enclosure which is maintained at a uniform temperature. The emissivity of the coating on the rod is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray.

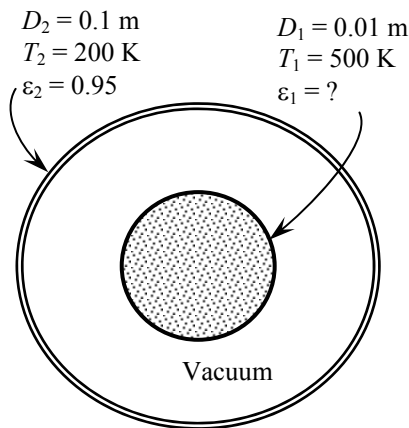
Properties The emissivity of the enclosure is given to be $\varepsilon_2 = 0.95$.

Analysis The emissivity of the coating on the rod is determined from

$$\begin{aligned}\dot{Q}_{12} &= \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1}{r_2} \right)} \\ 8 \text{ W} &= \frac{[\pi(0.01 \text{ m})(1 \text{ m})](5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(500 \text{ K})^4 - (200 \text{ K})^4]}{\frac{1}{\varepsilon_1} + \frac{1 - 0.95}{0.95} \left(\frac{1}{10} \right)}\end{aligned}$$

which gives

$$\varepsilon_1 = \mathbf{0.074}$$



15-75E The base and the dome of a long semicylindrical duct are maintained at uniform temperatures. The net rate of radiation heat transfer from the dome to the base surface is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.5$ and $\varepsilon_2 = 0.9$.

Analysis The view factor from the base to the dome is first determined from

$$F_{11} = 0 \text{ (flat surface)}$$

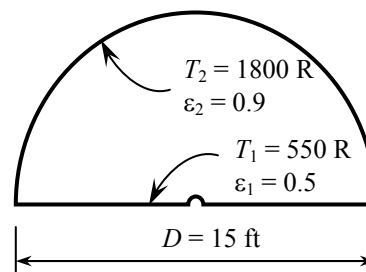
$$F_{11} + F_{12} = 1 \rightarrow F_{12} = 1 \text{ (summation rule)}$$

The net rate of radiation heat transfer from dome to the base surface can be determined from

$$\dot{Q}_{21} = -\dot{Q}_{12} = -\frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}} = -\frac{(0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)[(550 \text{ R})^4 - (1800 \text{ R})^4]}{\frac{1-0.5}{(15 \text{ ft}^2)(0.5)} + \frac{1}{(15 \text{ ft}^2)(1)} + \frac{1-0.9}{\left[\frac{\pi(15 \text{ ft})(1 \text{ ft})}{2}\right](0.9)}}$$

$$= \mathbf{129,200 \text{ Btu/h}} \text{ per ft length}$$

The positive sign indicates that the net heat transfer is from the dome to the base surface, as expected.



15-76 Two parallel disks whose back sides are insulated are black, and are maintained at a uniform temperature. The net rate of radiation heat transfer from the disks to the environment is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of all surfaces are $\varepsilon = 1$ since they are black.

Analysis Both disks possess same properties and they are black. Noting that environment can also be considered to be blackbody, we can treat this geometry as a three surface enclosure. We consider the two disks to be surfaces 1 and 2 and the environment to be surface 3. Then from Figure 15-7, we read

$$F_{12} = F_{21} = 0.26$$

$$F_{13} = 1 - 0.26 = 0.74 \quad \text{(summation rule)}$$

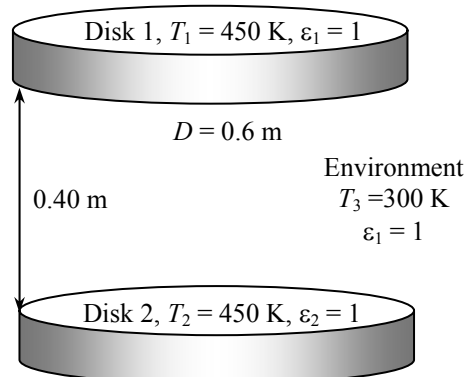
The net rate of radiation heat transfer from the disks into the environment then becomes

$$\dot{Q}_3 = \dot{Q}_{13} + \dot{Q}_{23} = 2\dot{Q}_{13}$$

$$\dot{Q}_3 = 2F_{13}A_1\sigma(T_1^4 - T_3^4)$$

$$= 2(0.74)[\pi(0.3 \text{ m})^2](5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(450 \text{ K})^4 - (300 \text{ K})^4]$$

$$= \mathbf{781 \text{ W}}$$



15-77 A furnace shaped like a long equilateral-triangular duct is considered. The temperature of the base surface is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered. 4 End effects are neglected.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.8$ and $\varepsilon_2 = 0.5$.

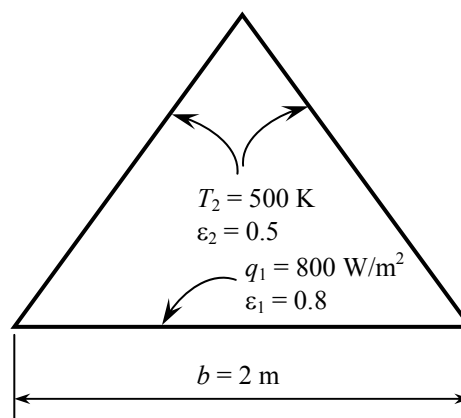
Analysis This geometry can be treated as a two surface enclosure since two surfaces have identical properties. We consider base surface to be surface 1 and other two surface to be surface 2. Then the view factor between the two becomes $F_{12} = 1$. The temperature of the base surface is determined from

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$800 \text{ W} = \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(T_1)^4 - (500 \text{ K})^4]}{\frac{1-0.8}{(1 \text{ m}^2)(0.8)} + \frac{1}{(1 \text{ m}^2)(1)} + \frac{1-0.5}{(2 \text{ m}^2)(0.5)}}$$

$$T_1 = \mathbf{543 \text{ K}}$$

Note that $A_1 = 1 \text{ m}^2$ and $A_2 = 2 \text{ m}^2$.



15-78 EES Prob. 15-77 is reconsidered. The effects of the rate of the heat transfer at the base surface and the temperature of the side surfaces on the temperature of the base surface are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

a=2 [m]
 epsilon_1=0.8
 epsilon_2=0.5
 Q_dot_12=800 [W]
 T_2=500 [K]
 sigma=5.67E-8 [W/m^2-K^4]

"ANALYSIS"

"Consider the base surface to be surface 1, the side surfaces to be surface 2"

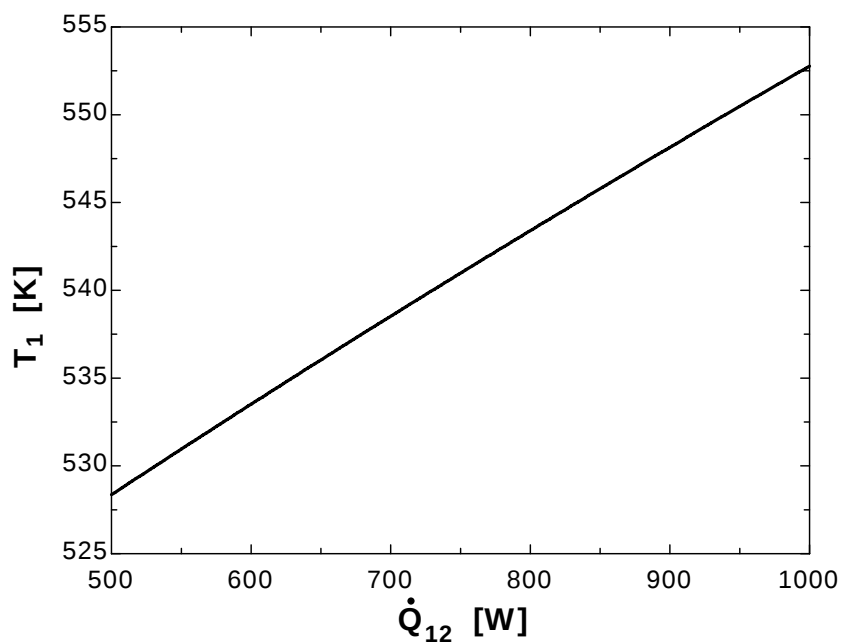
$$Q_{dot_{12}} = (\sigma(T_1^4 - T_2^4)) / ((1 - \epsilon_1) / (A_1 \epsilon_1) + 1 / (A_1 F_{12}) + (1 - \epsilon_2) / (A_2 \epsilon_2))$$

F_12=1

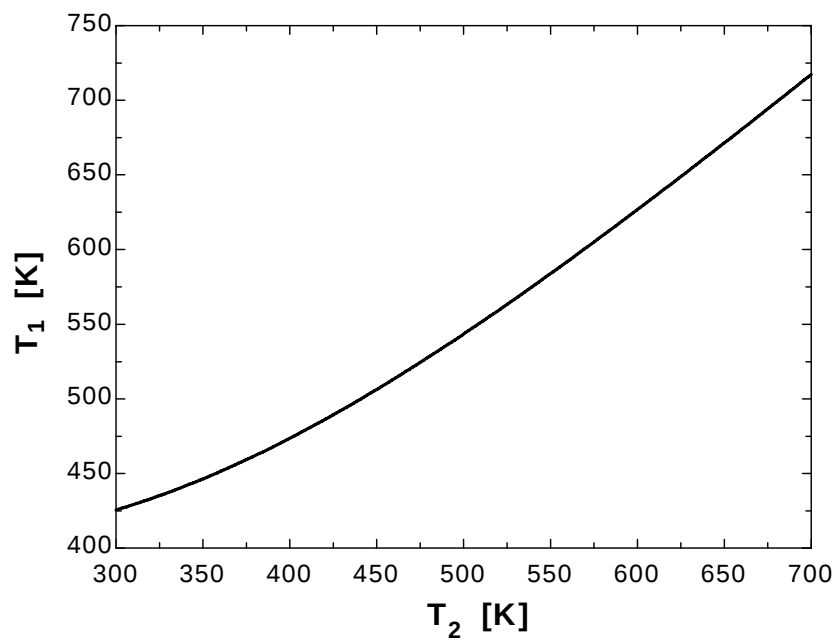
A_1=1 "[m^2], since rate of heat supply is given per meter square area"

A_2=2*A_1

Q ₁₂ [W]	T ₁ [K]
500	528.4
525	529.7
550	531
575	532.2
600	533.5
625	534.8
650	536
675	537.3
700	538.5
725	539.8
750	541
775	542.2
800	543.4
825	544.6
850	545.8
875	547
900	548.1
925	549.3
950	550.5
975	551.6
1000	552.8



T_2 [K]	T_1 [K]
300	425.5
325	435.1
350	446.4
375	459.2
400	473.6
425	489.3
450	506.3
475	524.4
500	543.4
525	563.3
550	583.8
575	605
600	626.7
625	648.9
650	671.4
675	694.2
700	717.3



15-79 The floor and the ceiling of a cubical furnace are maintained at uniform temperatures. The net rate of radiation heat transfer between the floor and the ceiling is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of all surfaces are $\varepsilon = 1$ since they are black or reradiating.

Analysis We consider the ceiling to be surface 1, the floor to be surface 2 and the side surfaces to be surface 3. The furnace can be considered to be three-surface enclosure. We assume that steady-state conditions exist. Since the side surfaces are reradiating, there is no heat transfer through them, and the entire heat lost by the ceiling must be gained by the floor. The view factor from the ceiling to the floor of the furnace is $F_{12} = 0.2$. Then the rate of heat loss from the ceiling can be determined from

$$\dot{Q}_1 = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13} + R_{23}} \right)^{-1}}$$

where

$$E_{b1} = \sigma T_1^4 = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1100 \text{ K})^4 = 83,015 \text{ W/m}^2$$

$$E_{b2} = \sigma T_2^4 = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(550 \text{ K})^4 = 5188 \text{ W/m}^2$$

and

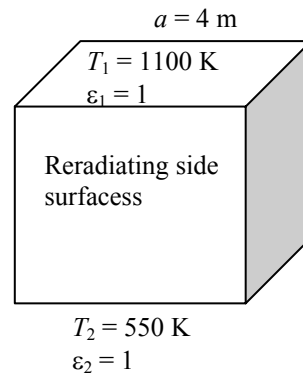
$$A_1 = A_2 = (4 \text{ m})^2 = 16 \text{ m}^2$$

$$R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{(16 \text{ m}^2)(0.2)} = 0.3125 \text{ m}^{-2}$$

$$R_{13} = R_{23} = \frac{1}{A_1 F_{13}} = \frac{1}{(16 \text{ m}^2)(0.8)} = 0.078125 \text{ m}^{-2}$$

Substituting,

$$\dot{Q}_{12} = \frac{(83,015 - 5188) \text{ W/m}^2}{\left(\frac{1}{0.3125 \text{ m}^{-2}} + \frac{1}{2(0.078125 \text{ m}^{-2})} \right)^{-1}} = 7.47 \times 10^5 \text{ W} = \mathbf{747 \text{ kW}}$$



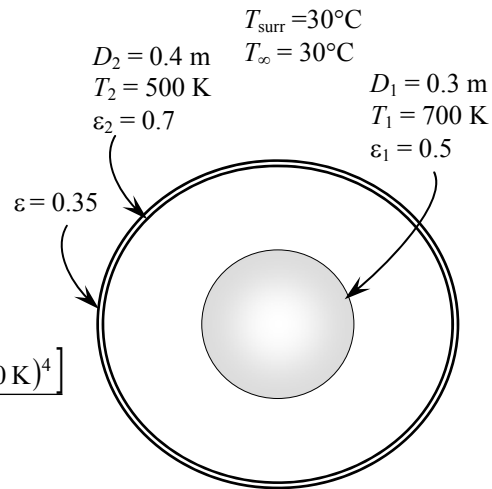
15-80 Two concentric spheres are maintained at uniform temperatures. The net rate of radiation heat transfer between the two spheres and the convection heat transfer coefficient at the outer surface are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.5$ and $\varepsilon_2 = 0.7$.

Analysis The net rate of radiation heat transfer between the two spheres is

$$\begin{aligned}\dot{Q}_{12} &= \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1^2}{r_2^2} \right)} \\ &= \frac{[\pi(0.3 \text{ m})^2] [5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4] [(700 \text{ K})^4 - (500 \text{ K})^4]}{\frac{1}{0.5} + \frac{1 - 0.7}{0.7} \left(\frac{0.15 \text{ m}}{0.2 \text{ m}} \right)^2} \\ &= \mathbf{1270 \text{ W}}\end{aligned}$$



Radiation heat transfer rate from the outer sphere to the surrounding surfaces are

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon F A_2 \sigma (T_2^4 - T_{surr}^4) \\ &= (0.35)(1)[\pi(0.4 \text{ m})^2] (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(500 \text{ K})^4 - (30 + 273 \text{ K})^4] \\ &= 539 \text{ W}\end{aligned}$$

The convection heat transfer rate at the outer surface of the cylinder is determined from requirement that heat transferred from the inner sphere to the outer sphere must be equal to the heat transfer from the outer surface of the outer sphere to the environment by convection and radiation. That is,

$$\dot{Q}_{conv} = \dot{Q}_{12} - \dot{Q}_{rad} = 1270 - 539 = 731 \text{ W}$$

Then the convection heat transfer coefficient becomes

$$\begin{aligned}\dot{Q}_{conv} &= h A_2 (T_2 - T_\infty) \\ 731 \text{ W} &= h [\pi(0.4 \text{ m})^2] (500 \text{ K} - 303 \text{ K}) \\ h &= \mathbf{7.4 \text{ W/m}^2 \cdot ^\circ\text{C}}\end{aligned}$$

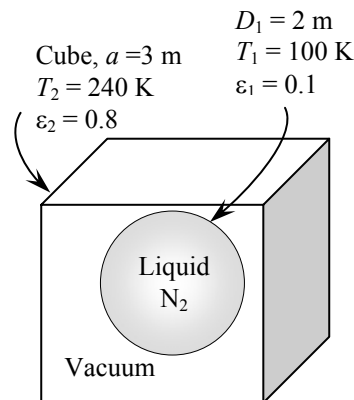
15-81 A spherical tank filled with liquid nitrogen is kept in an evacuated cubic enclosure. The net rate of radiation heat transfer to the liquid nitrogen is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered. 4 The thermal resistance of the tank is negligible.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.1$ and $\varepsilon_2 = 0.8$.

Analysis We take the sphere to be surface 1 and the surrounding cubic enclosure to be surface 2. Noting that $F_{12} = 1$, for this two-surface enclosure, the net rate of radiation heat transfer to liquid nitrogen can be determined from

$$\begin{aligned}\dot{Q}_{21} = -\dot{Q}_{12} &= -\frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{A_1}{A_2} \right)} \\ &= -\frac{[\pi(2 \text{ m})^2] [5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4] [(100 \text{ K})^4 - (240 \text{ K})^4]}{\frac{1}{0.1} + \frac{1 - 0.8}{0.8} \left[\frac{\pi(2 \text{ m})^2}{6(3 \text{ m})^2} \right]} \\ &= \mathbf{228 \text{ W}}\end{aligned}$$



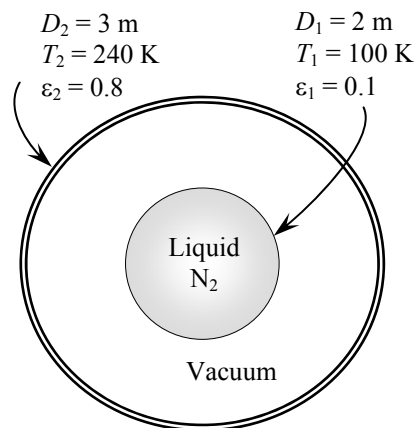
15-82 A spherical tank filled with liquid nitrogen is kept in an evacuated spherical enclosure. The net rate of radiation heat transfer to the liquid nitrogen is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered. 4 The thermal resistance of the tank is negligible.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.1$ and $\varepsilon_2 = 0.8$.

Analysis The net rate of radiation heat transfer to liquid nitrogen can be determined from

$$\begin{aligned}\dot{Q}_{12} &= \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1^2}{r_2^2} \right)} \\ &= \frac{[\pi(2 \text{ m})^2] [5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4] [(240 \text{ K})^4 - (100 \text{ K})^4]}{\frac{1}{0.1} + \frac{1 - 0.8}{0.8} \left[\frac{(1 \text{ m})^2}{(1.5 \text{ m})^2} \right]} \\ &= \mathbf{227 \text{ W}}\end{aligned}$$



15-83 EES Prob. 15-81 is reconsidered. The effects of the side length and the emissivity of the cubic enclosure, and the emissivity of the spherical tank on the net rate of radiation heat transfer are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D=2 [m]

a=3 [m]

T_1=100 [K]

T_2=240 [K]

epsilon_1=0.1

epsilon_2=0.8

sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzmann constant"

"ANALYSIS"

"Consider the sphere to be surface 1, the surrounding cubic enclosure to be surface 2"

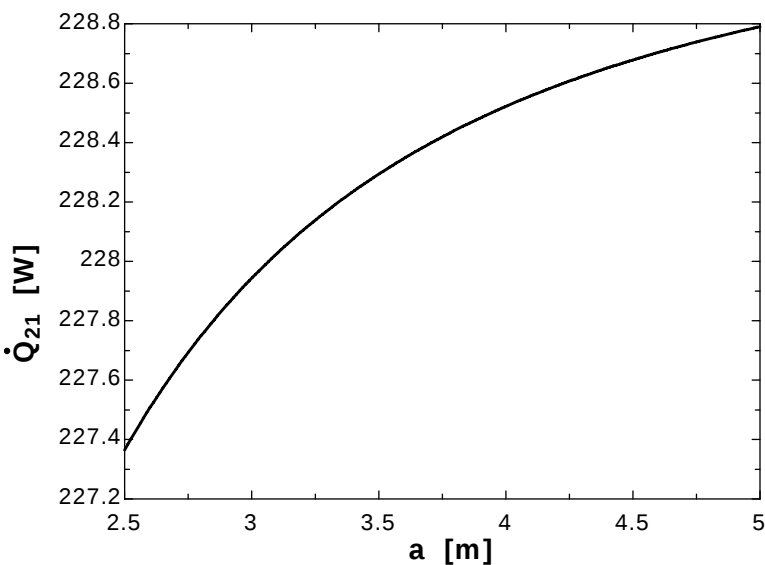
$\dot{Q}_{12} = (A_1 \sigma (T_1^4 - T_2^4)) / (1/\epsilon_1 + (1 - \epsilon_2)/\epsilon_2 * (A_1/A_2))$

$\dot{Q}_{21} = -\dot{Q}_{12}$

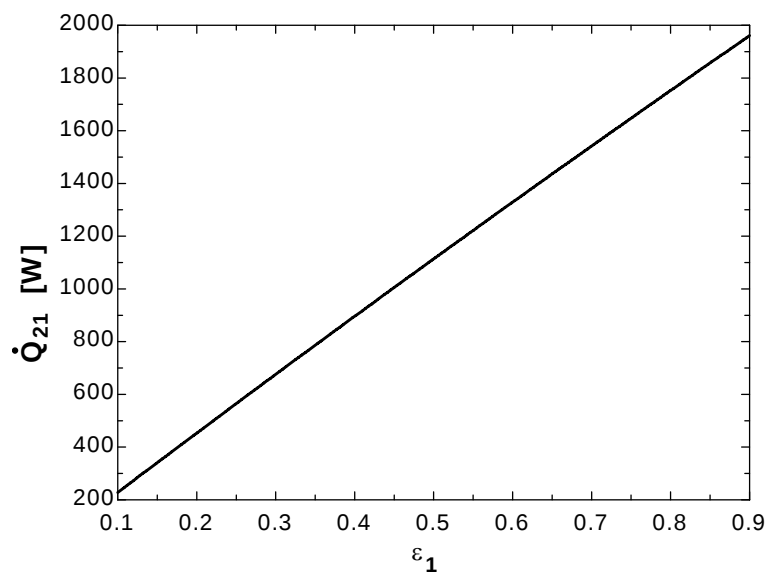
$A_1 = \pi D^2$

$A_2 = 6a^2$

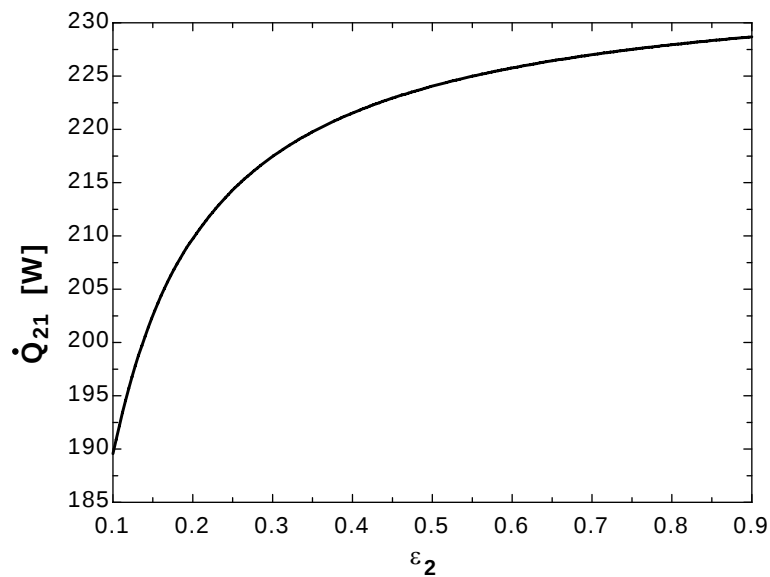
a [m]	$\dot{Q}_{21}$ [W]
2.5	227.4
2.625	227.5
2.75	227.7
2.875	227.8
3	227.9
3.125	228
3.25	228.1
3.375	228.2
3.5	228.3
3.625	228.4
3.75	228.4
3.875	228.5
4	228.5
4.125	228.6
4.25	228.6
4.375	228.6
4.5	228.7
4.625	228.7
4.75	228.7
4.875	228.8
5	228.8



ε_1	$\dot{Q}_{21}$ [W]
0.1	227.9
0.15	340.9
0.2	453.3
0.25	565
0.3	676
0.35	786.4
0.4	896.2
0.45	1005
0.5	1114
0.55	1222
0.6	1329
0.65	1436
0.7	1542
0.75	1648
0.8	1753
0.85	1857
0.9	1961



ε_2	$\dot{Q}_{21}$ [W]
0.1	189.6
0.15	202.6
0.2	209.7
0.25	214.3
0.3	217.5
0.35	219.8
0.4	221.5
0.45	222.9
0.5	224.1
0.55	225
0.6	225.8
0.65	226.4
0.7	227
0.75	227.5
0.8	227.9
0.85	228.3
0.9	228.7



15-84 A circular grill is considered. The bottom of the grill is covered with hot coal bricks, while the wire mesh on top of the grill is covered with steaks. The initial rate of radiation heat transfer from coal bricks to the steaks is to be determined for two cases.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities are $\varepsilon = 1$ for all surfaces since they are black or reradiating.

Analysis We consider the coal bricks to be surface 1, the steaks to be surface 2 and the side surfaces to be surface 3. First we determine the view factor between the bricks and the steaks (Table 15-3),

$$R_i = R_j = \frac{r_i}{L} = \frac{0.15 \text{ m}}{0.20 \text{ m}} = 0.75$$

$$S = 1 + \frac{1 + R_j^2}{R_i^2} = \frac{1 + 0.75^2}{0.75^2} = 3.7778$$

$$F_{12} = F_{ij} = \frac{1}{2} \left\{ S - \left[S^2 - 4 \left(\frac{R_j}{R_i} \right)^2 \right]^{1/2} \right\} = \frac{1}{2} \left\{ 3.7778 - \left[3.7778^2 - 4 \left(\frac{0.75}{0.75} \right)^2 \right]^{1/2} \right\} = 0.2864$$

(It can also be determined from Fig. 15-7).

Then the initial rate of radiation heat transfer from the coal bricks to the stakes becomes

$$\begin{aligned} \dot{Q}_{12} &= F_{12} A_1 \sigma (T_1^4 - T_2^4) \\ &= (0.2864) [\pi (0.3 \text{ m})^2 / 4] (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(950 \text{ K})^4 - (278 \text{ K})^4] \\ &= \mathbf{928 \text{ W}} \end{aligned}$$

When the side opening is closed with aluminum foil, the entire heat lost by the coal bricks must be gained by the steaks since there will be no heat transfer through a reradiating surface. The grill can be considered to be three-surface enclosure. Then the rate of heat loss from the coal bricks can be determined from

$$\dot{Q}_1 = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13} + R_{23}} \right)^{-1}}$$

where $E_{b1} = \sigma T_1^4 = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) (950 \text{ K})^4 = 46,183 \text{ W/m}^2$

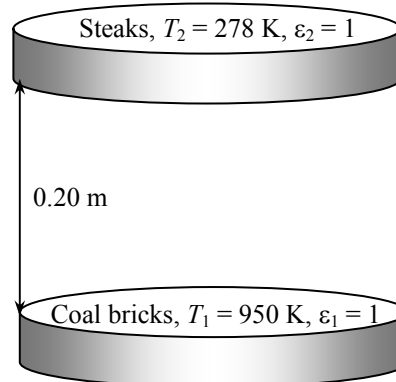
$$E_{b2} = \sigma T_2^4 = (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) (278 \text{ K})^4 = 339 \text{ W/m}^2$$

and $A_1 = A_2 = \frac{\pi (0.3 \text{ m})^2}{4} = 0.07069 \text{ m}^2$

$$R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{(0.07069 \text{ m}^2) (0.2864)} = 49.39 \text{ m}^{-2}$$

$$R_{13} = R_{23} = \frac{1}{A_1 F_{13}} = \frac{1}{(0.07069 \text{ m}^2) (1 - 0.2864)} = 19.82 \text{ m}^{-2}$$

Substituting, $\dot{Q}_{12} = \frac{(46,183 - 339) \text{ W/m}^2}{\left(\frac{1}{49.39 \text{ m}^{-2}} + \frac{1}{2(19.82 \text{ m}^{-2})} \right)^{-1}} = \mathbf{2085 \text{ W}}$

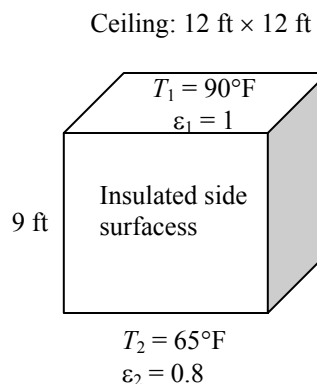


15-85E A room is heated by electric resistance heaters placed on the ceiling which is maintained at a uniform temperature. The rate of heat loss from the room through the floor is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered. 4 There is no heat loss through the side surfaces.

Properties The emissivities are $\varepsilon = 1$ for the ceiling and $\varepsilon = 0.8$ for the floor. The emissivity of insulated (or reradiating) surfaces is also 1.

Analysis The room can be considered to be three-surface enclosure with the ceiling surface 1, the floor surface 2 and the side surfaces surface 3. We assume steady-state conditions exist. Since the side surfaces are reradiating, there is no heat transfer through them, and the entire heat lost by the ceiling must be gained by the floor. Then the rate of heat loss from the room through its floor can be determined from



$$\dot{Q}_1 = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13} + R_{23}} \right)^{-1} + R_2}$$

where

$$E_{b1} = \sigma T_1^4 = (0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)(90 + 460 \text{ R})^4 = 157 \text{ Btu/h} \cdot \text{ft}^2$$

$$E_{b2} = \sigma T_2^4 = (0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)(65 + 460 \text{ R})^4 = 130 \text{ Btu/h} \cdot \text{ft}^2$$

and

$$A_1 = A_2 = (12 \text{ ft})^2 = 144 \text{ ft}^2$$

The view factor from the floor to the ceiling of the room is $F_{12} = 0.27$ (From Figure 15-5). The view factor from the ceiling or the floor to the side surfaces is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1 - F_{12} = 1 - 0.27 = 0.73$$

since the ceiling is flat and thus $F_{11} = 0$. Then the radiation resistances which appear in the equation above become

$$R_2 = \frac{1 - \varepsilon_2}{A_2 \varepsilon_2} = \frac{1 - 0.8}{(144 \text{ ft}^2)(0.8)} = 0.00174 \text{ ft}^{-2}$$

$$R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{(144 \text{ ft}^2)(0.27)} = 0.02572 \text{ ft}^{-2}$$

$$R_{13} = R_{23} = \frac{1}{A_1 F_{13}} = \frac{1}{(144 \text{ ft}^2)(0.73)} = 0.009513 \text{ ft}^{-2}$$

Substituting,

$$\dot{Q}_{12} = \frac{(157 - 130) \text{ Btu/h} \cdot \text{ft}^2}{\left(\frac{1}{0.02572 \text{ ft}^{-2}} + \frac{1}{2(0.009513 \text{ ft}^{-2})} \right)^{-1} + 0.00174 \text{ ft}^{-2}} = \mathbf{2130 \text{ Btu/h}}$$

15-86 Two perpendicular rectangular surfaces with a common edge are maintained at specified temperatures. The net rate of radiation heat transfers between the two surfaces and between the horizontal surface and the surroundings are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of the horizontal rectangle and the surroundings are $\varepsilon = 0.75$ and $\varepsilon = 0.85$, respectively.

Analysis We consider the horizontal rectangle to be surface 1, the vertical rectangle to be surface 2 and the surroundings to be surface 3. This system can be considered to be a three-surface enclosure. The view factor from surface 1 to surface 2 is determined from

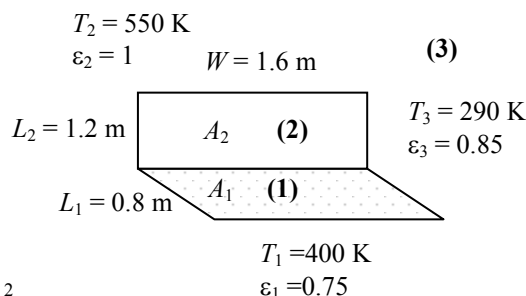
$$\left. \begin{aligned} \frac{L_1}{W} = \frac{0.8}{1.6} = 0.5 \\ \frac{L_2}{W} = \frac{1.2}{1.6} = 0.75 \end{aligned} \right\} F_{12} = 0.27 \text{ (Fig. 15-6)}$$

The surface areas are

$$A_1 = (0.8 \text{ m})(1.6 \text{ m}) = 1.28 \text{ m}^2$$

$$A_2 = (1.2 \text{ m})(1.6 \text{ m}) = 1.92 \text{ m}^2$$

$$A_3 = 2 \times \frac{1.2 \times 0.8}{2} + \sqrt{0.8^2 + 1.2^2} \times 1.6 = 3.268 \text{ m}^2$$



Note that the surface area of the surroundings is determined assuming that surroundings forms flat surfaces at all openings to form an enclosure. Then other view factors are determined to be

$$A_1 F_{12} = A_2 F_{21} \longrightarrow (1.28)(0.27) = (1.92)F_{21} \longrightarrow F_{21} = 0.18 \quad (\text{reciprocity rule})$$

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow 0 + 0.27 + F_{13} = 1 \longrightarrow F_{13} = 0.73 \quad (\text{summation rule})$$

$$F_{21} + F_{22} + F_{23} = 1 \longrightarrow 0.18 + 0 + F_{23} = 1 \longrightarrow F_{23} = 0.82 \quad (\text{summation rule})$$

$$A_1 F_{13} = A_3 F_{31} \longrightarrow (1.28)(0.73) = (3.268)F_{31} \longrightarrow F_{31} = 0.29 \quad (\text{reciprocity rule})$$

$$A_2 F_{23} = A_3 F_{32} \longrightarrow (1.92)(0.82) = (3.268)F_{32} \longrightarrow F_{32} = 0.48 \quad (\text{reciprocity rule})$$

We now apply Eq. 15-35b to each surface to determine the radiosities.

$$\begin{aligned} \sigma T_1^4 &= J_1 + \frac{1 - \varepsilon_1}{\varepsilon_1} [F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)] \\ \text{Surface 1: } (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(400 \text{ K})^4 &= J_1 + \frac{1 - 0.75}{0.75} [0.27(J_1 - J_2) + 0.73(J_1 - J_3)] \end{aligned}$$

$$\begin{aligned} \text{Surface 2: } \sigma T_2^4 &= J_2 \longrightarrow (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(550 \text{ K})^4 = J_2 \\ \sigma T_3^4 &= J_3 + \frac{1 - \varepsilon_3}{\varepsilon_3} [F_{31}(J_3 - J_1) + F_{32}(J_3 - J_2)] \end{aligned}$$

$$\begin{aligned} \text{Surface 3: } (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(290 \text{ K})^4 &= J_3 + \frac{1 - 0.85}{0.85} [0.29(J_3 - J_1) + 0.48(J_3 - J_2)] \end{aligned}$$

Solving the above equations, we find

$$J_1 = 1587 \text{ W/m}^2, \quad J_2 = 5188 \text{ W/m}^2, \quad J_3 = 811.5 \text{ W/m}^2$$

Then the net rate of radiation heat transfers between the two surfaces and between the horizontal surface and the surroundings are determined to be

$$\dot{Q}_{21} = -\dot{Q}_{12} = -A_1 F_{12} (J_1 - J_2) = -(1.28 \text{ m}^2)(0.27)(1587 - 5188) \text{ W/m}^2 = \mathbf{1245 \text{ W}}$$

$$\dot{Q}_{13} = A_1 F_{13} (J_1 - J_3) = (1.28 \text{ m}^2)(0.73)(1587 - 811.5) \text{ W/m}^2 = \mathbf{725 \text{ W}}$$

15-87 Two long parallel cylinders are maintained at specified temperatures. The rates of radiation heat transfer between the cylinders and between the hot cylinder and the surroundings are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are black. 3 Convection heat transfer is not considered.

Analysis We consider the hot cylinder to be surface 1, cold cylinder to be surface 2, and the surroundings to be surface 3. Using the crossed-strings method, the view factor between two cylinders facing each other is determined to be

$$F_{1-2} = \frac{\sum \text{Crossed strings} - \sum \text{Uncrossed strings}}{2 \times \text{String on surface 1}}$$

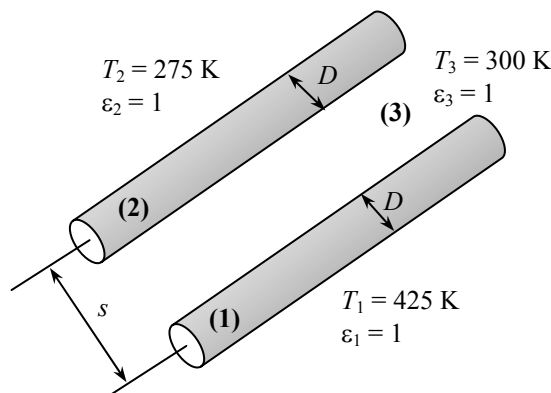
$$= \frac{2\sqrt{s^2 + D^2} - 2s}{2(\pi D / 2)}$$

or

$$F_{1-2} = \frac{2(\sqrt{s^2 + D^2} - s)}{\pi D}$$

$$= \frac{2(\sqrt{0.3^2 + 0.20^2} - 0.5)}{\pi(0.20)}$$

$$= 0.444$$



The view factor between the hot cylinder and the surroundings is

$$F_{13} = 1 - F_{12} = 1 - 0.444 = 0.556 \text{ (summation rule)}$$

The rate of radiation heat transfer between the cylinders per meter length is

$$A = \pi D L / 2 = \pi(0.20 \text{ m})(1 \text{ m}) / 2 = 0.3142 \text{ m}^2$$

$$\dot{Q}_{12} = A F_{12} \sigma (T_1^4 - T_2^4)$$

$$= (0.3142 \text{ m}^2)(0.444)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot ^\circ\text{C})(425^4 - 275^4) \text{ K}^4$$

$$= \mathbf{212.8 \text{ W}}$$

Note that half of the surface area of the cylinder is used, which is the only area that faces the other cylinder. The rate of radiation heat transfer between the hot cylinder and the surroundings per meter length of the cylinder is

$$A_1 = \pi D L = \pi(0.20 \text{ m})(1 \text{ m}) = 0.6283 \text{ m}^2$$

$$\dot{Q}_{13} = A_1 F_{13} \sigma (T_1^4 - T_3^4)$$

$$= (0.6283 \text{ m}^2)(0.556)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot ^\circ\text{C})(425^4 - 300^4) \text{ K}^4$$

$$= \mathbf{485.8 \text{ W}}$$

15-88 A long semi-cylindrical duct with specified temperature on the side surface is considered. The temperature of the base surface for a specified heat transfer rate is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivity of the side surface is $\varepsilon = 0.4$.

Analysis We consider the base surface to be surface 1, the side surface to be surface 2. This system is a two-surface enclosure, and we consider a unit length of the duct. The surface areas and the view factor are determined as

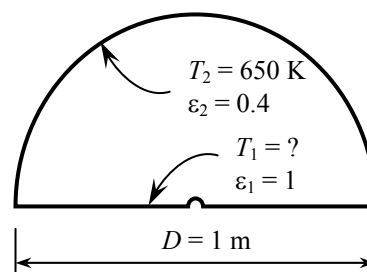
$$A_1 = (1.0 \text{ m})(1.0 \text{ m}) = 1.0 \text{ m}^2$$

$$A_2 = \pi DL / 2 = \pi(1.0 \text{ m})(1 \text{ m}) / 2 = 1.571 \text{ m}^2$$

$$F_{11} + F_{12} = 1 \longrightarrow 0 + F_{12} = 1 \longrightarrow F_{12} = 1 \quad (\text{summation rule})$$

The temperature of the base surface is determined from

$$\begin{aligned} \dot{Q}_{12} &= \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{A_1 F_{12}} + \frac{1 - \varepsilon_2}{A_2 \varepsilon_2}} \\ 1200 \text{ W} &= \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_1^4 - (650 \text{ K})^4]}{\frac{1}{(1.0 \text{ m}^2)(1)} + \frac{1 - 0.4}{(1.571 \text{ m}^2)(0.4)}} \\ T_1 &= \mathbf{684.8 \text{ K}} \end{aligned}$$



15-89 A hemisphere with specified base and dome temperatures and heat transfer rate is considered. The emissivity of the dome is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivity of the base surface is $\varepsilon = 0.55$.

Analysis We consider the base surface to be surface 1, the dome surface to be surface 2. This system is a two-surface enclosure. The surface areas and the view factor are determined as

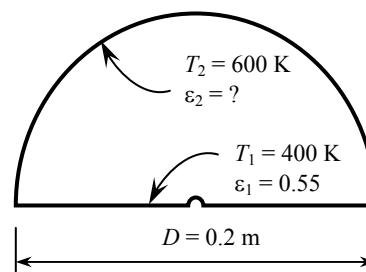
$$A_1 = \pi D^2 / 4 = \pi(0.2 \text{ m})^2 / 4 = 0.0314 \text{ m}^2$$

$$A_2 = \pi D^2 / 2 = \pi(0.2 \text{ m})^2 / 2 = 0.0628 \text{ m}^2$$

$$F_{11} + F_{12} = 1 \longrightarrow 0 + F_{12} = 1 \longrightarrow F_{12} = 1 \quad (\text{summation rule})$$

The emissivity of the dome is determined from

$$\begin{aligned} \dot{Q}_{21} = -\dot{Q}_{12} &= -\frac{\sigma(T_1^4 - T_2^4)}{\frac{1 - \varepsilon_1}{A_1 \varepsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1 - \varepsilon_2}{A_2 \varepsilon_2}} \\ 50 \text{ W} &= -\frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(400 \text{ K})^4 - (600 \text{ K})^4]}{\frac{1 - 0.55}{(0.0314 \text{ m}^2)(0.55)} + \frac{1}{(0.0314 \text{ m}^2)(1)} + \frac{1 - \varepsilon_2}{(0.0628 \text{ m}^2)\varepsilon_2}} \longrightarrow \varepsilon_2 = \mathbf{0.21} \end{aligned}$$



Review Problems

15-90 The variation of emissivity of an opaque surface at a specified temperature with wavelength is given. The average emissivity of the surface and its emissive power are to be determined.

Analysis The average emissivity of the surface can be determined from

$$\varepsilon(T) = \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (f_{\lambda_2} - f_{\lambda_1}) + \varepsilon_3 (1 - f_{\lambda_2})$$

where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$. These functions are determined from Table 15-2 to be

$$\lambda_1 T = (2 \mu\text{m})(1500 \text{ K}) = 3000 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.273232$$

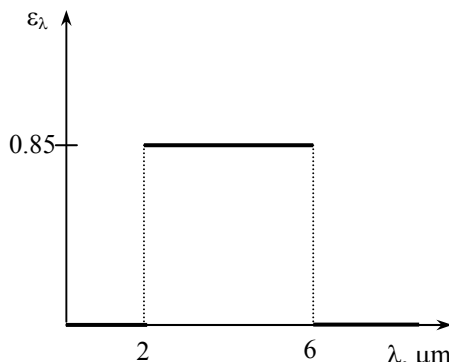
$$\lambda_2 T = (6 \mu\text{m})(1500 \text{ K}) = 9000 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.890029$$

and

$$\varepsilon = (0.0)(0.273232) + (0.85)(0.890029 - 0.273232) + (0.0)(1 - 0.890029) = \mathbf{0.5243}$$

Then the emissive flux of the surface becomes

$$E = \varepsilon \sigma T^4 = (0.5243)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(1500 \text{ K})^4 = \mathbf{150,500 \text{ W/m}^2}$$



15-91 The variation of transmissivity of glass with wavelength is given. The transmissivity of the glass for solar radiation and for light are to be determined.

Analysis For solar radiation, $T = 5800 \text{ K}$. The average transmissivity of the surface can be determined from

$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 (f_{\lambda_2} - f_{\lambda_1}) + \tau_3 (1 - f_{\lambda_2})$$

where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$. These functions are determined from Table 15-2 to be

$$\lambda_1 T = (0.35 \mu\text{m})(5800 \text{ K}) = 2030 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.071852$$

$$\lambda_2 T = (2.5 \mu\text{m})(5800 \text{ K}) = 14,500 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.966440$$

and

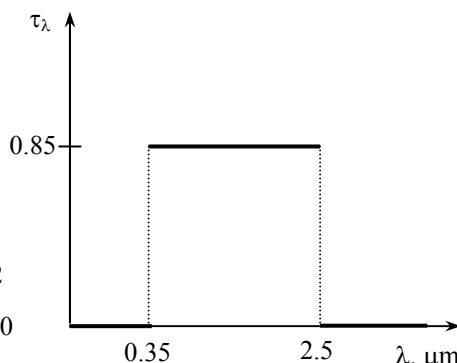
$$\tau = (0.0)(0.071852) + (0.85)(0.966440 - 0.071852) + (0.0)(1 - 0.966440) = \mathbf{0.760}$$

For light, we take $T = 300 \text{ K}$. Repeating the calculations at this temperature we obtain

$$\lambda_1 T = (0.35 \mu\text{m})(300 \text{ K}) = 105 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.00$$

$$\lambda_2 T = (2.5 \mu\text{m})(300 \text{ K}) = 750 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.000012$$

$$\tau = (0.0)(0.00) + (0.85)(0.000012 - 0.00) + (0.0)(1 - 0.000012) = \mathbf{0.00001}$$



15-92 A hole is drilled in a spherical cavity. The maximum rate of radiation energy streaming through the hole is to be determined.

Analysis The maximum rate of radiation energy streaming through the hole is the blackbody radiation, and it can be determined from

$$E = A\sigma T^4 = \pi(0.0025 \text{ m})^2 (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(600 \text{ K})^4 = \mathbf{0.144 \text{ W}}$$

The result would not change for a different diameter of the cavity.

15-93 The variation of absorptivity of a surface with wavelength is given. The average absorptivity of the surface is to be determined for two source temperatures.

Analysis (a) $T = 1000 \text{ K}$. The average absorptivity of the surface can be determined from

$$\begin{aligned} \alpha(T) &= \alpha_1 f_{0-\lambda_1} + \alpha_2 f_{\lambda_1-\lambda_2} + \alpha_3 f_{\lambda_2-\infty} \\ &= \alpha_1 f_{\lambda_1} + \alpha_2 (f_{\lambda_2} - f_{\lambda_1}) + \alpha_3 (1 - f_{\lambda_2}) \end{aligned}$$

where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$, determined from

$$\lambda_1 T = (0.3 \text{ } \mu\text{m})(1000 \text{ K}) = 300 \text{ } \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0$$

$$\lambda_2 T = (1.2 \text{ } \mu\text{m})(1000 \text{ K}) = 1200 \text{ } \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.002134$$

$$f_{0-\lambda_1} = f_{\lambda_1} - f_0 = f_{\lambda_1} \text{ since } f_0 = 0 \text{ and } f_{\lambda_2-\infty} = f_{\infty} - f_{\lambda_2} \text{ since } f_{\infty} = 1.$$

and,

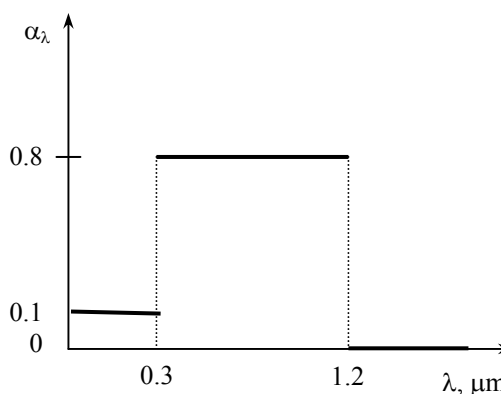
$$\alpha = (0.1)0.0 + (0.8)(0.002134 - 0.0) + (0.0)(1 - 0.002134) = \mathbf{0.0017}$$

(a) $T = 3000 \text{ K}$.

$$\lambda_1 T = (0.3 \text{ } \mu\text{m})(3000 \text{ K}) = 900 \text{ } \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.000169$$

$$\lambda_2 T = (1.2 \text{ } \mu\text{m})(3000 \text{ K}) = 3600 \text{ } \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.403607$$

$$\alpha = (0.1)0.000169 + (0.8)(0.403607 - 0.000169) + (0.0)(1 - 0.403607) = \mathbf{0.323}$$



15-94 The variation of absorptivity of a surface with wavelength is given. The surface receives solar radiation at a specified rate. The solar absorptivity of the surface and the rate of absorption of solar radiation are to be determined.

Analysis For solar radiation, $T = 5800$ K. The solar absorptivity of the surface is

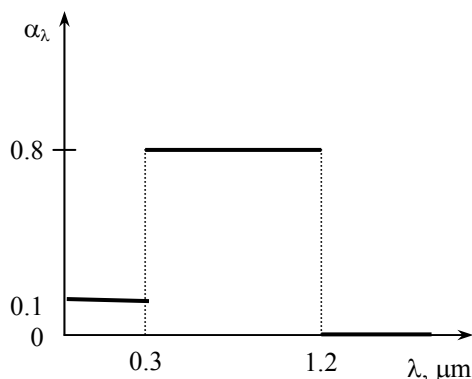
$$\lambda_1 T = (0.3 \mu\text{m})(5800 \text{ K}) = 1740 \mu\text{mK} \rightarrow f_{\lambda_1} = 0.033454$$

$$\lambda_2 T = (1.2 \mu\text{m})(5800 \text{ K}) = 6960 \mu\text{mK} \rightarrow f_{\lambda_2} = 0.805713$$

$$\begin{aligned} \alpha &= (0.1)(0.033454) + (0.8)(0.805713 - 0.033454) \\ &\quad + (0.0)(1 - 0.805713) \\ &= \mathbf{0.621} \end{aligned}$$

The rate of absorption of solar radiation is determined from

$$E_{\text{absorbed}} = \alpha I = 0.621(470 \text{ W/m}^2) = \mathbf{292 \text{ W/m}^2}$$



15-95 The spectral transmissivity of a glass cover used in a solar collector is given. Solar radiation is incident on the collector. The solar flux incident on the absorber plate, the transmissivity of the glass cover for radiation emitted by the absorber plate, and the rate of heat transfer to the cooling water are to be determined.

Analysis (a) For solar radiation, $T = 5800$ K. The average transmissivity of the surface can be determined from

$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 (f_{\lambda_2} - f_{\lambda_1}) + \tau_3 (1 - f_{\lambda_2})$$

where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$. These functions are determined from Table 15-2 to be

$$\lambda_1 T = (0.3 \mu\text{m})(5800 \text{ K}) = 1740 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.033454$$

$$\lambda_2 T = (3 \mu\text{m})(5800 \text{ K}) = 17,400 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.978746$$

and

$$\tau = (0.0)(0.033454) + (0.9)(0.978746 - 0.033454) + (0.0)(1 - 0.978746) = 0.851$$

Since the absorber plate is black, all of the radiation transmitted through the glass cover will be absorbed by the absorber plate and therefore, the solar flux incident on the absorber plate is same as the radiation absorbed by the absorber plate:

$$E_{\text{abs. plate}} = \tau I = 0.851(950 \text{ W/m}^2) = \mathbf{808.5 \text{ W/m}^2}$$

(b) For radiation emitted by the absorber plate, we take $T = 300$ K, and calculate the transmissivity as follows:

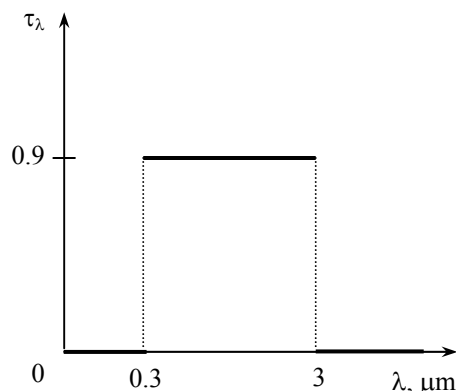
$$\lambda_1 T = (0.3 \mu\text{m})(300 \text{ K}) = 90 \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0$$

$$\lambda_2 T = (3 \mu\text{m})(300 \text{ K}) = 900 \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.000169$$

$$\tau = (0.0)(0.0) + (0.9)(0.000169 - 0.0) + (0.0)(1 - 0.000169) = \mathbf{0.00015}$$

(c) The rate of heat transfer to the cooling water is the difference between the radiation absorbed by the absorber plate and the radiation emitted by the absorber plate, and it is determined from

$$\dot{Q}_{\text{water}} = (\tau_{\text{solar}} - \tau_{\text{room}})I = (0.851 - 0.00015)(950 \text{ W/m}^2) = \mathbf{808.3 \text{ W/m}^2}$$



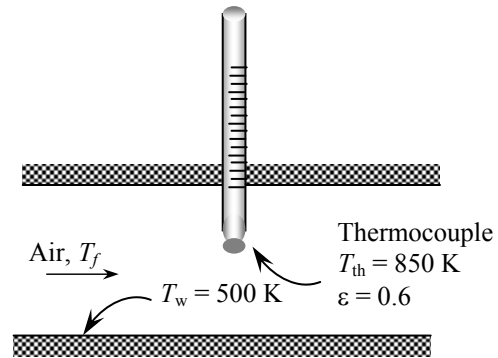
15-96 The temperature of air in a duct is measured by a thermocouple. The radiation effect on the temperature measurement is to be quantified, and the actual air temperature is to be determined.

Assumptions The surfaces are opaque, diffuse, and gray.

Properties The emissivity of thermocouple is given to be $\varepsilon=0.6$.

Analysis The actual temperature of the air can be determined from

$$\begin{aligned}
 T_f &= T_{th} + \frac{\varepsilon_{th} \sigma (T_{th}^4 - T_w^4)}{h} \\
 &= 850 \text{ K} + \frac{(0.6)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(850 \text{ K})^4 - (500 \text{ K})^4]}{60 \text{ W/m}^2 \cdot ^\circ\text{C}} \\
 &= \mathbf{1111 \text{ K}}
 \end{aligned}$$



15-97 Radiation heat transfer occurs between a tube-bank and a wall. The view factors, the net rate of radiation heat transfer, and the temperature of tube surface are to be determined.

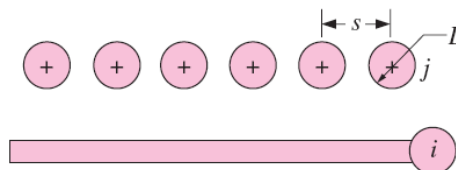
Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 The tube wall thickness and convection from the outer surface are negligible.

Properties The emissivities of the wall and tube bank are given to be $\varepsilon_i = 0.8$ and $\varepsilon_j = 0.9$, respectively.

Analysis (a) We take the wall to be surface i and the tube bank to be surface j . The view factor from surface i to surface j is determined from

$$F_{ij} = 1 - \left[1 - \left(\frac{D}{s} \right)^2 \right]^{0.5} + \left(\frac{D}{s} \right) \left\{ \tan^{-1} \left[\left(\frac{s}{D} \right)^2 - 1 \right]^{0.5} \right\}$$

$$= 1 - \left[1 - \left(\frac{1.5}{3} \right)^2 \right]^{0.5} + \left(\frac{1.5}{3} \right) \left\{ \tan^{-1} \left[\left(\frac{3}{1.5} \right)^2 - 1 \right]^{0.5} \right\} = \mathbf{0.658}$$



The view factor from surface j to surface i is determined from reciprocity relation. Taking s to be the width of the wall

$$A_i F_{ij} = A_j F_{ji} \longrightarrow F_{ji} = \frac{A_i}{A_j} F_{ij} = \frac{sL}{\pi DL} F_{ij} = \frac{s}{\pi D} F_{ij} = \frac{3}{\pi(1.5)} (0.658) = \mathbf{0.419}$$

(b) The net rate of radiation heat transfer between the surfaces can be determined from

$$\dot{q} = \frac{\sigma(T_i^4 - T_j^4)}{\left(\frac{1-\varepsilon_i}{\varepsilon_i} \right) \frac{1}{A_i} + \frac{1}{A_i F_{ij}} + \left(\frac{1-\varepsilon_j}{\varepsilon_j} \right) \frac{1}{A_j}} = \frac{\sigma(T_i^4 - T_j^4)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{ij}} + \left(\frac{1-\varepsilon_j}{\varepsilon_j} \right) \frac{A_i}{A_j}}$$

$$= \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(1173 \text{ K})^4 - (333 \text{ K})^4]}{\frac{1-0.8}{0.8} + \frac{1}{0.658} + \left(\frac{1-0.9}{0.9} \right) \frac{(0.03 \text{ m})}{\pi(0.015 \text{ m})}} = \mathbf{57,900 \text{ W/m}^2}$$

(c) Under steady conditions, the rate of radiation heat transfer from the wall to the tube surface is equal to the rate of convection heat transfer from the tube wall to the fluid. Denoting T_w to be the wall temperature,

$$\dot{q}_{rad} = \dot{q}_{conv}$$

$$\frac{\sigma(T_i^4 - T_w^4)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{ij}} + \left(\frac{1-\varepsilon_j}{\varepsilon_j} \right) \frac{A_i}{A_j}} = h A_j (T_w - T_j)$$

$$\frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(1173 \text{ K})^4 - T_w^4]}{\frac{1-0.8}{0.8} + \frac{1}{0.658} + \left(\frac{1-0.9}{0.9} \right) \frac{(0.03 \text{ m})}{\pi(0.015 \text{ m})}} = (2000 \text{ W/m}^2 \cdot \text{K}) \left[\frac{\pi(0.015 \text{ m})}{0.03 \text{ m}} \right] [T_w - (40 + 273 \text{ K})]$$

Solving this equation by an equation solver such as EES, we obtain

$$T_w = 331.4 \text{ K} = \mathbf{58.4^\circ\text{C}}$$

15-98E A sealed electronic box is placed in a vacuum chamber. The highest temperature at which the surrounding surfaces must be kept if this box is cooled by radiation alone is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered. 4 Heat transfer from the bottom surface of the box is negligible.

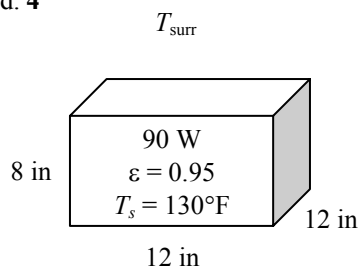
Properties The emissivity of the outer surface of the box is $\varepsilon = 0.95$.

Analysis The total surface area is

$$A_s = 4 \times (8 \times 1/12) + (1 \times 1) = 3.67 \text{ ft}^2$$

Then the temperature of the surrounding surfaces is determined to be

$$\begin{aligned} \dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ (90 \times 3.41214) \text{ Btu/h} &= (0.95)(3.67 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)[(590 \text{ R})^4 - T_{surr}^4] \\ \longrightarrow T_{surr} &= 514 \text{ R} = \mathbf{54^\circ\text{F}} \end{aligned}$$



15-99 A double-walled spherical tank is used to store iced water. The air space between the two walls is evacuated. The rate of heat transfer to the iced water and the amount of ice that melts a 24-h period are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray.

Properties The emissivities of both surfaces are given to be $\varepsilon_1 = \varepsilon_2 = 0.15$.

Analysis (a) Assuming the conduction resistance s of the walls to be negligible, the rate of heat transfer to the iced water in the tank is determined to be

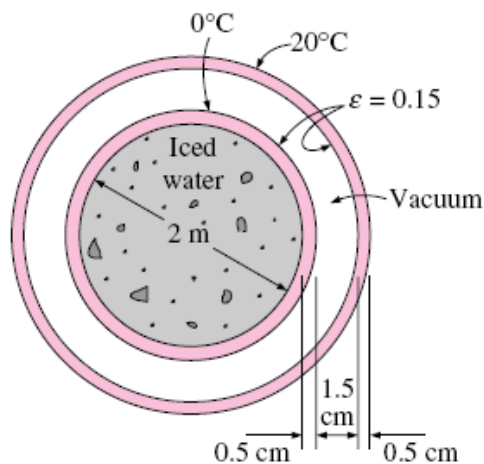
$$\begin{aligned} A_1 &= \pi D_1^2 = \pi (2.01 \text{ m})^2 = 12.69 \text{ m}^2 \\ \dot{Q}_{12} &= \frac{A_1 \sigma (T_2^4 - T_1^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{D_1}{D_2} \right)^2} \\ &= \frac{(12.69 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(20 + 273 \text{ K})^4 - (0 + 273 \text{ K})^4]}{\frac{1}{0.15} + \frac{1 - 0.15}{0.15} \left(\frac{2.01}{2.04} \right)^2} \\ &= \mathbf{107.4 \text{ W}} \end{aligned}$$

(b) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q} \Delta t = (0.1074 \text{ kJ/s})(24 \times 3600 \text{ s}) = 9279 \text{ kJ}$$

The amount of ice that melts during this period then becomes

$$Q = m h_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{9279 \text{ kJ}}{333.7 \text{ kJ/kg}} = \mathbf{27.8 \text{ kg}}$$



15-100 Two concentric spheres which are maintained at uniform temperatures are separated by air at 1 atm pressure. The rate of heat transfer between the two spheres by natural convection and radiation is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Air is an ideal gas with constant properties.

Properties The emissivities of the surfaces are given to be $\varepsilon_1 = \varepsilon_2 = 0.75$. The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (350 + 275)/2 = 312.5$ K = 39.5°C are (Table A-22)

$$k = 0.02658 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.697 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7256$$

$$\beta = \frac{1}{312.5 \text{ K}} = 0.0032 \text{ K}^{-1}$$

Analysis (a) Noting that $D_i = D_1$ and $D_o = D_2$, the characteristic length is

$$L_c = \frac{1}{2}(D_o - D_i) = \frac{1}{2}(0.25 \text{ m} - 0.15 \text{ m}) = 0.05 \text{ m}$$

Then

$$Ra = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003200 \text{ K}^{-1})(350 - 275 \text{ K})(0.05 \text{ m})^3}{(1.697 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7256) = 7.415 \times 10^5$$

The effective thermal conductivity is

$$F_{\text{sph}} = \frac{L_c}{(D_i D_o)^4 (D_i^{-7/5} + D_o^{-7/5})^5} = \frac{0.05 \text{ m}}{[(0.15 \text{ m})(0.25 \text{ m})]^4 [(0.15 \text{ m})^{-7/5} + (0.25 \text{ m})^{-7/5}]^5} = 0.005900$$

$$k_{\text{eff}} = 0.74k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{sph}} Ra)^{1/4}$$

$$= 0.74(0.02658 \text{ W/m} \cdot ^\circ\text{C}) \left(\frac{0.7256}{0.861 + 0.7256} \right)^{1/4} [(0.00590)(7.415 \times 10^5)]^{1/4}$$

$$= 0.1315 \text{ W/m} \cdot ^\circ\text{C}$$

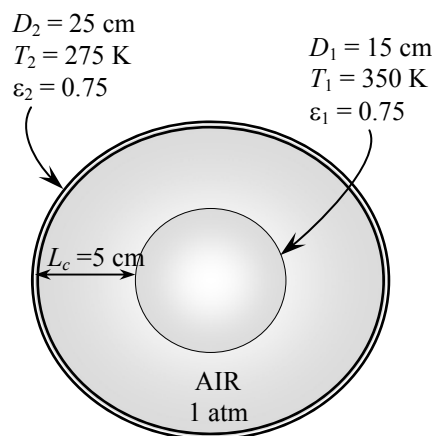
Then the rate of heat transfer between the spheres becomes

$$\dot{Q} = k_{\text{eff}} \pi \left(\frac{D_i D_o}{L_c} \right) (T_i - T_o) = (0.1315 \text{ W/m} \cdot ^\circ\text{C}) \pi \left[\frac{(0.15 \text{ m})(0.25 \text{ m})}{(0.05 \text{ m})} \right] (350 - 275) \text{ K} = \mathbf{23.2 \text{ W}}$$

(b) The rate of heat transfer by radiation is determined from

$$A_1 = \pi D_1^2 = \pi (0.15 \text{ m})^2 = 0.0707 \text{ m}^2$$

$$\dot{Q}_{12} = \frac{A_1 \sigma (T_2^4 - T_1^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{D_1}{D_2} \right)^2} = \frac{(0.0707 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(350 \text{ K})^4 - (275 \text{ K})^4]}{\frac{1}{0.75} + \frac{1 - 0.75}{0.75} \left(\frac{0.15}{0.25} \right)^2} = \mathbf{25.6 \text{ W}}$$



15-101 A solar collector is considered. The absorber plate and the glass cover are maintained at uniform temperatures, and are separated by air. The rate of heat loss from the absorber plate by natural convection and radiation is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Air is an ideal gas with constant properties.

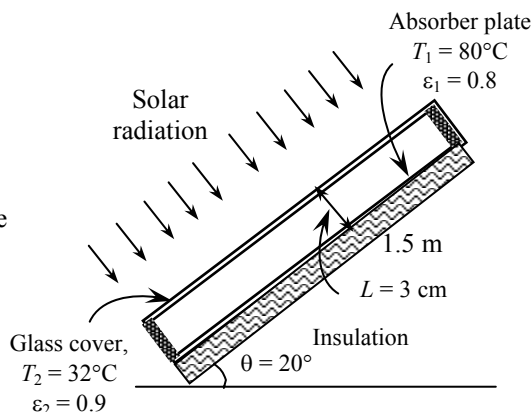
Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.9$ for glass and $\varepsilon_2 = 0.8$ for the absorber plate. The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (80 + 32)/2 = 56^\circ\text{C}$ are (Table A-22)

$$k = 0.02779 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.857 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7212$$

$$\beta = \frac{1}{T_f} = \frac{1}{(56 + 273)\text{K}} = 0.003040 \text{ K}^{-1}$$



Analysis For $\theta = 0^\circ$, we have horizontal rectangular enclosure. The characteristic length in this case is the distance between the two glasses $L_c = L = 0.03 \text{ m}$. Then,

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00304 \text{ K}^{-1})(80 - 32 \text{ K})(0.03 \text{ m})^3}{(1.857 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7212) = 8.083 \times 10^4$$

$$A_s = H \times W = (1.5 \text{ m})(3 \text{ m}) = 4.5 \text{ m}^2$$

$$\begin{aligned} \text{Nu} &= 1 + 1.44 \left[1 - \frac{1708}{\text{Ra} \cos \theta} \right]^+ \left[1 - \frac{1708(\sin 1.8\theta)^{1.6}}{\text{Ra} \cos \theta} \right] + \left[\frac{(\text{Ra} \cos \theta)^{1/3}}{18} - 1 \right]^+ \\ &= 1 + 1.44 \left[1 - \frac{1708}{(8.083 \times 10^4) \cos(20^\circ)} \right]^+ \left[1 - \frac{1708[\sin(1.8 \times 20)]^{1.6}}{(8.083 \times 10^4) \cos(20^\circ)} \right] + \left[\frac{[(8.083 \times 10^4) \cos(20^\circ)]^{1/3}}{18} - 1 \right]^+ \\ &= 3.747 \end{aligned}$$

$$\dot{Q} = k\text{Nu}A_s \frac{T_1 - T_2}{L} = (0.02779 \text{ W/m}\cdot^\circ\text{C})(3.747)(4.5 \text{ m}^2) \frac{(80 - 32)^\circ\text{C}}{0.03 \text{ m}} = \mathbf{750 \text{ W}}$$

Neglecting the end effects, the rate of heat transfer by radiation is determined from

$$\dot{Q}_{\text{rad}} = \frac{A_s \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} = \frac{(4.5 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(80 + 273 \text{ K})^4 - (32 + 273 \text{ K})^4]}{\frac{1}{0.8} + \frac{1}{0.9} - 1} = \mathbf{1289 \text{ W}}$$

Discussion The rates of heat loss by natural convection for the horizontal and vertical cases would be as follows (Note that the Ra number remains the same):

Horizontal:

$$\text{Nu} = 1 + 1.44 \left[1 - \frac{1708}{\text{Ra}} \right]^+ + \left[\frac{\text{Ra}^{1/3}}{18} - 1 \right]^+ = 1 + 1.44 \left[1 - \frac{1708}{8.083 \times 10^4} \right]^+ + \left[\frac{(8.083 \times 10^4)^{1/3}}{18} - 1 \right]^+ = 3.812$$

$$\dot{Q} = k\text{Nu}A_s \frac{T_1 - T_2}{L} = (0.02779 \text{ W/m}\cdot^\circ\text{C})(3.812)(6 \text{ m}^2) \frac{(80 - 32)^\circ\text{C}}{0.03 \text{ m}} = \mathbf{1017 \text{ W}}$$

Vertical:

$$\text{Nu} = 0.42 \text{Ra}^{1/4} \text{Pr}^{0.012} \left(\frac{H}{L} \right)^{-0.3} = 0.42(8.083 \times 10^4)^{1/4} (0.7212)^{0.012} \left(\frac{2 \text{ m}}{0.03 \text{ m}} \right)^{-0.3} = 2.001$$

$$\dot{Q} = k\text{Nu}A_s \frac{T_1 - T_2}{L} = (0.02779 \text{ W/m}\cdot^\circ\text{C})(2.001)(6 \text{ m}^2) \frac{(80 - 32)^\circ\text{C}}{0.03 \text{ m}} = \mathbf{534 \text{ W}}$$

15-102E CD EES The circulating pump of a solar collector that consists of a horizontal tube and its glass cover fails. The equilibrium temperature of the tube is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The tube and its cover are isothermal. 3 Air is an ideal gas. 4 The surfaces are opaque, diffuse, and gray for infrared radiation. 5 The glass cover is transparent to solar radiation.

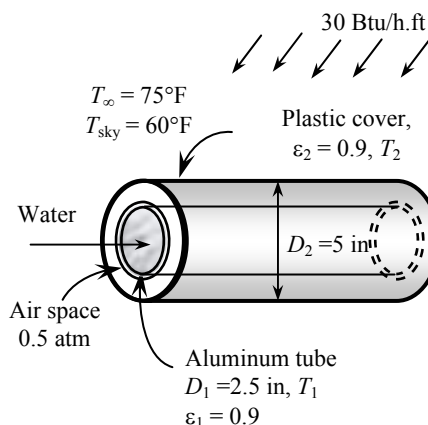
Properties The properties of air should be evaluated at the average temperature. But we do not know the exit temperature of the air in the duct, and thus we cannot determine the bulk fluid and glass cover temperatures at this point, and thus we cannot evaluate the average temperatures. Therefore, we will assume the glass temperature to be 85°F, and use properties at an anticipated average temperature of $(75+85)/2 = 80^\circ\text{F}$ (Table A-22E),

$$k = 0.01481 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F}$$

$$\nu = 1.697 \times 10^{-4} \text{ ft}^2 / \text{s}$$

$$\text{Pr} = 0.7290$$

$$\beta = \frac{1}{T_{\text{ave}}} = \frac{1}{540 \text{ R}}$$



Analysis We have a horizontal cylindrical enclosure filled with air at 0.5 atm pressure. The problem involves heat transfer from the aluminum tube to the glass cover and from the outer surface of the glass cover to the surrounding ambient air. When steady operation is reached, these two heat transfer rates must equal the rate of heat gain. That is,

$$\dot{Q}_{\text{tube-glass}} = \dot{Q}_{\text{glass-ambient}} = \dot{Q}_{\text{solar gain}} = 30 \text{ Btu/h} \quad (\text{per foot of tube})$$

The heat transfer surface area of the glass cover is

$$A_o = A_{\text{glass}} = (\pi D_o W) = \pi(5/12 \text{ ft})(1 \text{ ft}) = 1.309 \text{ ft}^2 \quad (\text{per foot of tube})$$

To determine the Rayleigh number, we need to know the surface temperature of the glass, which is not available. Therefore, solution will require a trial-and-error approach. Assuming the glass cover temperature to be 85°F, the Rayleigh number, the Nusselt number, the convection heat transfer coefficient, and the rate of natural convection heat transfer from the glass cover to the ambient air are determined to be

$$\begin{aligned} \text{Ra}_{D_o} &= \frac{g\beta(T_o - T_\infty)D_o^3}{\nu^2} \text{Pr} \\ &= \frac{(32.2 \text{ ft/s}^2)[1/(540 \text{ R})](85 - 75 \text{ R})(5/12 \text{ ft})^3}{(1.697 \times 10^{-4} \text{ ft}^2/\text{s})^2} (0.7290) = 1.092 \times 10^6 \\ \text{Nu} &= \left\{ 0.6 + \frac{0.387 \text{ Ra}_D^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387(1.092 \times 10^6)^{1/6}}{\left[1 + (0.559 / 0.7290)^{9/16} \right]^{8/27}} \right\}^2 \\ &= 14.95 \end{aligned}$$

$$h_o = \frac{k}{D_o} \text{Nu} = \frac{0.01481 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F}}{5/12 \text{ ft}} (14.95) = 0.5315 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

$$\dot{Q}_{o,\text{conv}} = h_o A_o (T_o - T_\infty) = (0.5315 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F})(1.309 \text{ ft}^2)(85 - 75)^\circ\text{F} = 6.96 \text{ Btu/h}$$

Also,

$$\begin{aligned} \dot{Q}_{o,\text{rad}} &= \varepsilon_o \sigma A_o (T_o^4 - T_{\text{sky}}^4) \\ &= (0.9)(0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)(1.309 \text{ ft}^2) \left[(545 \text{ R})^4 - (520 \text{ R})^4 \right] \\ &= 30.5 \text{ Btu/h} \end{aligned}$$

Then the total rate of heat loss from the glass cover becomes

$$\dot{Q}_{o,\text{total}} = \dot{Q}_{o,\text{conv}} + \dot{Q}_{o,\text{rad}} = 7.0 + 30.5 = 37.5 \text{ Btu/h}$$

which is more than 30 Btu/h. Therefore, the assumed temperature of 85°F for the glass cover is high. Repeating the calculations with lower temperatures (including the evaluation of properties), the glass cover temperature corresponding to 30 Btu/h is determined to be 81.5°F.

The temperature of the aluminum tube is determined in a similar manner using the natural convection and radiation relations for two horizontal concentric cylinders. The characteristic length in this case is the distance between the two cylinders, which is

$$L_c = (D_o - D_i) / 2 = (5 - 2.5) / 2 = 1.25 \text{ in} = 1.25/12 \text{ ft}$$

Also,

$$A_i = A_{\text{tube}} = (\pi D_i W) = \pi(2.5/12 \text{ ft})(1 \text{ ft}) = 0.6545 \text{ ft}^2 \text{ (per foot of tube)}$$

We start the calculations by assuming the tube temperature to be 118.5°F, and thus an average temperature of $(81.5 + 118.5)/2 = 100^\circ\text{F} = 560 \text{ R}$. Using properties at 100°F,

$$\text{Ra}_L = \frac{g\beta(T_i - T_o)L^3}{\nu^2} \text{Pr} = \frac{(32.2 \text{ ft/s}^2)[1/(560 \text{ R})](118.5 - 81.5 \text{ R})(1.25/12 \text{ ft})^3}{[(1.809 \times 10^{-4} \text{ ft}^2/\text{s})/0.5]^2} (0.726) = 1.334 \times 10^4$$

The effective thermal conductivity is

$$F_{\text{cyc}} = \frac{[\ln(D_o/D_i)]^4}{L_c^3(D_i^{-3/5} + D_o^{-3/5})^5} = \frac{[\ln(5/2.5)]^4}{(1.25/12 \text{ ft})^3[(2.5/12 \text{ ft})^{-3/5} + (5/12 \text{ ft})^{-3/5}]^5} = 0.1466$$

$$\begin{aligned} k_{\text{eff}} &= 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyc}} \text{Ra}_L)^{1/4} \\ &= 0.386(0.01529 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F}) \left(\frac{0.726}{0.861 + 0.726} \right) (0.1466 \times 1.334 \times 10^4)^{1/4} \\ &= 0.03227 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} \end{aligned}$$

Then the rate of heat transfer between the cylinders by convection becomes

$$\dot{Q}_{i,\text{conv}} = \frac{2\pi k_{\text{eff}}}{\ln(D_o/D_i)} (T_i - T_o) = \frac{2\pi(0.03227 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F})}{\ln(5/2.5)} (118.5 - 81.5)^\circ\text{F} = 10.8 \text{ Btu/h}$$

Also,

$$\begin{aligned} \dot{Q}_{i,\text{rad}} &= \frac{\sigma A_i (T_i^4 - T_o^4)}{\frac{1}{\varepsilon_i} + \frac{1 - \varepsilon_o}{\varepsilon_o} \left(\frac{D_i}{D_o} \right)} \\ &= \frac{(0.1714 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4)(0.6545 \text{ ft}^2)[(578.5 \text{ R})^4 - (541.5 \text{ R})^4]}{\frac{1}{0.9} + \frac{1 - 0.9}{0.9} \left(\frac{2.5 \text{ in}}{5 \text{ in}} \right)} = 25.0 \text{ Btu/h} \end{aligned}$$

Then the total rate of heat loss from the glass cover becomes

$$\dot{Q}_{i,\text{total}} = \dot{Q}_{i,\text{conv}} + \dot{Q}_{i,\text{rad}} = 10.8 + 25.0 = 35.8 \text{ Btu/h}$$

which is more than 30 Btu/h. Therefore, the assumed temperature of 118.5°F for the tube is high. By trying other values, the tube temperature corresponding to 30 Btu/h is determined to be **113°F**. Therefore, the tube will reach an equilibrium temperature of 113°F when the pump fails.

15-103 A double-pane window consists of two sheets of glass separated by an air space. The rates of heat transfer through the window by natural convection and radiation are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Air is an ideal gas with constant specific heats. 4 Heat transfer through the window is one-dimensional and the edge effects are negligible.

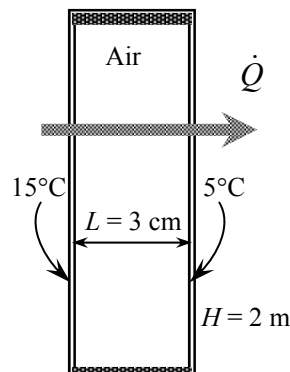
Properties The emissivities of glass surfaces are given to be $\epsilon_1 = \epsilon_2 = 0.9$. The properties of air at 0.3 atm and the average temperature of $(T_1 + T_2)/2 = (15 + 5)/2 = 10^\circ\text{C}$ are (Table A-22)

$$k = 0.02439 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = \nu_{1\text{atm}} / 0.3 = 1.426 \times 10^{-5} / 0.3 = 4.753 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7336$$

$$\beta = \frac{1}{(10 + 273) \text{ K}} = 0.003534 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the distance between the glasses, $L_c = L = 0.03 \text{ m}$

$$Ra = \frac{g\beta(T_1 - T_2)L^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003534 \text{ K}^{-1})(15 - 5)\text{K}(0.03 \text{ m})^3}{(4.753 \times 10^{-5} \text{ m}^2/\text{s})^2}(0.7336) = 3040$$

$$Nu = 0.197 Ra^{1/4} \left(\frac{H}{L} \right)^{-1/9} = 0.197(3040)^{1/4} \left(\frac{2}{0.05} \right)^{-1/9} = 0.971$$

Note that heat transfer through the air space is less than that by pure conduction as a result of partial evacuation of the space. Then the rate of heat transfer through the air space becomes

$$A_s = (2 \text{ m})(5 \text{ m}) = 10 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = kNuA_s \frac{T_1 - T_2}{L} = (0.02439 \text{ W/m}\cdot^\circ\text{C})(0.971)(10 \text{ m}^2) \frac{(15 - 5)^\circ\text{C}}{0.03 \text{ m}} = \mathbf{78.9 \text{ W}}$$

The rate of heat transfer by radiation is determined from

$$\dot{Q}_{\text{rad}} = \frac{A_s \sigma (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} = \frac{(10 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(15 + 273 \text{ K})^4 - (5 + 273 \text{ K})^4]}{\frac{1}{0.9} + \frac{1}{0.9} - 1} = \mathbf{421 \text{ W}}$$

Then the rate of total heat transfer becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 79 + 421 = \mathbf{500 \text{ W}}$$

Discussion Note that heat transfer through the window is mostly by radiation.

15-104 CD EES A simple solar collector is built by placing a clear plastic tube around a garden hose. The rate of heat loss from the water in the hose by natural convection and radiation is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Air is an ideal gas with constant specific heats.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = \varepsilon_2 = 0.9$. The properties of air are at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (40 + 25)/2 = 32.5^\circ\text{C}$ are (Table A-22)

$$k = 0.02607 \text{ W/m}\cdot^\circ\text{C}$$

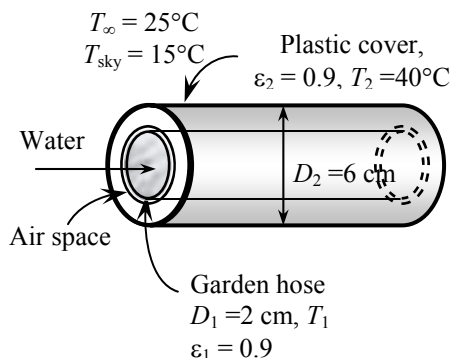
$$\nu = 1.632 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7275$$

$$\beta = \frac{1}{(32.5 + 273) \text{ K}} = 0.003273 \text{ K}^{-1}$$

Analysis Under steady conditions, the heat transfer rate from the water in the hose equals to the rate of heat loss from the clear plastic tube to the surroundings by natural convection and radiation. The characteristic length in this case is the diameter of the plastic tube,

$$L_c = D_{\text{plastic}} = D_2 = 0.06 \text{ m}.$$



$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D_2^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003273 \text{ K}^{-1})(40 - 25)\text{K}(0.06 \text{ m})^3}{(1.632 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7275) = 2.842 \times 10^5$$

$$\text{Nu} = \left\{ 0.6 + \frac{0.387 \text{ Ra}_D^{1/6}}{\left[1 + (0.559 / \text{Pr})^{9/16} \right]^{8/27}} \right\}^2 = \left\{ 0.6 + \frac{0.387 (2.842 \times 10^5)^{1/6}}{\left[1 + (0.559 / 0.7241)^{9/16} \right]^{8/27}} \right\}^2 = 10.30$$

$$h = \frac{k}{D_2} \text{Nu} = \frac{0.02607 \text{ W/m}\cdot^\circ\text{C}}{0.06 \text{ m}} (10.30) = 4.475 \text{ W/m}^2\cdot^\circ\text{C}$$

$$A_{\text{plastic}} = A_2 = \pi D_2 L = \pi (0.06 \text{ m}) (1 \text{ m}) = 0.1885 \text{ m}^2$$

Then the rate of heat transfer from the outer surface by natural convection becomes

$$\dot{Q}_{\text{conv}} = hA_2(T_s - T_\infty) = (4.475 \text{ W/m}^2\cdot^\circ\text{C})(0.1885 \text{ m}^2)(40 - 25)^\circ\text{C} = \mathbf{12.7 \text{ W}}$$

The rate of heat transfer by radiation from the outer surface is determined from

$$\begin{aligned} \dot{Q}_{\text{rad}} &= \varepsilon A_2 \sigma (T_s^4 - T_{\text{sky}}^4) \\ &= (0.90)(0.1885 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(40 + 273 \text{ K})^4 - (15 + 273 \text{ K})^4] \\ &= \mathbf{26.1 \text{ W}} \end{aligned}$$

Finally,

$$\dot{Q}_{\text{total, loss}} = 12.7 + 26.1 = 38.8 \text{ W}$$

Discussion Note that heat transfer is mostly by radiation.

15-105 A solar collector consists of a horizontal copper tube enclosed in a concentric thin glass tube. The annular space between the copper and the glass tubes is filled with air at 1 atm. The rate of heat loss from the collector by natural convection and radiation is to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Air is an ideal gas with constant specific heats.

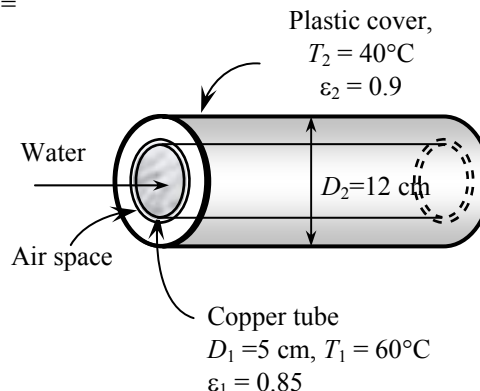
Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.85$ for the tube surface and $\varepsilon_2 = 0.9$ for glass cover. The properties of air at 1 atm and the average temperature of $(T_1 + T_2)/2 = (60 + 40)/2 = 50^\circ\text{C}$ are (Table A-22)

$$k = 0.02735 \text{ W/m}\cdot^\circ\text{C}$$

$$\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7228$$

$$\beta = \frac{1}{(50 + 273) \text{ K}} = 0.003096 \text{ K}^{-1}$$



Analysis The characteristic length in this case is

$$L_c = \frac{1}{2}(D_2 - D_1) = \frac{1}{2}(0.12 \text{ m} - 0.05 \text{ m}) = 0.07 \text{ m}$$

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} \text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003096 \text{ K}^{-1})(60 - 40)\text{K}(0.035 \text{ m})^3}{(1.798 \times 10^{-5} \text{ m}^2/\text{s})^2} (0.7228) = 5.823 \times 10^4$$

The effective thermal conductivity is

$$F_{\text{cyl}} = \frac{[\ln(D_o/D_i)]^4}{L_c^3(D_i^{-3/5} + D_o^{-3/5})^5} = \frac{[\ln(0.12/0.05)]^4}{(0.035 \text{ m})^3[(0.05 \text{ m})^{-3/5} + (0.12 \text{ m})^{-3/5}]^5} = 0.1678$$

$$k_{\text{eff}} = 0.386k \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} (F_{\text{cyl}} \text{Ra})^{1/4}$$

$$= 0.386(0.02735 \text{ W/m}\cdot^\circ\text{C}) \left(\frac{0.7228}{0.861 + 0.7228} \right)^{1/4} [(0.1678)(5.823 \times 10^4)]^{1/4} = 0.08626 \text{ W/m}\cdot^\circ\text{C}$$

Then the rate of heat transfer between the cylinders becomes

$$\dot{Q}_{\text{conv}} = \frac{2\pi k_{\text{eff}}}{\ln(D_o/D_i)} (T_i - T_o) = \frac{2\pi(0.08626 \text{ W/m}\cdot^\circ\text{C})}{\ln(0.12/0.05)} (60 - 40)^\circ\text{C} = \mathbf{12.4 \text{ W}}$$

The rate of heat transfer by radiation is determined from

$$\dot{Q}_{\text{rad}} = \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{D_1}{D_2} \right)} = \frac{[\pi(0.05 \text{ m})(1 \text{ m})](5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(60 + 273 \text{ K})^4 - (40 + 273 \text{ K})^4]}{\frac{1}{0.85} + \frac{1 - 0.9}{0.9} \left(\frac{5}{12} \right)}$$

$$= \mathbf{19.7 \text{ W}}$$

Finally,

$$\dot{Q}_{\text{total, loss}} = 12.4 + 19.7 = 32.1 \text{ W} \quad (\text{per m length})$$

15-106 A cylindrical furnace with specified top and bottom surface temperatures and specified heat transfer rate at the bottom surface is considered. The temperature of the side surface and the net rates of heat transfer between the top and the bottom surfaces, and between the bottom and the side surfaces are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of the top, bottom, and side surfaces are 0.70, 0.50, and 0.40, respectively.

Analysis We consider the top surface to be surface 1, the bottom surface to be surface 2, and the side surface to be surface 3. This system is a three-surface enclosure. The view factor from surface 1 to surface 2 is determined from

$$\left. \begin{aligned} \frac{L}{r} &= \frac{1.2}{0.6} = 2 \\ \frac{r}{L} &= \frac{0.6}{1.2} = 0.5 \end{aligned} \right\} F_{12} = 0.17 \text{ (Fig. 15-7)} \quad h = 1.2 \text{ m}$$

The surface areas are

$$A_1 = A_2 = \pi D^2 / 4 = \pi (1.2 \text{ m})^2 / 4 = 1.131 \text{ m}^2$$

$$A_3 = \pi DL = \pi (1.2 \text{ m})(1.2 \text{ m}) = 4.524 \text{ m}^2$$

Then other view factors are determined to be

$$F_{12} = F_{21} = 0.17$$

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow 0 + 0.17 + F_{13} = 1 \longrightarrow F_{13} = 0.83 \text{ (summation rule), } F_{23} = F_{13} = 0.83$$

$$A_1 F_{13} = A_3 F_{31} \longrightarrow (1.131)(0.83) = (4.524) F_{31} \longrightarrow F_{31} = 0.21 \text{ (reciprocity rule), } F_{32} = F_{31} = 0.21$$

We now apply Eq. 15-35 to each surface

$$\sigma T_1^4 = J_1 + \frac{1 - \varepsilon_1}{\varepsilon_1} [F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)]$$

Surface 1:

$$(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(500 \text{ K})^4 = J_1 + \frac{1 - 0.70}{0.70} [0.17(J_1 - J_2) + 0.83(J_1 - J_3)]$$

$$\sigma T_2^4 = J_2 + \frac{1 - \varepsilon_2}{\varepsilon_2} [F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)]$$

Surface 2:

$$(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(500 \text{ K})^4 = J_2 + \frac{1 - 0.50}{0.50} [0.17(J_2 - J_1) + 0.83(J_2 - J_3)]$$

$$\sigma T_3^4 = J_3 + \frac{1 - \varepsilon_3}{\varepsilon_3} [F_{31}(J_3 - J_1) + F_{32}(J_3 - J_2)]$$

Surface 3:

$$(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) T_3^4 = J_3 + \frac{1 - 0.40}{0.40} [0.21(J_3 - J_1) + 0.21(J_3 - J_2)]$$

We now apply Eq. 15-34 to surface 2

$$\dot{Q}_2 = A_2 [F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)] = (1.131 \text{ m}^2) [0.17(J_2 - J_1) + 0.83(J_2 - J_3)]$$

Solving the above four equations, we find

$$T_3 = 631 \text{ K}, \quad J_1 = 4974 \text{ W/m}^2, \quad J_2 = 8883 \text{ W/m}^2, \quad J_3 = 8193 \text{ W/m}^2$$

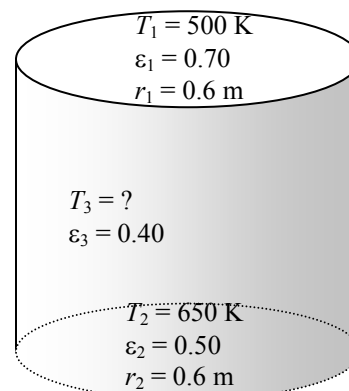
The rate of heat transfer between the bottom and the top surface is

$$\dot{Q}_{21} = A_2 F_{21} (J_2 - J_1) = (1.131 \text{ m}^2)(0.17)(8883 - 4974) \text{ W/m}^2 = 751.6 \text{ W}$$

The rate of heat transfer between the bottom and the side surface is

$$\dot{Q}_{23} = A_2 F_{23} (J_2 - J_3) = (1.131 \text{ m}^2)(0.83)(8883 - 8193) \text{ W/m}^2 = 648.0 \text{ W}$$

Discussion The sum of these two heat transfer rates are $751.6 + 644 = 1395.6 \text{ W}$, which is practically equal to 1400 W heat supply rate from surface 2. This must be satisfied to maintain the surfaces at the specified temperatures under steady operation. Note that the difference is due to round-off error.



15-107 A cylindrical furnace with specified top and bottom surface temperatures and specified heat transfer rate at the bottom surface is considered. The emissivity of the top surface and the net rates of heat transfer between the top and the bottom surfaces, and between the bottom and the side surfaces are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivity of the bottom surface is 0.90.

Analysis We consider the top surface to be surface 1, the base surface to be surface 2, and the side surface to be surface 3. This system is a three-surface enclosure. The view factor from the base to the top surface of the cube is from Fig. 15-5 $F_{12} = 0.2$. The view factor from the base or the top to the side surfaces is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1 - F_{12} = 1 - 0.2 = 0.8$$

since the base surface is flat and thus $F_{11} = 0$. Other view factors are

$$F_{21} = F_{12} = 0.20, \quad F_{23} = F_{13} = 0.80, \quad F_{31} = F_{32} = 0.20$$

We now apply Eq. 9-35 to each surface

$$\sigma T_1^4 = J_1 + \frac{1 - \varepsilon_1}{\varepsilon_1} [F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)]$$

Surface 1:

$$(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(700 \text{ K})^4 = J_1 + \frac{1 - \varepsilon_1}{\varepsilon_1} [0.20(J_1 - J_2) + 0.80(J_1 - J_3)]$$

$$\sigma T_2^4 = J_2 + \frac{1 - \varepsilon_2}{\varepsilon_2} [F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)]$$

Surface 2:

$$(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(950 \text{ K})^4 = J_2 + \frac{1 - 0.90}{0.90} [0.20(J_2 - J_1) + 0.80(J_2 - J_3)]$$

$$\sigma T_3^4 = J_3$$

Surface 3:

$$(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)(450 \text{ K})^4 = J_3$$

We now apply Eq. 9-34 to surface 2

$$\dot{Q}_2 = A_2 [F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)] = (9 \text{ m}^2) [0.20(J_2 - J_1) + 0.80(J_2 - J_3)]$$

Solving the above four equations, we find

$$\varepsilon_1 = \mathbf{0.44}, \quad J_1 = 11,736 \text{ W/m}^2, \quad J_2 = 41,985 \text{ W/m}^2, \quad J_3 = 2325 \text{ W/m}^2$$

The rate of heat transfer between the bottom and the top surface is

$$A_1 = A_2 = (3 \text{ m})^2 = 9 \text{ m}^2$$

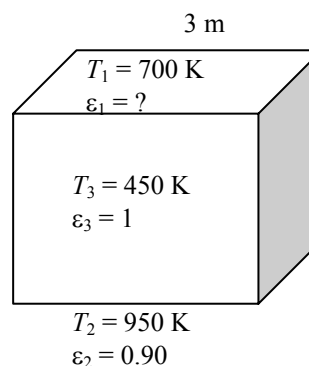
$$\dot{Q}_{21} = A_2 F_{21} (J_2 - J_1) = (9 \text{ m}^2)(0.20)(41,985 - 11,736) \text{ W/m}^2 = \mathbf{54.4 \text{ kW}}$$

The rate of heat transfer between the bottom and the side surface is

$$A_3 = 4A_1 = 4(9 \text{ m}^2) = 36 \text{ m}^2$$

$$\dot{Q}_{23} = A_2 F_{23} (J_2 - J_3) = (9 \text{ m}^2)(0.8)(41,985 - 2325) \text{ W/m}^2 = \mathbf{285.6 \text{ kW}}$$

Discussion The sum of these two heat transfer rates are $54.4 + 285.6 = 340 \text{ kW}$, which is equal to 340 kW heat supply rate from surface 2.



15-108 Radiation heat transfer occurs between two square parallel plates. The view factors, the rate of radiation heat transfer and the temperature of a third plate to be inserted are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of plate a, b, and c are given to be $\varepsilon_a = 0.8$, $\varepsilon_b = 0.4$, and $\varepsilon_c = 0.1$, respectively.

Analysis (a) The view factor from surface a to surface b is determined as follows

$$A = \frac{a}{L} = \frac{20}{40} = 0.5, \quad B = \frac{b}{L} = \frac{60}{40} = 1.5$$

$$F_{ab} = \frac{1}{2A} \left\{ \left[(B+A)^2 + 4 \right]^{0.5} - \left[(B-A)^2 + 4 \right]^{0.5} \right\} = \frac{1}{2(0.5)} \left\{ \left[(1.5+0.5)^2 + 4 \right]^{0.5} - \left[(1.5-0.5)^2 + 4 \right]^{0.5} \right\} = \mathbf{0.592}$$

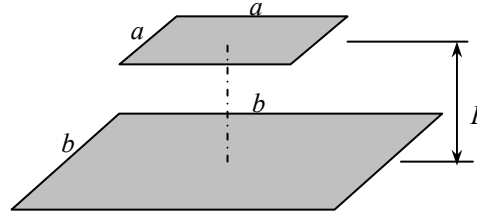
The view factor from surface b to surface a is determined from reciprocity relation:

$$A_a = (0.2 \text{ m})(0.2 \text{ m}) = 0.04 \text{ m}^2$$

$$A_b = (0.6 \text{ m})(0.6 \text{ m}) = 0.36 \text{ m}^2$$

$$A_a F_{ab} = A_b F_{ba}$$

$$(0.04)(0.592) = (0.36)F_{ba} \longrightarrow F_{ba} = \mathbf{0.0658}$$



(b) The net rate of radiation heat transfer between the surfaces can be determined from

$$\dot{Q}_{ab} = \frac{\sigma(T_a^4 - T_b^4)}{\frac{1-\varepsilon_a}{A_a\varepsilon_a} + \frac{1}{A_aF_{ab}} + \frac{1-\varepsilon_b}{A_b\varepsilon_b}} = \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) [(1073 \text{ K})^4 - (473 \text{ K})^4]}{\frac{1-0.8}{(0.04 \text{ m}^2)(0.8)} + \frac{1}{(0.04 \text{ m}^2)(0.592)} + \frac{1-0.4}{(0.36 \text{ m}^2)(0.4)}} = \mathbf{1374 \text{ W}}$$

(c) In this case we have

$$A = \frac{a}{L} = \frac{0.2 \text{ m}}{0.2 \text{ m}} = 1, \quad C = \frac{c}{L} = \frac{2.0 \text{ m}}{0.2 \text{ m}} = 10$$

$$F_{ac} = \frac{1}{2A} \left\{ \left[(C+A)^2 + 4 \right]^{0.5} - \left[(C-A)^2 + 4 \right]^{0.5} \right\} = \frac{1}{2(0.5)} \left\{ \left[(10+0.5)^2 + 4 \right]^{0.5} - \left[(10-0.5)^2 + 4 \right]^{0.5} \right\} = 0.981$$

$$B = \frac{b}{L} = \frac{0.6 \text{ m}}{0.2 \text{ m}} = 3, \quad C = \frac{c}{L} = \frac{2.0 \text{ m}}{0.2 \text{ m}} = 10$$

$$F_{bc} = \frac{1}{2A} \left\{ \left[(C+B)^2 + 4 \right]^{0.5} - \left[(C-B)^2 + 4 \right]^{0.5} \right\}$$

$$= \frac{1}{2(3)} \left\{ \left[(10+3)^2 + 4 \right]^{0.5} - \left[(10-3)^2 + 4 \right]^{0.5} \right\} = 0.979$$

$$A_b F_{bc} = A_c F_{cb}$$

$$(0.36)(0.979) = (4.0)F_{cb} \longrightarrow F_{ba} = 0.0881$$

An energy balance gives

$$\dot{Q}_{ac} = \dot{Q}_{cb}$$

$$\frac{\sigma(T_a^4 - T_c^4)}{\frac{1-\varepsilon_a}{A_a\varepsilon_a} + \frac{1}{A_aF_{ac}} + \frac{1-\varepsilon_c}{A_c\varepsilon_c}} = \frac{\sigma(T_c^4 - T_b^4)}{\frac{1-\varepsilon_c}{A_c\varepsilon_c} + \frac{1}{A_cF_{cb}} + \frac{1-\varepsilon_b}{A_b\varepsilon_b}}$$

$$\frac{(1073 \text{ K})^4 - T_c^4}{\frac{1-0.8}{(0.04 \text{ m}^2)(0.8)} + \frac{1}{(0.04 \text{ m}^2)(0.981)} + \frac{1-0.1}{(4 \text{ m}^2)(0.1)}} = \frac{T_c^4 - (473 \text{ K})^4}{\frac{1-0.1}{(4 \text{ m}^2)(0.1)} + \frac{1}{(4 \text{ m}^2)(0.0881)} + \frac{1-0.4}{(0.36 \text{ m}^2)(0.4)}}$$

Solving the equation with an equation solver such as EES, we obtain $T_c = 754 \text{ K} = \mathbf{481^\circ\text{C}}$

15-109 Radiation heat transfer occurs between two concentric disks. The view factors and the net rate of radiation heat transfer for two cases are to be determined.

Assumptions 1 Steady operating conditions exist 2 The surfaces are opaque, diffuse, and gray. 3 Convection heat transfer is not considered.

Properties The emissivities of disk 1 and 2 are given to be $\varepsilon_a = 0.6$ and $\varepsilon_b = 0.8$, respectively.

Analysis (a) The view factor from surface 1 to surface 2 is determined using Fig. 15-7 as

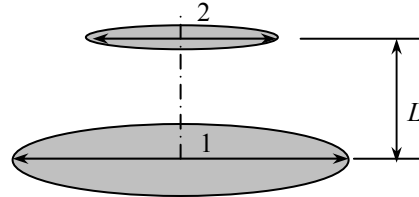
$$\frac{L}{r_1} = \frac{0.10}{0.20} = 0.5, \quad \frac{r_2}{L} = \frac{0.10}{0.10} = 1 \longrightarrow F_{12} = \mathbf{0.19}$$

Using reciprocity rule,

$$A_1 = \pi(0.2 \text{ m})^2 = 0.1257 \text{ m}^2$$

$$A_2 = \pi(0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$

$$F_{21} = \frac{A_1}{A_2} F_{12} = \frac{0.1257 \text{ m}^2}{0.0314 \text{ m}^2} (0.19) = \mathbf{0.76}$$



(b) The net rate of radiation heat transfer between the surfaces can be determined from

$$\dot{Q} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}} = \frac{(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(1073 \text{ K})^4 - (573 \text{ K})^4]}{\frac{1-0.6}{(0.1257 \text{ m}^2)(0.6)} + \frac{1}{(0.1257 \text{ m}^2)(0.19)} + \frac{1-0.8}{(0.0314 \text{ m}^2)(0.8)}} = \mathbf{1250 \text{ W}}$$

(c) When the space between the disks is completely surrounded by a refractory surface, the net rate of radiation heat transfer can be determined from

$$\begin{aligned} \dot{Q} &= \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{A_1 + A_2 - 2A_1F_{12}}{A_2 - A_1F_{12}^2} + \left(\frac{1}{\varepsilon_1} - 1\right) + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)} \\ &= \frac{(0.1257 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(1073 \text{ K})^4 - (573 \text{ K})^4]}{\frac{0.1257 + 0.0314 - 2(0.1257)(0.19)}{0.0314 - (0.1257)(0.19)^2} + \left(\frac{1}{0.6} - 1\right) + \frac{0.1257}{0.0314}\left(\frac{1}{0.8} - 1\right)} = \mathbf{1510 \text{ W}} \end{aligned}$$

Discussion The rate of heat transfer in part (c) is 21 percent higher than that in part (b).

15-110 15-113 Design and Essay Problems



Solutions Manual
for
Introduction to Thermodynamics and Heat Transfer
Yunus A. Cengel
2nd Edition, 2008

Chapter 16
HEAT EXCHANGERS

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Types of Heat Exchangers

16-1C Heat exchangers are classified according to the flow type as parallel flow, counter flow, and cross-flow arrangement. In parallel flow, both the hot and cold fluids enter the heat exchanger at the same end and move in the same direction. In counter-flow, the hot and cold fluids enter the heat exchanger at opposite ends and flow in opposite direction. In cross-flow, the hot and cold fluid streams move perpendicular to each other.

16-2C In terms of construction type, heat exchangers are classified as compact, shell and tube and regenerative heat exchangers. Compact heat exchangers are specifically designed to obtain large heat transfer surface areas per unit volume. The large surface area in compact heat exchangers is obtained by attaching closely spaced thin plate or corrugated fins to the walls separating the two fluids. Shell and tube heat exchangers contain a large number of tubes packed in a shell with their axes parallel to that of the shell. Regenerative heat exchangers involve the alternate passage of the hot and cold fluid streams through the same flow area. In compact heat exchangers, the two fluids usually move perpendicular to each other.

16-3C A heat exchanger is classified as being compact if $\beta > 700 \text{ m}^2/\text{m}^3$ or $(200 \text{ ft}^2/\text{ft}^3)$ where β is the ratio of the heat transfer surface area to its volume which is called the area density. The area density for double-pipe heat exchanger can not be in the order of 700. Therefore, it can not be classified as a compact heat exchanger.

16-4C In counter-flow heat exchangers, the hot and the cold fluids move parallel to each other but both enter the heat exchanger at opposite ends and flow in opposite direction. In cross-flow heat exchangers, the two fluids usually move perpendicular to each other. The cross-flow is said to be unmixed when the plate fins force the fluid to flow through a particular interfin spacing and prevent it from moving in the transverse direction. When the fluid is free to move in the transverse direction, the cross-flow is said to be mixed.

16-5C In the shell and tube exchangers, baffles are commonly placed in the shell to force the shell side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes. Baffles disrupt the flow of fluid, and an increased pumping power will be needed to maintain flow. On the other hand, baffles eliminate dead spots and increase heat transfer rate.

16-6C Using six-tube passes in a shell and tube heat exchanger increases the heat transfer surface area, and the rate of heat transfer increases. But it also increases the manufacturing costs.

16-7C Using so many tubes increases the heat transfer surface area which in turn increases the rate of heat transfer.

16-8C Regenerative heat exchanger involves the alternate passage of the hot and cold fluid streams through the same flow area. The static type regenerative heat exchanger is basically a porous mass which has a large heat storage capacity, such as a ceramic wire mesh. Hot and cold fluids flow through this porous mass alternately. Heat is transferred from the hot fluid to the matrix of the regenerator during the flow of the hot fluid and from the matrix to the cold fluid. Thus the matrix serves as a temporary heat storage medium. The dynamic type regenerator involves a rotating drum and continuous flow of the hot and cold fluid through different portions of the drum so that any portion of the drum passes periodically through the hot stream, storing heat and then through the cold stream, rejecting this stored heat. Again the drum serves as the medium to transport the heat from the hot to the cold fluid stream.

The Overall Heat Transfer Coefficient

16-9C Heat is first transferred from the hot fluid to the wall by convection, through the wall by conduction and from the wall to the cold fluid again by convection.

16-10C When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, which is usually the case, the thermal resistance of the tube is negligible.

16-11C The heat transfer surface areas are $A_i = \pi D_1 L$ and $A_o = \pi D_2 L$. When the thickness of inner tube is small, it is reasonable to assume $A_i \cong A_o \cong A_s$.

16-12C No, it is not reasonable to say $h_i \approx h_o \approx h$

16-13C When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, the thermal resistance of the tube is negligible and the inner and the outer surfaces of the tube are almost identical ($A_i \cong A_o \cong A_s$). Then the overall heat transfer coefficient of a heat exchanger can be determined to from $U = (1/h_i + 1/h_o)^{-1}$

16-14C None.

16-15C When one of the convection coefficients is much smaller than the other $h_i \ll h_o$, and $A_i \approx A_o \approx A_s$. Then we have $(1/h_i \gg 1/h_o)$ and thus $U_i = U_o = U \cong h_i$.

16-16C The most common type of fouling is the precipitation of solid deposits in a fluid on the heat transfer surfaces. Another form of fouling is corrosion and other chemical fouling. Heat exchangers may also be fouled by the growth of algae in warm fluids. This type of fouling is called the biological fouling. Fouling represents additional resistance to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease, and the pressure drop to increase.

16-17C The effect of fouling on a heat transfer is represented by a fouling factor R_f . Its effect on the heat transfer coefficient is accounted for by introducing a thermal resistance R_f/A_s . The fouling increases with increasing temperature and decreasing velocity.

16-18 The heat transfer coefficients and the fouling factors on tube and shell side of a heat exchanger are given. The thermal resistance and the overall heat transfer coefficients based on the inner and outer areas are to be determined.

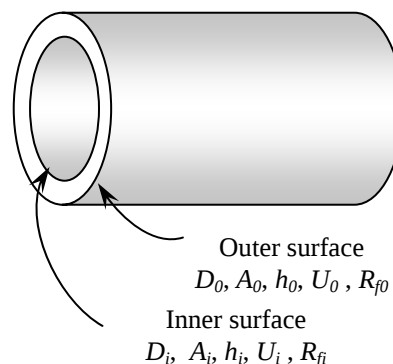
Assumptions 1 The heat transfer coefficients and the fouling factors are constant and uniform.

Analysis (a) The total thermal resistance of the heat exchanger per unit length is

$$\begin{aligned}
 R &= \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o} \\
 R &= \frac{1}{(700 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.012 \text{ m})(1 \text{ m})]} + \frac{(0.0005 \text{ m}^2 \cdot ^\circ\text{C/W})}{[\pi(0.012 \text{ m})(1 \text{ m})]} \\
 &\quad + \frac{\ln(1.6 / 1.2)}{2\pi(380 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} + \frac{(0.0002 \text{ m}^2 \cdot ^\circ\text{C/W})}{[\pi(0.016 \text{ m})(1 \text{ m})]} \\
 &\quad + \frac{1}{(700 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.016 \text{ m})(1 \text{ m})]} \\
 &= \mathbf{0.0837 ^\circ\text{C/W}}
 \end{aligned}$$

(b) The overall heat transfer coefficient based on the inner and the outer surface areas of the tube per length are

$$\begin{aligned}
 R &= \frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} \\
 U_i &= \frac{1}{RA_i} = \frac{1}{(0.0837 ^\circ\text{C/W})[\pi(0.012 \text{ m})(1 \text{ m})]} = \mathbf{317 \text{ W/m}^2 \cdot ^\circ\text{C}} \\
 U_o &= \frac{1}{RA_o} = \frac{1}{(0.0837 ^\circ\text{C/W})[\pi(0.016 \text{ m})(1 \text{ m})]} = \mathbf{238 \text{ W/m}^2 \cdot ^\circ\text{C}}
 \end{aligned}$$



16-19 EES Prob. 16-18 is reconsidered. The effects of pipe conductivity and heat transfer coefficients on the thermal resistance of the heat exchanger are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

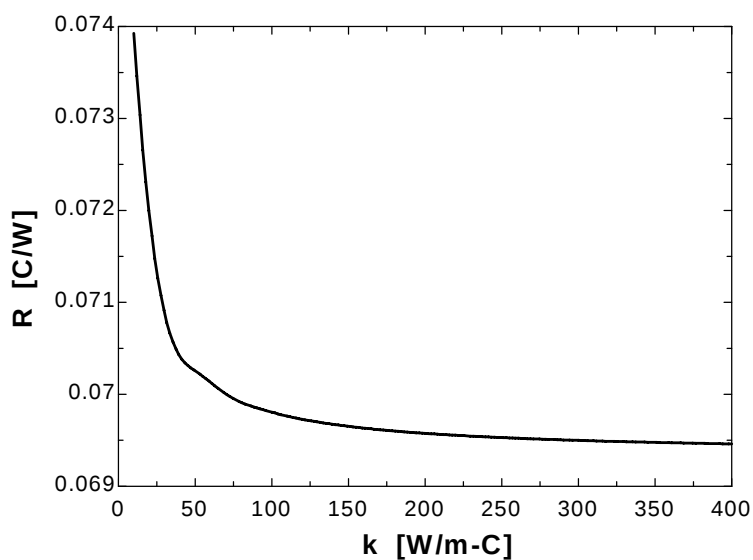
"GIVEN"

$k=380$ [W/m-C]
 $D_i=0.012$ [m]
 $D_o=0.016$ [m]
 $D_2=0.03$ [m]
 $h_i=700$ [W/m²-C]
 $h_o=1400$ [W/m²-C]
 $R_{f_i}=0.0005$ [m²-C/W]
 $R_{f_o}=0.0002$ [m²-C/W]

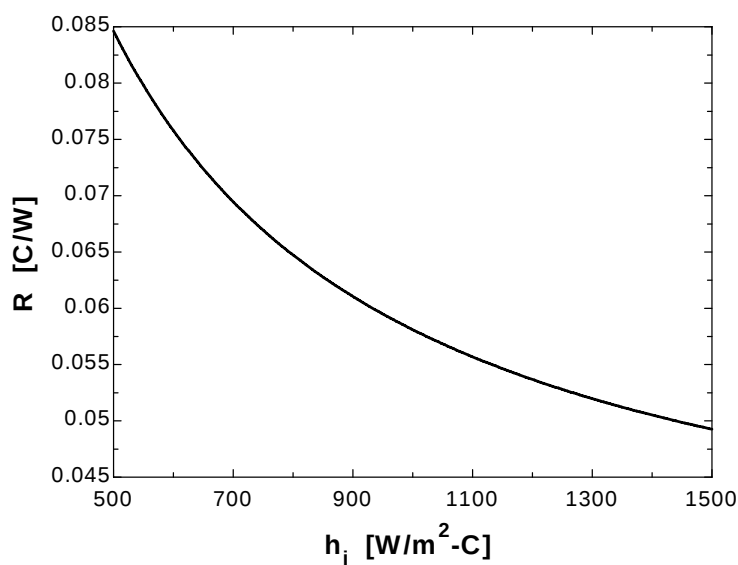
"ANALYSIS"

$R=1/(h_i A_i)+R_{f_i}/A_i+\ln(D_o/D_i)/(2\pi k L)+R_{f_o}/A_o+1/(h_o A_o)$
 $L=1$ [m] "a unit length of the heat exchanger is considered"
 $A_i=\pi D_i L$
 $A_o=\pi D_o L$

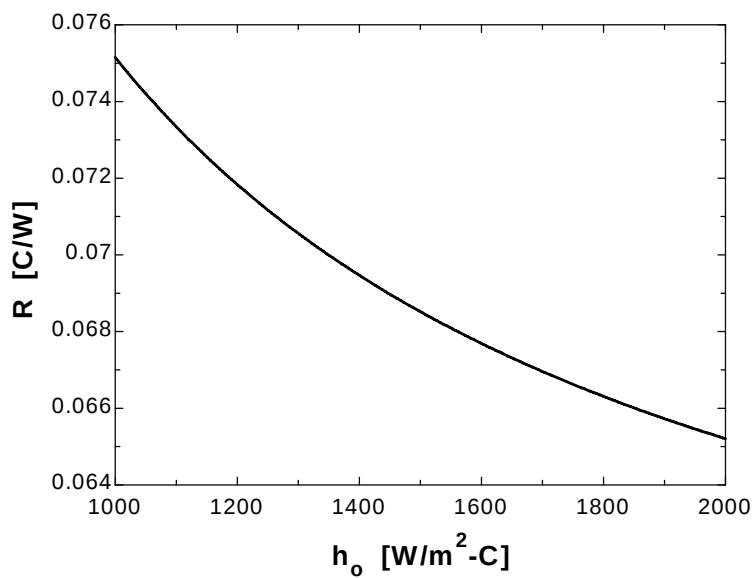
k [W/m-C]	R [C/W]
10	0.07392
30.53	0.07085
51.05	0.07024
71.58	0.06999
92.11	0.06984
112.6	0.06975
133.2	0.06969
153.7	0.06964
174.2	0.06961
194.7	0.06958
215.3	0.06956
235.8	0.06954
256.3	0.06952
276.8	0.06951
297.4	0.0695
317.9	0.06949
338.4	0.06948
358.9	0.06947
379.5	0.06947
400	0.06946



h_i [W/m ² -C]	R [C/W]
500	0.08462
550	0.0798
600	0.07578
650	0.07238
700	0.06947
750	0.06694
800	0.06473
850	0.06278
900	0.06105
950	0.05949
1000	0.0581
1050	0.05684
1100	0.05569
1150	0.05464
1200	0.05368
1250	0.05279
1300	0.05198
1350	0.05122
1400	0.05052
1450	0.04987
1500	0.04926



h_o [W/m ² -C]	R [C/W]
1000	0.07515
1050	0.0742
1100	0.07334
1150	0.07256
1200	0.07183
1250	0.07117
1300	0.07056
1350	0.06999
1400	0.06947
1450	0.06898
1500	0.06852
1550	0.06809
1600	0.06769
1650	0.06731
1700	0.06696
1750	0.06662
1800	0.06631
1850	0.06601
1900	0.06573
1950	0.06546
2000	0.0652



16-20 A water stream is heated by a jacketed-agitated vessel, fitted with a turbine agitator. The mass flow rate of water is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The properties of water at 54°C are (Table A-15)

$$k = 0.648 \text{ W/m} \cdot ^\circ\text{C}$$

$$\rho = 985.8 \text{ kg/m}^3$$

$$\mu = 0.513 \times 10^{-3} \text{ kg/m} \cdot \text{s}$$

$$\text{Pr} = 3.31$$

The specific heat of water at the average temperature of $(10+54)/2=32^\circ\text{C}$ is $4178 \text{ J/kg} \cdot ^\circ\text{C}$ (Table A-15)

Analysis We first determine the heat transfer coefficient on the inner wall of the vessel

$$\text{Re} = \frac{\dot{n} D_a^2 \rho}{\mu} = \frac{(60/60 \text{ s}^{-1})(0.2 \text{ m})^2 (985.8 \text{ kg/m}^3)}{0.513 \times 10^{-3} \text{ kg/m} \cdot \text{s}} = 76,865$$

$$\text{Nu} = 0.76 \text{Re}^{2/3} \text{Pr}^{1/3} = 0.76(76,865)^{2/3} (3.31)^{1/3} = 2048$$

$$h_j = \frac{k}{D_t} \text{Nu} = \frac{0.648 \text{ W/m} \cdot ^\circ\text{C}}{0.6 \text{ m}} (2048) = 2211 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The heat transfer coefficient on the outer side is determined as follows

$$h_o = 13,100(T_g - T_w)^{-0.25} = 13,100(100 - T_w)^{-0.25}$$

$$h_o(T_g - T_w) = h_j(T_w - 54)$$

$$13,100(100 - T_w)^{-0.25}(100 - T_w) = 2211(T_w - 54)$$

$$13,100(100 - T_w)^{0.75} = 2211(T_w - 54)$$

$$\rightarrow T_w = 89.2^\circ\text{C}$$

$$h_o = 13,100(100 - T_w)^{-0.25} = 13,100(100 - 89.2)^{-0.25} = 7226 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Neglecting the wall resistance and the thickness of the wall, the overall heat transfer coefficient can be written as

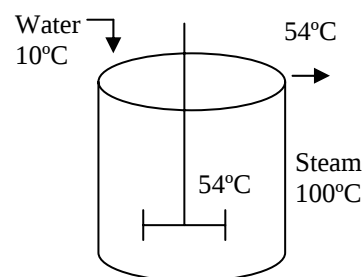
$$U = \left(\frac{1}{h_j} + \frac{1}{h_o} \right)^{-1} = \left(\frac{1}{2211} + \frac{1}{7226} \right)^{-1} = 1694 \text{ W/m}^2 \cdot ^\circ\text{C}$$

From an energy balance

$$[\dot{m}c(T_{out} - T_{in})]_{\text{water}} = UA\Delta T$$

$$\dot{m}_w(4178)(54 - 10) = (1694)(\pi \times 0.6 \times 0.6)(100 - 54)$$

$$\dot{m}_w = 0.479 \text{ kg/s} = \mathbf{1725 \text{ kg/h}}$$



16-21 Water flows through the tubes in a boiler. The overall heat transfer coefficient of this boiler based on the inner surface area is to be determined.

Assumptions 1 Water flow is fully developed. 2 Properties of the water are constant.

Properties The properties of water at 110°C are (Table A-15)

$$\nu = \mu / \rho = 0.268 \times 10^{-6} \text{ m}^2/\text{s}$$

$$k = 0.682 \text{ W/m}^2 \cdot \text{K}$$

$$\text{Pr} = 1.58$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.5 \text{ m/s})(0.01 \text{ m})}{0.268 \times 10^{-6} \text{ m}^2/\text{s}} = 130,600$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow,

$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(130,600)^{0.8} (1.58)^{0.4} = 342$$

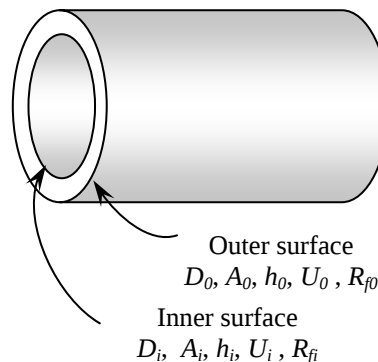
$$\text{and } h = \frac{k}{D_h} \text{Nu} = \frac{0.682 \text{ W/m} \cdot ^\circ\text{C}}{0.01 \text{ m}} (342) = 23,324 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The total resistance of this heat exchanger is then determined from

$$\begin{aligned} R = R_{\text{total}} &= R_i + R_{\text{wall}} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{1}{h_o A_o} \\ &= \frac{1}{(23,324 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.01 \text{ m})(5 \text{ m})]} + \frac{\ln(1.4 / 1)}{[2\pi(14.2 \text{ W/m} \cdot ^\circ\text{C})(5 \text{ m})]} \\ &\quad + \frac{1}{(8400 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.014 \text{ m})(5 \text{ m})]} \\ &= 0.00157^\circ\text{C/W} \end{aligned}$$

and

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{R A_i} = \frac{1}{(0.00157^\circ\text{C/W})[\pi(0.01 \text{ m})(5 \text{ m})]} = 4055 \text{ W/m}^2 \cdot ^\circ\text{C}$$



16-22 Water is flowing through the tubes in a boiler. The overall heat transfer coefficient of this boiler based on the inner surface area is to be determined.

Assumptions 1 Water flow is fully developed. 2 Properties of water are constant. 3 The heat transfer coefficient and the fouling factor are constant and uniform.

Properties The properties of water at 110°C are (Table A-15)

$$\nu = \mu / \rho = 0.268 \times 10^{-6} \text{ m}^2/\text{s}$$

$$k = 0.682 \text{ W/m}^2 \cdot \text{K}$$

$$\text{Pr} = 1.58$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.5 \text{ m/s})(0.01 \text{ m})}{0.268 \times 10^{-6} \text{ m}^2/\text{s}} = 130,600$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow,

$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(130,600)^{0.8} (1.58)^{0.4} = 342$$

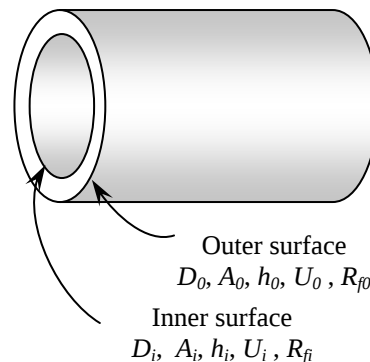
$$\text{and } h = \frac{k}{D_h} \text{Nu} = \frac{0.682 \text{ W/m} \cdot ^\circ\text{C}}{0.01 \text{ m}} (342) = 23,324 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The thermal resistance of heat exchanger with a fouling factor of $R_{f,i} = 0.0005 \text{ m}^2 \cdot ^\circ\text{C/W}$ is determined from

$$\begin{aligned} R &= \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{1}{h_o A_o} \\ R &= \frac{1}{(23,324 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.01 \text{ m})(5 \text{ m})]} + \frac{0.0005 \text{ m}^2 \cdot ^\circ\text{C/W}}{[\pi(0.01 \text{ m})(5 \text{ m})]} \\ &\quad + \frac{\ln(1.4 / 1)}{2\pi(14.2 \text{ W/m} \cdot ^\circ\text{C})(5 \text{ m})} + \frac{1}{(8400 \text{ W/m}^2 \cdot ^\circ\text{C})[\pi(0.014 \text{ m})(5 \text{ m})]} \\ &= 0.00475^\circ\text{C/W} \end{aligned}$$

Then,

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{R A_i} = \frac{1}{(0.00475^\circ\text{C/W})[\pi(0.01 \text{ m})(5 \text{ m})]} = \mathbf{1340 \text{ W/m}^2 \cdot ^\circ\text{C}}$$



16-23 EES Prob. 16-22 is reconsidered. The overall heat transfer coefficient based on the inner surface as a function of fouling factor is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

T_w=110 [C]
 Vel=3.5 [m/s]
 L=5 [m]
 k_pipe=14.2 [W/m-C]
 D_i=0.010 [m]
 D_o=0.014 [m]
 h_o=8400 [W/m^2-C]
 R_f_i=0.0005 [m^2-C/W]

"PROPERTIES"

k=conductivity(Water, T=T_w, P=300)
 Pr=Prandtl(Water, T=T_w, P=300)
 rho=density(Water, T=T_w, P=300)
 mu=viscosity(Water, T=T_w, P=300)
 nu=mu/rho

"ANALYSIS"

Re=(Vel*D_i)/nu "Re is calculated to be greater than 10,000. Therefore, the flow is turbulent."

Nusselt=0.023*Re^0.8*Pr^0.4

h_i=k/D_i*Nusselt

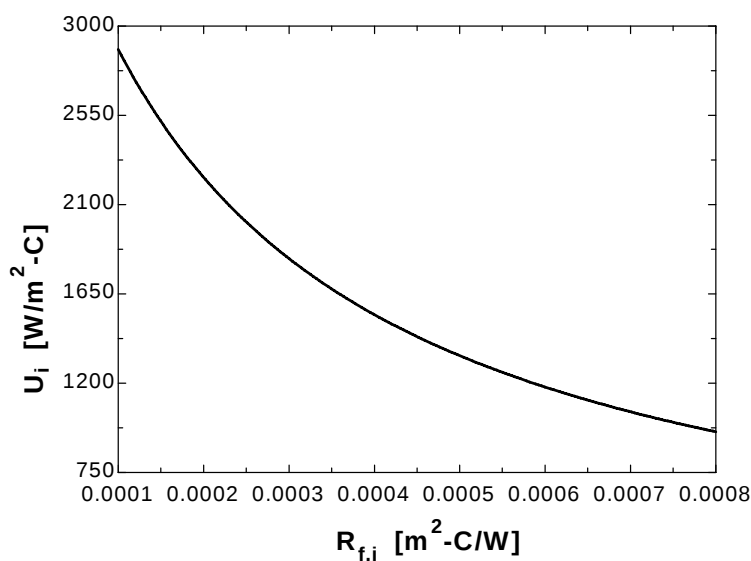
A_i=pi*D_i*L

A_o=pi*D_o*L

R=1/(h_i*A_i)+R_f_i/A_i+ln(D_o/D_i)/(2*pi*k_pipe*L)+1/(h_o*A_o)

U_i=1/(R*A_i)

R _{f,i} [m ² -C/W]	U _i [W/m ² -C]
0.0001	2883
0.00015	2520
0.0002	2238
0.00025	2013
0.0003	1829
0.00035	1675
0.0004	1546
0.00045	1435
0.0005	1339
0.00055	1255
0.0006	1181
0.00065	1115
0.0007	1056
0.00075	1003
0.0008	955.2



16-24 Refrigerant-134a is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions 1 The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. 2 Both the water and refrigerant-134a flow are fully developed. 3 Properties of the water and refrigerant-134a are constant.

Properties The properties of water at 20°C are (Table A-15)

$$\rho = 998 \text{ kg/m}^3$$

$$\nu = \mu / \rho = 1.004 \times 10^{-6} \text{ m}^2/\text{s}$$

$$k = 0.598 \text{ W/m} \cdot ^\circ\text{C}$$

$$\text{Pr} = 7.01$$

Analysis The hydraulic diameter for annular space is

$$D_h = D_o - D_i = 0.025 - 0.01 = 0.015 \text{ m}$$

The average velocity of water in the tube and the Reynolds number are

$$V_{avg} = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho \left(\pi \frac{D_o^2 - D_i^2}{4} \right)} = \frac{0.3 \text{ kg/s}}{(998 \text{ kg/m}^3) \left(\pi \frac{(0.025 \text{ m})^2 - (0.01 \text{ m})^2}{4} \right)} = 0.729 \text{ m/s}$$

$$\text{Re} = \frac{V_{avg} D_h}{\nu} = \frac{(0.729 \text{ m/s})(0.015 \text{ m})}{1.004 \times 10^{-6} \text{ m}^2/\text{s}} = 10,890$$

which is greater than 4000. Therefore flow is turbulent. Assuming fully developed flow,

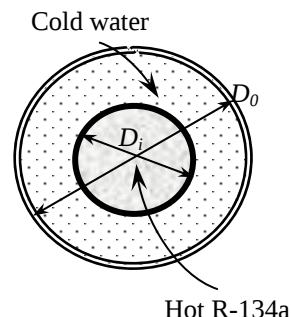
$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(10,890)^{0.8} (7.01)^{0.4} = 85.0$$

and

$$h_o = \frac{k}{D_h} \text{Nu} = \frac{0.598 \text{ W/m} \cdot ^\circ\text{C}}{0.015 \text{ m}} (85.0) = 3390 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5000 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{1}{3390 \text{ W/m}^2 \cdot ^\circ\text{C}}} = 2020 \text{ W/m}^2 \cdot ^\circ\text{C}$$



16-25 Refrigerant-134a is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions 1 The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. 2 Both the water and refrigerant-134a flows are fully developed. 3 Properties of the water and refrigerant-134a are constant. 4 The limestone layer can be treated as a plain layer since its thickness is very small relative to its diameter.

Properties The properties of water at 20°C are (Table A-15)

$$\rho = 998 \text{ kg/m}^3$$

$$\nu = \mu / \rho = 1.004 \times 10^{-6} \text{ m}^2/\text{s}$$

$$k = 0.598 \text{ W/m} \cdot ^\circ\text{C}$$

$$\text{Pr} = 7.01$$

Analysis The hydraulic diameter for annular space is

$$D_h = D_o - D_i = 0.025 - 0.01 = 0.015 \text{ m}$$

The average velocity of water in the tube and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho \left(\pi \frac{D_o^2 - D_i^2}{4} \right)} = \frac{0.3 \text{ kg/s}}{(998 \text{ kg/m}^3) \left(\pi \frac{(0.025 \text{ m})^2 - (0.01 \text{ m})^2}{4} \right)} = 0.729 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.729 \text{ m/s})(0.015 \text{ m})}{1.004 \times 10^{-6} \text{ m}^2/\text{s}} = 10,890$$

which is greater than 10,000. Therefore flow is turbulent. Assuming fully developed flow,

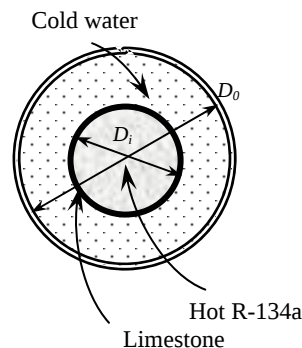
$$Nu = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(10,890)^{0.8} (7.01)^{0.4} = 85.0$$

and

$$h_o = \frac{k}{D_h} Nu = \frac{0.598 \text{ W/m} \cdot ^\circ\text{C}}{0.015 \text{ m}} (85.0) = 3390 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Disregarding the curvature effects, the overall heat transfer coefficient is determined to be

$$U = \frac{1}{\frac{1}{h_i} + \left(\frac{L}{k} \right)_{\text{limestone}} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5000 \text{ W/m}^2 \cdot ^\circ\text{C}} + \frac{0.002 \text{ m}}{1.3 \text{ W/m} \cdot ^\circ\text{C}} + \frac{1}{3390 \text{ W/m}^2 \cdot ^\circ\text{C}}} = 493 \text{ W/m}^2 \cdot ^\circ\text{C}$$



16-26 EES Prob. 16-25 is reconsidered. The overall heat transfer coefficient as a function of the limestone thickness is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

D_i=0.010 [m]
 D_o=0.025 [m]
 T_w=20 [C]
 h_i=5000 [W/m²-C]
 m_{dot}=0.3 [kg/s]
 L_{limestone}=2 [mm]
 k_{limestone}=1.3 [W/m-C]

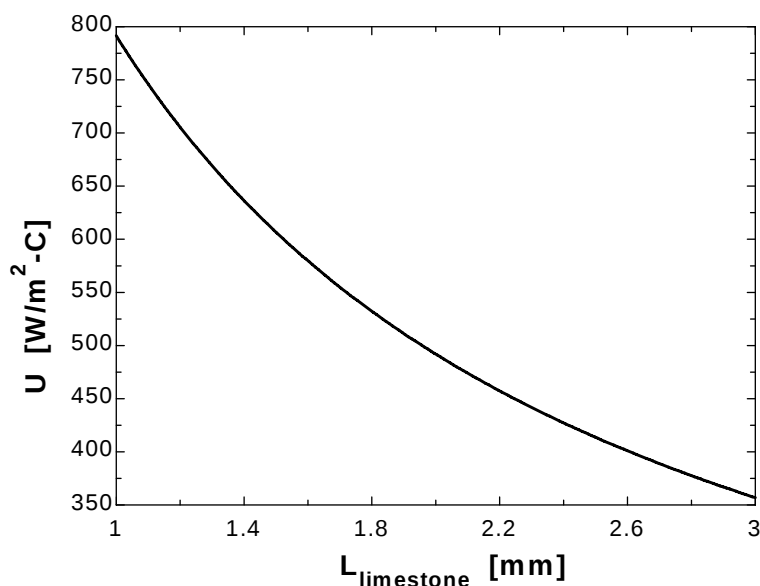
"PROPERTIES"

k=conductivity(Water, T=T_w, P=100)
 Pr=Prandtl(Water, T=T_w, P=100)
 rho=density(Water, T=T_w, P=100)
 mu=viscosity(Water, T=T_w, P=100)
 nu=mu/rho

"ANALYSIS"

D_h=D_o-D_i
 Vel=m_{dot}/(rho*A_c)
 A_c=pi*(D_o²-D_i²)/4
 Re=(Vel*D_h)/nu
 "Re is calculated to be greater than 10,000. Therefore, the flow is turbulent."
 Nusselt=0.023*Re^{0.8}*Pr^{0.4}
 h_o=k/D_h*Nusselt
 U=1/(1/h_i+(L_{limestone}*Convert(mm, m))/k_{limestone}+1/h_o)

L _{limestone} [mm]	U [W/m ² -C]
1	791.4
1.1	746
1.2	705.5
1.3	669.2
1.4	636.4
1.5	606.7
1.6	579.7
1.7	554.9
1.8	532.2
1.9	511.3
2	491.9
2.1	474
2.2	457.3
2.3	441.8
2.4	427.3
2.5	413.7
2.6	400.9
2.7	388.9
2.8	377.6
2.9	367
3	356.9



16-27E Water is cooled by air in a cross-flow heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions 1 The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and air flow are fully developed. **3** Properties of the water and air are constant.

Properties The properties of water at 180°F are (Table A-15E)

$$k = 0.388 \text{ Btu/h.ft.}^\circ\text{F}$$

$$\nu = 3.825 \times 10^{-6} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 2.15$$

The properties of air at 80°F are (Table A-22E)

$$k = 0.01481 \text{ Btu/h.ft.}^\circ\text{F}$$

$$\nu = 1.697 \times 10^{-4} \text{ ft}^2/\text{s}$$

$$\text{Pr} = 0.7290$$

Analysis The overall heat transfer coefficient can be determined from

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

The Reynolds number of water is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4 \text{ ft/s})(0.75/12 \text{ ft})}{3.825 \times 10^{-6} \text{ ft}^2/\text{s}} = 65,360$$

which is greater than 10,000. Therefore the flow of water is turbulent. Assuming the flow to be fully developed, the Nusselt number is determined from

$$\text{Nu} = \frac{h D_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(65,360)^{0.8} (2.15)^{0.4} = 222$$

$$\text{and } h_i = \frac{k}{D_h} \text{Nu} = \frac{0.388 \text{ Btu/h.ft.}^\circ\text{F}}{0.75/12 \text{ ft}} (222) = 1378 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

The Reynolds number of air is

$$\text{Re} = \frac{VD}{\nu} = \frac{(12 \text{ ft/s})[3/(4 \times 12) \text{ ft}]}{1.697 \times 10^{-4} \text{ ft}^2/\text{s}} = 4420$$

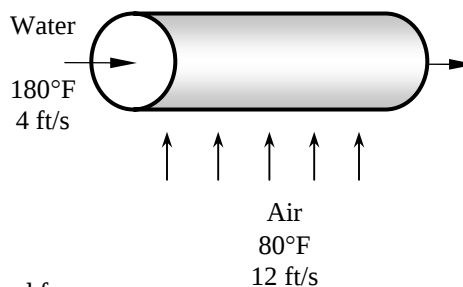
The flow of air is across the cylinder. The proper relation for Nusselt number in this case is

$$\begin{aligned} \text{Nu} = \frac{hD}{k} &= 0.3 + \frac{0.62 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5} \\ &= 0.3 + \frac{0.62(4420)^{0.5} (0.7290)^{1/3}}{\left[1 + (0.4/0.7290)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{4420}{282,000}\right)^{5/8}\right]^{4/5} = 34.86 \end{aligned}$$

$$\text{and } h_o = \frac{k}{D} \text{Nu} = \frac{0.01481 \text{ Btu/h.ft.}^\circ\text{F}}{0.75/12 \text{ ft}} (34.86) = 8.26 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{1378 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}} + \frac{1}{8.26 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}}} = \mathbf{8.21 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}}$$



Analysis of Heat Exchangers

16-28C The heat exchangers usually operate for long periods of time with no change in their operating conditions, and then they can be modeled as steady-flow devices. As such, the mass flow rate of each fluid remains constant and the fluid properties such as temperature and velocity at any inlet and outlet remain constant. The kinetic and potential energy changes are negligible. The specific heat of a fluid can be treated as constant in a specified temperature range. Axial heat conduction along the tube is negligible. Finally, the outer surface of the heat exchanger is assumed to be perfectly insulated so that there is no heat loss to the surrounding medium and any heat transfer thus occurs is between the two fluids only.

16-29C That relation is valid under steady operating conditions, constant specific heats, and negligible heat loss from the heat exchanger.

16-30C The product of the mass flow rate and the specific heat of a fluid is called the heat capacity rate and is expressed as $C = \dot{m}c_p$. When the heat capacity rates of the cold and hot fluids are equal, the temperature change is the same for the two fluids in a heat exchanger. That is, the temperature rise of the cold fluid is equal to the temperature drop of the hot fluid. A heat capacity of infinity for a fluid in a heat exchanger is experienced during a phase-change process in a condenser or boiler.

16-31C The mass flow rate of the cooling water can be determined from $\dot{Q} = (\dot{m}c_p\Delta T)_{\text{cooling water}}$. The rate of condensation of the steam is determined from $\dot{Q} = (\dot{m}h_{fg})_{\text{steam}}$, and the total thermal resistance of the condenser is determined from $R = \dot{Q} / \Delta T$.

16-32C When the heat capacity rates of the cold and hot fluids are identical, the temperature rise of the cold fluid will be equal to the temperature drop of the hot fluid.

The Log Mean Temperature Difference Method

16-33C ΔT_{lm} is called the log mean temperature difference, and is expressed as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

where

$$\Delta T_1 = T_{h,in} - T_{c,in} \quad \Delta T_2 = T_{h,out} - T_{c,out} \quad \text{for parallel-flow heat exchangers and}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} \quad \Delta T_2 = T_{h,out} - T_{c,in} \quad \text{for counter-flow heat exchangers}$$

16-34C The temperature difference between the two fluids decreases from ΔT_1 at the inlet to ΔT_2 at the outlet, and arithmetic mean temperature difference is defined as $\Delta T_{am} = \frac{\Delta T_1 + \Delta T_2}{2}$. The logarithmic mean temperature difference ΔT_{lm} is obtained by tracing the actual temperature profile of the fluids along the heat exchanger, and is an exact representation of the average temperature difference between the hot and cold fluids. It truly reflects the exponential decay of the local temperature difference. The logarithmic mean temperature difference is always less than the arithmetic mean temperature.

16-35C ΔT_{lm} cannot be greater than both ΔT_1 and ΔT_2 because ΔT_{lm} is always less than or equal to ΔT_m (arithmetic mean) which can not be greater than both ΔT_1 and ΔT_2 .

16-36C No, it cannot. When ΔT_1 is less than ΔT_2 the ratio of them must be less than one and the natural logarithms of the numbers which are less than 1 are negative. But the numerator is also negative in this case. When ΔT_1 is greater than ΔT_2 , we obtain positive numbers at the both numerator and denominator.

16-37C In the parallel-flow heat exchangers the hot and cold fluids enter the heat exchanger at the same end, and the temperature of the hot fluid decreases and the temperature of the cold fluid increases along the heat exchanger. But the temperature of the cold fluid can never exceed that of the hot fluid. In case of the counter-flow heat exchangers the hot and cold fluids enter the heat exchanger from the opposite ends and the outlet temperature of the cold fluid may exceed the outlet temperature of the hot fluid.

16-38C The ΔT_{lm} will be greatest for double-pipe counter-flow heat exchangers.

16-39C The factor F is called as correction factor which depends on the geometry of the heat exchanger and the inlet and the outlet temperatures of the hot and cold fluid streams. It represents how closely a heat exchanger approximates a counter-flow heat exchanger in terms of its logarithmic mean temperature difference. F cannot be greater than unity.

16-40C In this case it is not practical to use the LMTD method because it requires tedious iterations. Instead, the effectiveness-NTU method should be used.

16-41C First heat transfer rate is determined from $\dot{Q} = \dot{m}c_p[T_{in} - T_{out}]$, ΔT_{lm} from $\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$,

correction factor from the figures, and finally the surface area of the heat exchanger from

$$\dot{Q} = UA F \Delta T_{lm,CF}$$

16-42 Ethylene glycol is heated in a tube while steam condenses on the outside tube surface. The tube length is to be determined.

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the tubes are smooth. 3 Heat transfer to the surroundings is negligible.

Properties The properties of ethylene glycol are given to be $\rho = 1109 \text{ kg/m}^3$, $c_p = 2428 \text{ J/kg}\cdot\text{K}$, $k = 0.253 \text{ W/m}\cdot\text{K}$, $\mu = 0.01545 \text{ kg/m}\cdot\text{s}$, $\text{Pr} = 148.5$. The thermal conductivity of copper is given to be $386 \text{ W/m}\cdot\text{K}$.

Analysis The rate of heat transfer is

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (1 \text{ kg/s})(2428 \text{ J/kg}\cdot^\circ\text{C})(40 - 20)^\circ\text{C} = 48,560 \text{ W}$$

The fluid velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{1 \text{ kg/s}}{(1109 \text{ kg/m}^3) [\pi (0.02 \text{ m})^2 / 4]} = 2.870 \text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(1109 \text{ kg/m}^3)(2.870 \text{ m/s})(0.02 \text{ m})}{0.01545 \text{ kg/m}\cdot\text{s}} = 4121$$

which is greater than 2300 and smaller than 10,000. Therefore, we have transitional flow. We assume fully developed flow and evaluate the Nusselt number from turbulent flow relation:

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(4121)^{0.8} (148.5)^{0.4} = 132.5$$

Heat transfer coefficient on the inner surface is

$$h_i = \frac{k}{D} \text{Nu} = \frac{0.253 \text{ W/m}\cdot^\circ\text{C}}{0.02 \text{ m}} (132.5) = 1677 \text{ W/m}^2\cdot^\circ\text{C}$$

Assuming a wall temperature of 100°C , the heat transfer coefficient on the outer surface is determined to be

$$h_o = 9200(T_g - T_w)^{-0.25} = 9200(110 - 100)^{-0.25} = 5174 \text{ W/m}^2\cdot^\circ\text{C}$$

Let us check if the assumption for the wall temperature holds:

$$h_i A_i (T_w - T_{b,\text{avg}}) = h_o A_o (T_g - T_w)$$

$$h_i \pi D_i L (T_w - T_{b,\text{avg}}) = h_o \pi D_o L (T_g - T_w)$$

$$1677 \times 0.02 (T_w - 30) = 5174 \times 0.025 (110 - T_w) \longrightarrow T_w = 93.5^\circ\text{C}$$

Now we assume a wall temperature of 90°C :

$$h_o = 9200(T_g - T_w)^{-0.25} = 9200(110 - 90)^{-0.25} = 4350 \text{ W/m}^2\cdot^\circ\text{C}$$

Again checking, $1677 \times 0.02 (T_w - 30) = 4350 \times 0.025 (110 - T_w) \longrightarrow T_w = 91.1^\circ\text{C}$

which is sufficiently close to the assumed value of 90°C . Now that both heat transfer coefficients are available, we use thermal resistance concept to find overall heat transfer coefficient based on the outer surface area as follows:

$$U_o = \frac{1}{\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_2 / D_1)}{2k_{\text{copper}}} + \frac{1}{h_o}} = \frac{1}{\frac{0.025}{(1677)(0.02)} + \frac{(0.025) \ln(2.5 / 2)}{2(386)} + \frac{1}{4350}} = 1018 \text{ W/m}^2\cdot^\circ\text{C}$$

The rate of heat transfer can be expressed as

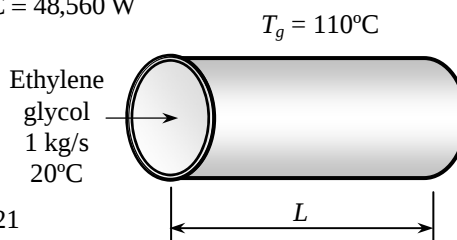
$$\dot{Q} = U_o A_o \Delta T_{\text{lm}}$$

where the logarithmic mean temperature difference is

$$\Delta T_{\text{lm}} = \frac{(T_g - T_e) - (T_g - T_i)}{\ln \left(\frac{T_g - T_e}{T_g - T_i} \right)} = \frac{(110 - 40) - (110 - 20)}{\ln \left(\frac{110 - 40}{110 - 20} \right)} = 79.58^\circ\text{C}$$

Substituting, the tube length is determined to be

$$\dot{Q} = U_o A_o \Delta T_{\text{lm}} \longrightarrow 48,560 = (1018) \pi (0.025) L (79.58) \longrightarrow L = 7.63 \text{ m}$$



16-43 Water is heated in a double-pipe, parallel-flow uninsulated heat exchanger by geothermal water. The rate of heat transfer to the cold water and the log mean temperature difference for this heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heat of hot water is given to be 4.25 kJ/kg·°C.

Analysis The rate of heat given up by the hot water is

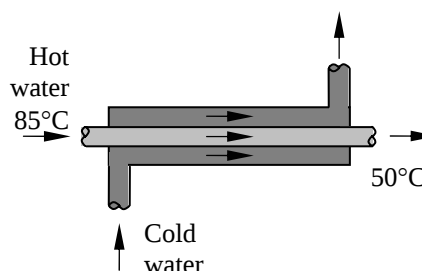
$$\begin{aligned}\dot{Q}_h &= [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \\ &= (1.4 \text{ kg/s})(4.25 \text{ kJ/kg} \cdot ^\circ\text{C})(85^\circ\text{C} - 50^\circ\text{C}) = 208.3 \text{ kW}\end{aligned}$$

The rate of heat picked up by the cold water is

$$\dot{Q}_c = (1 - 0.03)\dot{Q}_h = (1 - 0.03)(208.3 \text{ kW}) = \mathbf{202.0 \text{ kW}}$$

The log mean temperature difference is

$$\dot{Q} = UA\Delta T_{lm} \longrightarrow \Delta T_{lm} = \frac{\dot{Q}}{UA} = \frac{202.0 \text{ kW}}{(1.15 \text{ kW/m}^2 \cdot ^\circ\text{C})(4 \text{ m}^2)} = \mathbf{43.9^\circ\text{C}}$$



16-44 A stream of hydrocarbon is cooled by water in a double-pipe counterflow heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of hydrocarbon and water are given to be 2.2 and 4.18 kJ/kg·°C, respectively.

Analysis The rate of heat transfer is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{HC}} = (720 / 3600 \text{ kg/s})(2.2 \text{ kJ/kg} \cdot ^\circ\text{C})(150^\circ\text{C} - 40^\circ\text{C}) = 48.4 \text{ kW}$$

The outlet temperature of water is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_w \\ 48.4 \text{ kW} &= (540 / 3600 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(T_{w,out} - 10^\circ\text{C}) \\ T_{w,out} &= 87.2^\circ\text{C}\end{aligned}$$

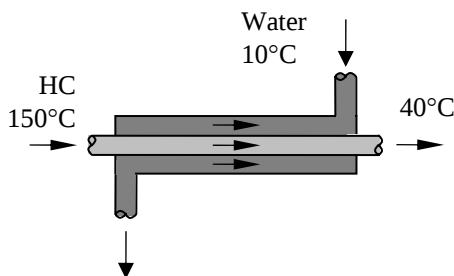
The logarithmic mean temperature difference is

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out} = 150^\circ\text{C} - 87.2^\circ\text{C} = 62.8^\circ\text{C} \\ \Delta T_2 &= T_{h,out} - T_{c,in} = 40^\circ\text{C} - 10^\circ\text{C} = 30^\circ\text{C}\end{aligned}$$

$$\text{and } \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{62.8 - 30}{\ln(62.8 / 30)} = 44.4^\circ\text{C}$$

The overall heat transfer coefficient is determined from

$$\begin{aligned}\dot{Q} &= UA\Delta T_{lm} \\ 48.4 \text{ kW} &= U(\pi \times 0.025 \times 6.0)(44.4^\circ\text{C}) \\ U &= \mathbf{2.31 \text{ kW/m}^2 \cdot \text{K}}\end{aligned}$$



16-45 Oil is heated by water in a 1-shell pass and 6-tube passes heat exchanger. The rate of heat transfer and the heat transfer surface area are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heat of oil is given to be $2.0 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{oil} = (10 \text{ kg/s})(2.0 \text{ kJ/kg} \cdot ^\circ\text{C})(46^\circ\text{C} - 25^\circ\text{C}) = \mathbf{420 \text{ kW}}$$

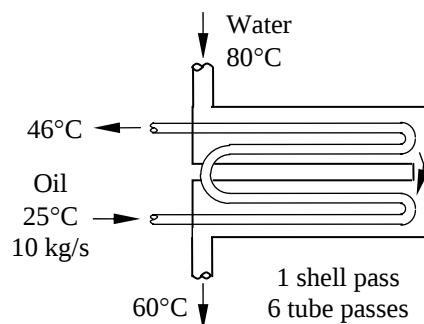
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80^\circ\text{C} - 46^\circ\text{C} = 34^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60^\circ\text{C} - 25^\circ\text{C} = 35^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{34 - 35}{\ln(34 / 35)} = 34.5^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{46 - 25}{80 - 25} = 0.38 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80 - 60}{46 - 25} = 0.95 \end{aligned} \right\} F = 0.94$$



Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = UA_s F \Delta T_{lm,CF} \longrightarrow A_s = \frac{\dot{Q}}{UF \Delta T_{lm,CF}} = \frac{420 \text{ kW}}{(1.0 \text{ kW/m}^2 \cdot ^\circ\text{C})(0.94)(34.5^\circ\text{C})} = \mathbf{13.0 \text{ m}^2}$$

16-46 Steam is condensed by cooling water in the condenser of a power plant. The mass flow rate of the cooling water and the rate of condensation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The heat of vaporization of water at 50°C is given to be $h_{fg} = 2383 \text{ kJ/kg}$ and specific heat of cold water at the average temperature of 22.5°C is given to be $c_p = 4180 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis The temperature differences between the steam and the cooling water at the two ends of the condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 50^\circ\text{C} - 27^\circ\text{C} = 23^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 50^\circ\text{C} - 18^\circ\text{C} = 32^\circ\text{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{23 - 32}{\ln(23 / 32)} = 27.3^\circ\text{C}$$

Then the heat transfer rate in the condenser becomes

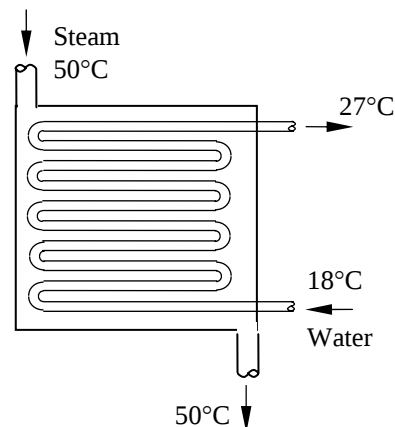
$$\dot{Q} = UA_s \Delta T_{lm} = (2400 \text{ W/m}^2 \cdot ^\circ\text{C})(42 \text{ m}^2)(27.3^\circ\text{C}) = 2752 \text{ kW}$$

The mass flow rate of the cooling water and the rate of condensation of steam are determined from

$$\dot{Q} = [\dot{m} c_p (T_{out} - T_{in})]_{\text{cooling water}}$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{2752 \text{ kJ/s}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(27^\circ\text{C} - 18^\circ\text{C})} = \mathbf{73.1 \text{ kg/s}}$$

$$\dot{Q} = (\dot{m} h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{2752 \text{ kJ/s}}{2383 \text{ kJ/kg}} = \mathbf{1.15 \text{ kg/s}}$$



16-47 Water is heated in a double-pipe parallel-flow heat exchanger by geothermal water. The required length of tube is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg·°C, respectively.

Analysis The rate of heat transfer in the heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (0.2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(60^\circ\text{C} - 25^\circ\text{C}) = 29.26 \text{ kW}$$

Then the outlet temperature of the geothermal water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{geot. water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 140^\circ\text{C} - \frac{29.26 \text{ kW}}{(0.3 \text{ kg/s})(4.31 \text{ kJ/kg} \cdot ^\circ\text{C})} = 117.4^\circ\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,in} = 140^\circ\text{C} - 25^\circ\text{C} = 115^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = 117.4^\circ\text{C} - 60^\circ\text{C} = 57.4^\circ\text{C}$$

and

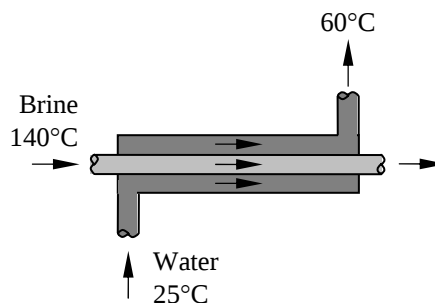
$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{115 - 57.4}{\ln(115 / 57.4)} = 82.9^\circ\text{C}$$

The surface area of the heat exchanger is determined from

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{29.26 \text{ kW}}{(0.55 \text{ kW/m}^2)(82.9^\circ\text{C})} = 0.642 \text{ m}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.642 \text{ m}^2}{\pi(0.008 \text{ m})} = \mathbf{25.5 \text{ m}}$$



16-48 EES Prob. 16-47 is reconsidered. The effects of temperature and mass flow rate of geothermal water on the length of the tube are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

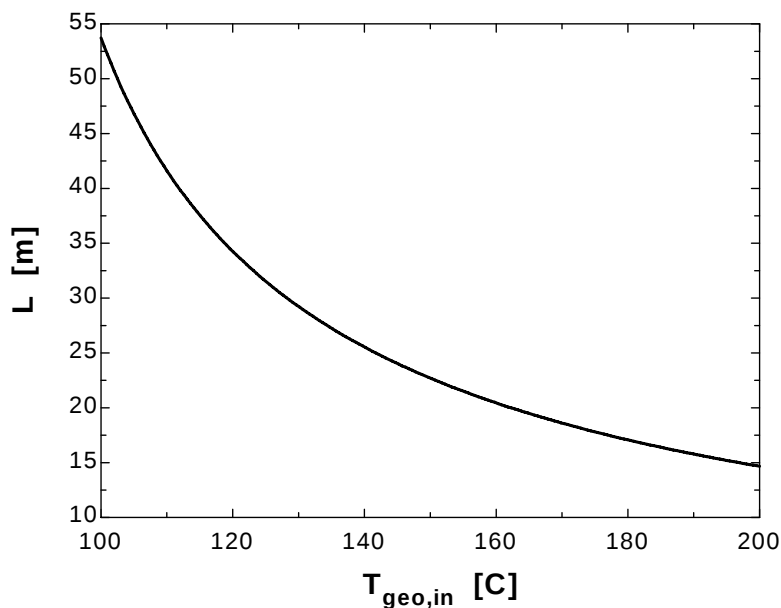
"GIVEN"

T_w_in=25 [C]
 T_w_out=60 [C]
 m_dot_w=0.2 [kg/s]
 c_p_w=4.18 [kJ/kg-C]
 T_geo_in=140 [C]
 m_dot_geo=0.3 [kg/s]
 c_p_geo=4.31 [kJ/kg-C]
 D=0.008 [m]
 U=0.55 [kW/m^2-C]

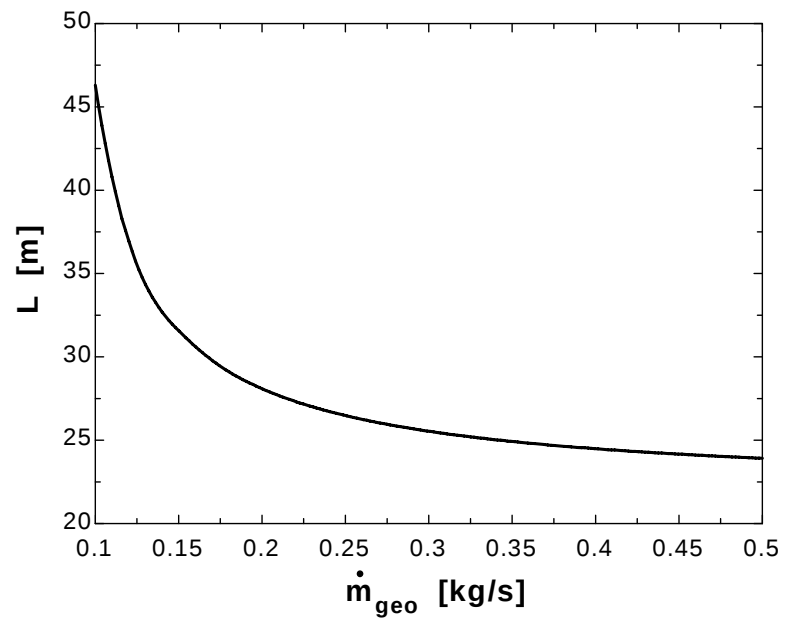
"ANALYSIS"

Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
 Q_dot=m_dot_geo*c_p_geo*(T_geo_in-T_geo_out)
 DELTAT_1=T_geo_in-T_w_in
 DELTAT_2=T_geo_out-T_w_out
 DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
 Q_dot=U*A*DELTAT_lm
 A=pi*D*L

T _{geo,in} [C]	L [m]
100	53.73
105	46.81
110	41.62
115	37.56
120	34.27
125	31.54
130	29.24
135	27.26
140	25.54
145	24.04
150	22.7
155	21.51
160	20.45
165	19.48
170	18.61
175	17.81
180	17.08
185	16.4
190	15.78
195	15.21
200	14.67



$\dot{m}_{\text{geo}}$ [kg/s]	L [m]
0.1	46.31
0.125	35.52
0.15	31.57
0.175	29.44
0.2	28.1
0.225	27.16
0.25	26.48
0.275	25.96
0.3	25.54
0.325	25.21
0.35	24.93
0.375	24.69
0.4	24.49
0.425	24.32
0.45	24.17
0.475	24.04
0.5	23.92



16-49E Glycerin is heated by hot water in a 1-shell pass and 8-tube passes heat exchanger. The rate of heat transfer for the cases of fouling and no fouling are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Heat transfer coefficients and fouling factors are constant and uniform. 5 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of glycerin and water are given to be 0.60 and 1.0 Btu/lbm.°F, respectively.

Analysis (a) The tubes are thin walled and thus we assume the inner surface area of the tube to be equal to the outer surface area. Then the heat transfer surface area of this heat exchanger becomes

$$A_s = n\pi DL = 8\pi(0.5/12 \text{ ft})(500 \text{ ft}) = 523.6 \text{ ft}^2$$

The temperature differences at the two ends of the heat exchanger are

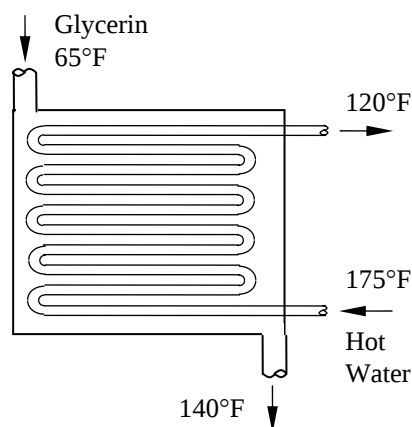
$$\Delta T_1 = T_{h,in} - T_{c,out} = 175^\circ\text{F} - 140^\circ\text{F} = 35^\circ\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 120^\circ\text{F} - 65^\circ\text{F} = 55^\circ\text{F}$$

$$\text{and } \Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{35 - 55}{\ln(35/55)} = 44.25^\circ\text{F}$$

The correction factor is

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{120 - 175}{65 - 175} = 0.5 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{65 - 140}{120 - 175} = 1.36 \end{aligned} \right\} F = 0.70$$



In case of no fouling, the overall heat transfer coefficient is determined from

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{50 \text{ Btu/h.ft}^2.\text{°F}} + \frac{1}{4 \text{ Btu/h.ft}^2.\text{°F}}} = 3.7 \text{ Btu/h.ft}^2.\text{°F}$$

Then the rate of heat transfer becomes

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (3.7 \text{ Btu/h.ft}^2.\text{°F})(523.6 \text{ ft}^2)(0.70)(44.25^\circ\text{F}) = \mathbf{60,000 \text{ Btu/h}}$$

(b) The thermal resistance of the heat exchanger with a fouling factor is

$$\begin{aligned} R &= \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{1}{h_o A_o} \\ &= \frac{1}{(50 \text{ Btu/h.ft}^2.\text{°F})(523.6 \text{ ft}^2)} + \frac{0.002 \text{ h.ft}^2.\text{°F/Btu}}{523.6 \text{ ft}^2} + \frac{1}{(4 \text{ Btu/h.ft}^2.\text{°F})(523.6 \text{ ft}^2)} \\ &= 0.0005195 \text{ h.°F/Btu} \end{aligned}$$

The overall heat transfer coefficient in this case is

$$R = \frac{1}{UA_s} \longrightarrow U = \frac{1}{RA_s} = \frac{1}{(0.0005195 \text{ h.°F/Btu})(523.6 \text{ ft}^2)} = 3.68 \text{ Btu/h.ft}^2.\text{°F}$$

Then rate of heat transfer becomes

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (3.68 \text{ Btu/h.ft}^2.\text{°F})(523.6 \text{ ft}^2)(0.70)(44.25^\circ\text{F}) = \mathbf{59,680 \text{ Btu/h}}$$

16-50 During an experiment, the inlet and exit temperatures of water and oil and the mass flow rate of water are measured. The overall heat transfer coefficient based on the inner surface area is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4180 and 2150 J/kg.°C, respectively.

Analysis The rate of heat transfer from the oil to the water is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (3 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(55^\circ\text{C} - 20^\circ\text{C}) = 438.9 \text{ kW}$$

The heat transfer area on the tube side is

$$A_i = n\pi D_i L = 24\pi(0.012 \text{ m})(2 \text{ m}) = 1.8 \text{ m}^2$$

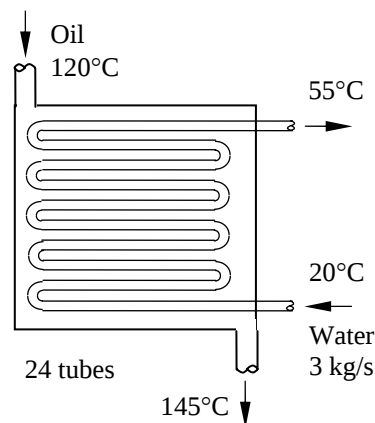
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 120^\circ\text{C} - 55^\circ\text{C} = 65^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 45^\circ\text{C} - 20^\circ\text{C} = 25^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{65 - 25}{\ln(65 / 25)} = 41.9^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{55 - 20}{120 - 20} = 0.35 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{120 - 45}{55 - 20} = 2.14 \end{aligned} \right\} F = 0.70$$



Then the overall heat transfer coefficient becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{lm,CF}} = \frac{438.9 \text{ kW}}{(1.8 \text{ m}^2)(0.70)(41.9^\circ\text{C})} = 8.31 \text{ kW/m}^2 \cdot ^\circ\text{C}$$

16-51 Ethylene glycol is cooled by water in a double-pipe counter-flow heat exchanger. The rate of heat transfer, the mass flow rate of water, and the heat transfer surface area on the inner side of the tubes are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

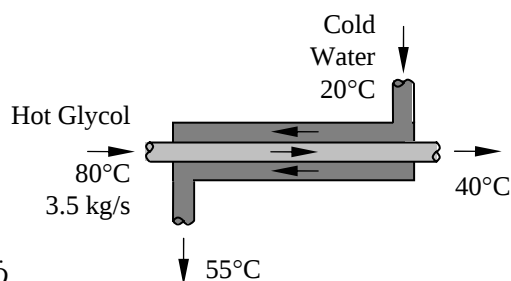
Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg·°C, respectively.

Analysis (a) The rate of heat transfer is

$$\begin{aligned}\dot{Q} &= [\dot{m} c_p (T_{in} - T_{out})]_{\text{glycol}} \\ &= (3.5 \text{ kg/s})(2.56 \text{ kJ/kg} \cdot ^\circ\text{C})(80^\circ\text{C} - 40^\circ\text{C}) \\ &= \mathbf{358.4 \text{ kW}}\end{aligned}$$

(b) The rate of heat transfer from water must be equal to the rate of heat transfer to the glycol. Then,

$$\begin{aligned}\dot{Q} &= [\dot{m} c_p (T_{out} - T_{in})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} \\ &= \frac{358.4 \text{ kJ/s}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(55^\circ\text{C} - 20^\circ\text{C})} = \mathbf{2.45 \text{ kg/s}}\end{aligned}$$



(c) The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80^\circ\text{C} - 55^\circ\text{C} = 25^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40^\circ\text{C} - 20^\circ\text{C} = 20^\circ\text{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25 - 20}{\ln(25 / 20)} = 22.4^\circ\text{C}$$

Then the heat transfer surface area becomes

$$\dot{Q} = U_i A_i \Delta T_{lm} \longrightarrow A_i = \frac{\dot{Q}}{U_i \Delta T_{lm}} = \frac{358.4 \text{ kW}}{(0.25 \text{ kW/m}^2 \cdot ^\circ\text{C})(22.4^\circ\text{C})} = \mathbf{64.0 \text{ m}^2}$$

16-52 Water is heated by steam in a double-pipe counter-flow heat exchanger. The required length of the tubes is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heat of water is given to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. The heat of condensation of steam at 120°C is given to be 2203 kJ/kg .

Analysis The rate of heat transfer is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} \\ &= (3 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(80^\circ\text{C} - 17^\circ\text{C}) \\ &= 790.02 \text{ kW}\end{aligned}$$

The logarithmic mean temperature difference is

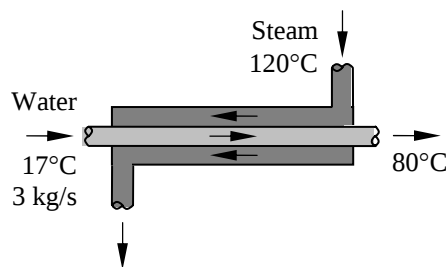
$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out} = 120^\circ\text{C} - 80^\circ\text{C} = 40^\circ\text{C} \\ \Delta T_2 &= T_{h,in} - T_{c,in} = 120^\circ\text{C} - 17^\circ\text{C} = 103^\circ\text{C} \\ \Delta T_{lm} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 103}{\ln(40 / 103)} = 66.6^\circ\text{C}\end{aligned}$$

The heat transfer surface area is

$$\dot{Q} = U_i A_i \Delta T_{lm} \longrightarrow A_i = \frac{\dot{Q}}{U_i \Delta T_{lm}} = \frac{790.02 \text{ kW}}{(1.5 \text{ kW} / \text{m}^2 \cdot ^\circ\text{C})(66.6^\circ\text{C})} = 7.9 \text{ m}^2$$

Then the length of tube required becomes

$$A_i = \pi D_i L \longrightarrow L = \frac{A_i}{\pi D_i} = \frac{7.9 \text{ m}^2}{\pi(0.025 \text{ m})} = \mathbf{100.6 \text{ m}}$$



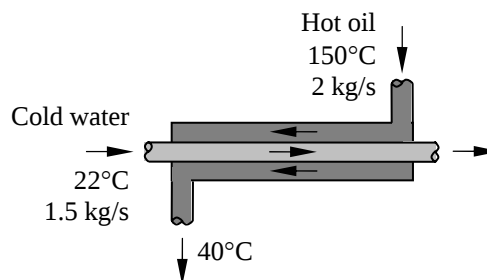
16-53 Oil is cooled by water in a thin-walled double-pipe counter-flow heat exchanger. The overall heat transfer coefficient of the heat exchanger is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant. 6 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg·°C, respectively.

Analysis The rate of heat transfer from the water to the oil is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{in} - T_{out})]_{oil} \\ &= (2 \text{ kg/s})(2.2 \text{ kJ/kg} \cdot ^\circ\text{C})(150^\circ\text{C} - 40^\circ\text{C}) \\ &= 484 \text{ kW}\end{aligned}$$



The outlet temperature of the water is determined from

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} \\ &= 22^\circ\text{C} + \frac{484 \text{ kW}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})} = 99.2^\circ\text{C}\end{aligned}$$

The logarithmic mean temperature difference is

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out} = 150^\circ\text{C} - 99.2^\circ\text{C} = 50.8^\circ\text{C} \\ \Delta T_2 &= T_{h,out} - T_{c,in} = 40^\circ\text{C} - 22^\circ\text{C} = 18^\circ\text{C} \\ \Delta T_{lm} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{50.8 - 18}{\ln(50.8 / 18)} = 31.6^\circ\text{C}\end{aligned}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{\dot{Q}}{A_s \Delta T_{lm}} = \frac{484 \text{ kW}}{\pi(0.025 \text{ m})(6 \text{ m})(31.6^\circ\text{C})} = \mathbf{32.5 \text{ kW/m}^2 \cdot ^\circ\text{C}}$$

16-54 EES Prob. 16-53 is reconsidered. The effects of oil exit temperature and water inlet temperature on the overall heat transfer coefficient of the heat exchanger are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

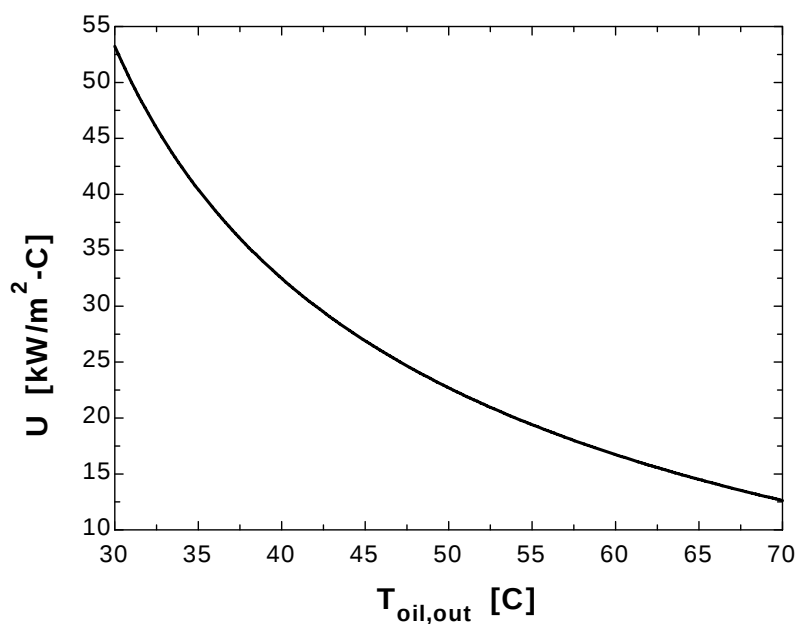
"GIVEN"

T_oil_in=150 [C]
 T_oil_out=40 [C]
 m_dot_oil=2 [kg/s]
 c_p_oil=2.20 [kJ/kg-C]
 T_w_in=22 [C]
 m_dot_w=1.5 [kg/s]
 C_p_w=4.18 [kJ/kg-C]
 D=0.025 [m]
 L=6 [m]

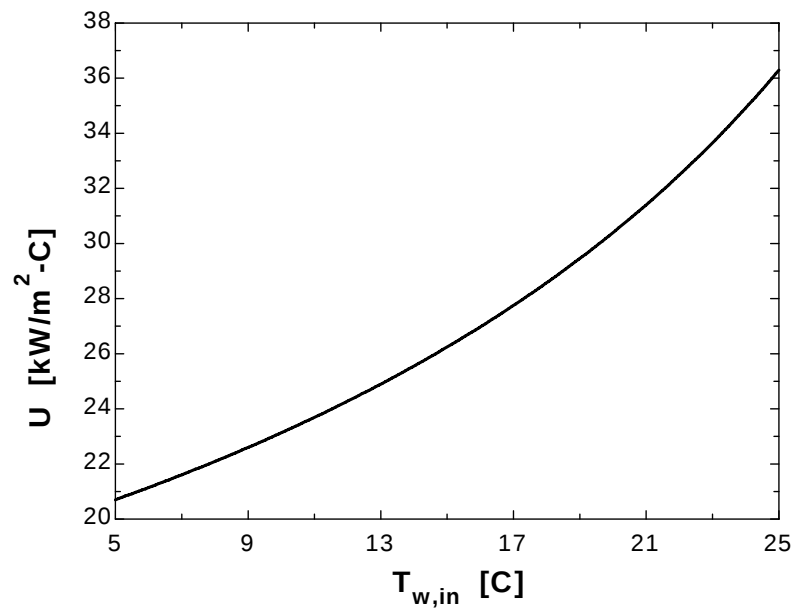
"ANALYSIS"

Q_dot=m_dot_oil*c_p_oil*(T_oil_in-T_oil_out)
 Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
 DELTAT_1=T_oil_in-T_w_out
 DELTAT_2=T_oil_out-T_w_in
 DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
 Q_dot=U*A*DELTAT_lm
 A=pi*D*L

T _{oil,out} [C]	U [kW/m ² -C]
30	53.22
32.5	45.94
35	40.43
37.5	36.07
40	32.49
42.5	29.48
45	26.9
47.5	24.67
50	22.7
52.5	20.96
55	19.4
57.5	18
60	16.73
62.5	15.57
65	14.51
67.5	13.53
70	12.63



$T_{w,in}$ [C]	U [kW/m ² C]
5	20.7
6	21.15
7	21.61
8	22.09
9	22.6
10	23.13
11	23.69
12	24.28
13	24.9
14	25.55
15	26.24
16	26.97
17	27.75
18	28.58
19	29.46
20	30.4
21	31.4
22	32.49
23	33.65
24	34.92
25	36.29



16-55 The inlet and outlet temperatures of the cold and hot fluids in a double-pipe heat exchanger are given. It is to be determined whether this is a parallel-flow or counter-flow heat exchanger.

Analysis In parallel-flow heat exchangers, the temperature of the cold water can never exceed that of the hot fluid. In this case $T_{\text{cold out}} = 50^\circ\text{C}$ which is greater than $T_{\text{hot out}} = 45^\circ\text{C}$. Therefore this must be a counter-flow heat exchanger.

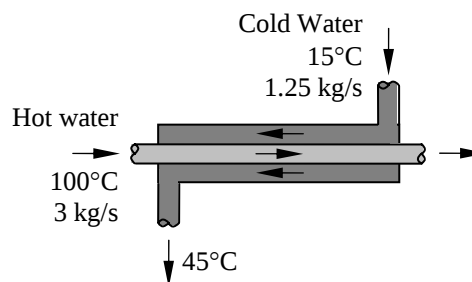
16-56 Cold water is heated by hot water in a double-pipe counter-flow heat exchanger. The rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant. 6 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg·°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cold water}} \\ &= (1.25 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(45^\circ\text{C} - 15^\circ\text{C}) \\ &= \mathbf{156.8 \text{ kW}}\end{aligned}$$



The outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{hot water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 100^\circ\text{C} - \frac{156.8 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg}\cdot^\circ\text{C})} = 87.5^\circ\text{C}$$

The temperature differences at the two ends of the heat exchanger are

$$\begin{aligned}\Delta T_1 &= T_{h,\text{in}} - T_{c,\text{out}} = 100^\circ\text{C} - 45^\circ\text{C} = 55^\circ\text{C} \\ \Delta T_2 &= T_{h,\text{out}} - T_{c,\text{in}} = 87.5^\circ\text{C} - 15^\circ\text{C} = 72.5^\circ\text{C}\end{aligned}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{55 - 72.5}{\ln(55 / 72.5)} = 63.3^\circ\text{C}$$

Then the surface area of this heat exchanger becomes

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{156.8 \text{ kW}}{(0.880 \text{ kW/m}^2\cdot^\circ\text{C})(63.3^\circ\text{C})} = \mathbf{2.81 \text{ m}^2}$$

16-57 Engine oil is heated by condensing steam in a condenser. The rate of heat transfer and the length of the tube required are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant. 6 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heat of engine oil is given to be $2.1 \text{ kJ/kg} \cdot ^\circ\text{C}$. The heat of condensation of steam at 130°C is given to be 2174 kJ/kg .

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{oil}} = (0.3 \text{ kg/s})(2.1 \text{ kJ/kg} \cdot ^\circ\text{C})(60^\circ\text{C} - 20^\circ\text{C}) = \mathbf{25.2 \text{ kW}}$$

The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,\text{in}} - T_{c,\text{out}} = 130^\circ\text{C} - 60^\circ\text{C} = 70^\circ\text{C}$$

$$\Delta T_2 = T_{h,\text{out}} - T_{c,\text{in}} = 130^\circ\text{C} - 20^\circ\text{C} = 110^\circ\text{C}$$

and

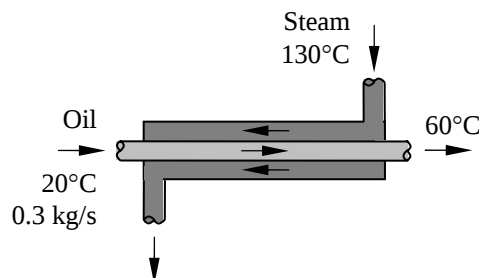
$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{70 - 110}{\ln(70 / 110)} = 88.5^\circ\text{C}$$

The surface area is

$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{25.2 \text{ kW}}{(0.65 \text{ kW/m}^2 \cdot ^\circ\text{C})(88.5^\circ\text{C})} = 0.44 \text{ m}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.44 \text{ m}^2}{\pi(0.02 \text{ m})} = \mathbf{7.0 \text{ m}}$$



16-58E Water is heated by geothermal water in a double-pipe counter-flow heat exchanger. The mass flow rate of each fluid and the total thermal resistance of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of water and geothermal fluid are given to be 1.0 and 1.03 Btu/lbm.°F, respectively.

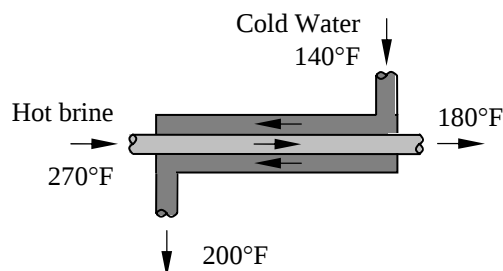
Analysis The mass flow rate of each fluid is determined from

$$\dot{Q} = [\dot{m} c_p (T_{out} - T_{in})]_{\text{water}}$$

$$\dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{40 \text{ Btu/s}}{(1.0 \text{ Btu/lbm.}^\circ\text{F})(200^\circ\text{F} - 140^\circ\text{F})} = \mathbf{0.667 \text{ lbm/s}}$$

$$\dot{Q} = [\dot{m} c_p (T_{out} - T_{in})]_{\text{geo. water}}$$

$$\dot{m}_{\text{geo. water}} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{40 \text{ Btu/s}}{(1.03 \text{ Btu/lbm.}^\circ\text{F})(270^\circ\text{F} - 180^\circ\text{F})} = \mathbf{0.431 \text{ lbm/s}}$$



The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 270^\circ\text{F} - 200^\circ\text{F} = 70^\circ\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 180^\circ\text{F} - 140^\circ\text{F} = 40^\circ\text{F}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{70 - 40}{\ln(70 / 40)} = 53.61^\circ\text{F}$$

Then

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow UA_s = \frac{\dot{Q}}{\Delta T_{lm}} = \frac{40 \text{ Btu/s}}{53.61^\circ\text{F}} = 0.7462 \text{ Btu/s.}^\circ\text{F}$$

$$U = \frac{1}{RA_s} \longrightarrow R = \frac{1}{UA_s} = \frac{1}{0.7462 \text{ Btu/s.}^\circ\text{F}} = \mathbf{1.34 \text{ s.}^\circ\text{F/Btu}}$$

16-59 Glycerin is heated by ethylene glycol in a thin-walled double-pipe parallel-flow heat exchanger. The rate of heat transfer, the outlet temperature of the glycerin, and the mass flow rate of the ethylene glycol are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant. 6 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of glycerin and ethylene glycol are given to be 2.4 and 2.5 kJ/kg·°C, respectively.

Analysis (a) The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,in} = 60^\circ\text{C} - 20^\circ\text{C} = 40^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = T_{h,out} - (T_{h,out} - 15^\circ\text{C}) = 15^\circ\text{C}$$

$$\text{and } \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 15}{\ln(40 / 15)} = 25.5^\circ\text{C}$$

Then the rate of heat transfer becomes

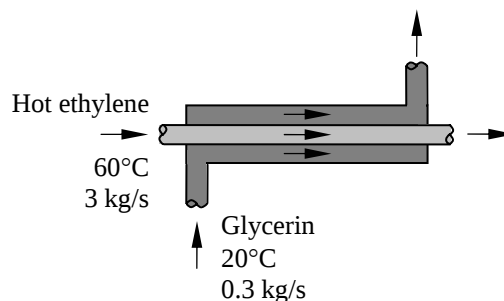
$$\dot{Q} = UA_s \Delta T_{lm} = (240 \text{ W/m}^2 \cdot ^\circ\text{C})(3.2 \text{ m}^2)(25.5^\circ\text{C}) = 19,584 \text{ W} = \mathbf{19.58 \text{ kW}}$$

(b) The outlet temperature of the glycerin is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{glycerin}} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 20^\circ\text{C} + \frac{19.584 \text{ kW}}{(0.3 \text{ kg/s})(2.4 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{47.2^\circ\text{C}}$$

(c) Then the mass flow rate of ethylene glycol becomes

$$\begin{aligned} \dot{Q} &= [\dot{m}c_p(T_{in} - T_{out})]_{\text{ethylene glycol}} \\ \dot{m}_{\text{ethylene glycol}} &= \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{19.584 \text{ kJ/s}}{(2.5 \text{ kJ/kg} \cdot ^\circ\text{C})[(47.2 + 15)^\circ\text{C} - 60^\circ\text{C}]} = \mathbf{3.56 \text{ kg/s}} \end{aligned}$$



16-60 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1005 and 1100 J/kg·°C, respectively.

Analysis The rate of heat transfer is

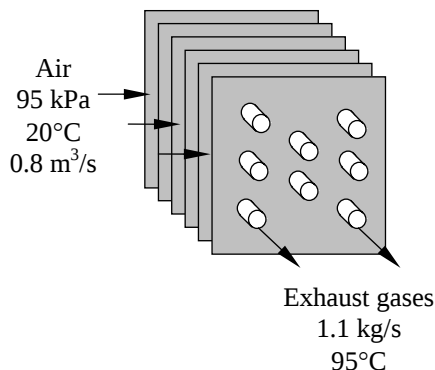
$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{in} - T_{out})]_{\text{gas}} \\ &= (1.1 \text{ kg/s})(1.1 \text{ kJ/kg} \cdot ^\circ\text{C})(180^\circ\text{C} - 95^\circ\text{C}) \\ &= \mathbf{103 \text{ kW}}\end{aligned}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95 \text{ kPa})(0.8 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}) \times 293 \text{ K}} = 0.904 \text{ kg/s}$$

Then the outlet temperature of the air becomes

$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}c_p} = 20^\circ\text{C} + \frac{103 \times 10^3 \text{ W}}{(0.904 \text{ kg/s})(1005 \text{ J/kg} \cdot ^\circ\text{C})} = \mathbf{133^\circ\text{C}}$$



16-61 Water is heated by hot oil in a 2-shell passes and 12-tube passes heat exchanger. The heat transfer surface area on the tube side is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(70^\circ\text{C} - 20^\circ\text{C}) = 940.5 \text{ kW}$$

The outlet temperature of the oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170^\circ\text{C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg} \cdot ^\circ\text{C})} = 129^\circ\text{C}$$

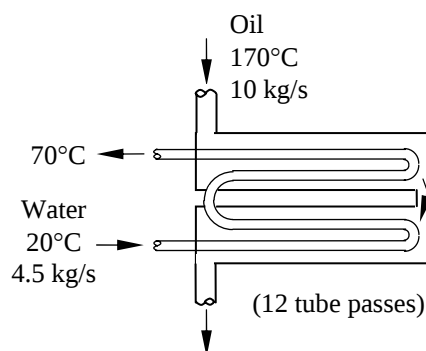
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 170^\circ\text{C} - 70^\circ\text{C} = 100^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 129^\circ\text{C} - 20^\circ\text{C} = 109^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{100 - 109}{\ln(100 / 109)} = 104.4^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 20}{170 - 20} = 0.33 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{170 - 129}{70 - 20} = 0.82 \end{aligned} \right\} F = 1.0$$



Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = UA_s F \Delta T_{lm,CF} \longrightarrow A_s = \frac{\dot{Q}}{U F \Delta T_{lm,CF}} = \frac{940.5 \text{ kW}}{(0.350 \text{ kW/m}^2 \cdot ^\circ\text{C})(1.0)(104.4^\circ\text{C})} = 25.7 \text{ m}^2$$

16-62 Water is heated by hot oil in a 2-shell passes and 12-tube passes heat exchanger. The heat transfer surface area on the tube side is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(70^\circ\text{C} - 20^\circ\text{C}) = 418 \text{ kW}$$

The outlet temperature of the oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170^\circ\text{C} - \frac{418 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg} \cdot ^\circ\text{C})} = 151.8^\circ\text{C}$$

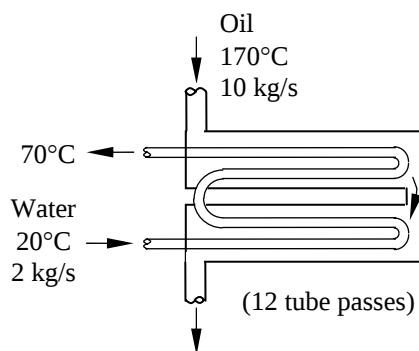
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 170^\circ\text{C} - 70^\circ\text{C} = 100^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 151.8^\circ\text{C} - 20^\circ\text{C} = 131.8^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{100 - 131.8}{\ln(100 / 131.8)} = 115.2^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 20}{170 - 20} = 0.33 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{170 - 151.8}{70 - 20} = 0.36 \end{aligned} \right\} F = 1.0$$



Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{418 \text{ kW}}{(0.350 \text{ kW/m}^2 \cdot ^\circ\text{C})(1.0)(115.2^\circ\text{C})} = \mathbf{10.4 \text{ m}^2}$$

16-63 Ethyl alcohol is heated by water in a 2-shell passes and 8-tube passes heat exchanger. The heat transfer surface area of the heat exchanger is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of water and ethyl alcohol are given to be 4.19 and 2.67 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{ethyl alcohol}} = (2.1 \text{ kg/s})(2.67 \text{ kJ/kg} \cdot ^\circ\text{C})(70^\circ\text{C} - 25^\circ\text{C}) = 252.3 \text{ kW}$$

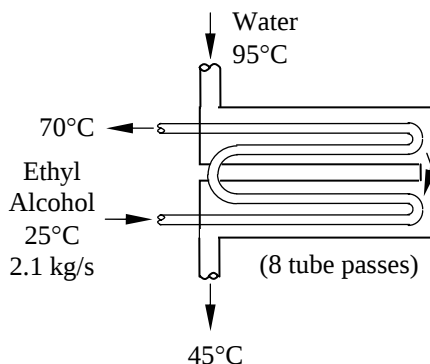
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 95^\circ\text{C} - 70^\circ\text{C} = 25^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 45^\circ\text{C} - 25^\circ\text{C} = 20^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25 - 20}{\ln(25 / 20)} = 22.4^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 25}{95 - 25} = 0.64 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{95 - 45}{70 - 25} = 1.1 \end{aligned} \right\} F = 0.82$$



Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{252.3 \text{ kW}}{(0.950 \text{ kW/m}^2 \cdot ^\circ\text{C})(0.82)(22.4^\circ\text{C})} = \mathbf{14.5 \text{ m}^2}$$

16-64 Water is heated by ethylene glycol in a 2-shell passes and 12-tube passes heat exchanger. The rate of heat transfer and the heat transfer surface area on the tube side are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.68 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is :

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (0.8 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(70^\circ\text{C} - 22^\circ\text{C}) = \mathbf{160.5 \text{ kW}}$$

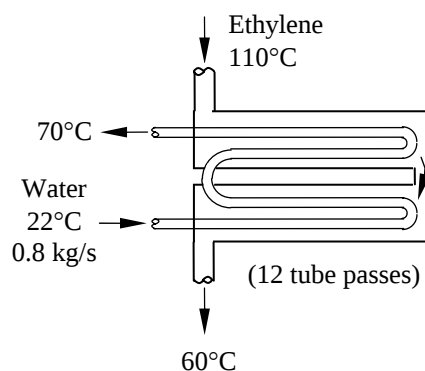
The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 110^\circ\text{C} - 70^\circ\text{C} = 40^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60^\circ\text{C} - 22^\circ\text{C} = 38^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 38}{\ln(40 / 38)} = 39^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 22}{110 - 22} = 0.55 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{110 - 60}{70 - 22} = 1.04 \end{aligned} \right\} F = 0.92$$



Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{160.5 \text{ kW}}{(0.28 \text{ kW/m}^2 \cdot ^\circ\text{C})(0.92)(39^\circ\text{C})} = \mathbf{16.0 \text{ m}^2}$$

16-65 EES Prob. 16-64 is reconsidered. The effect of the mass flow rate of water on the rate of heat transfer and the tube-side surface area is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

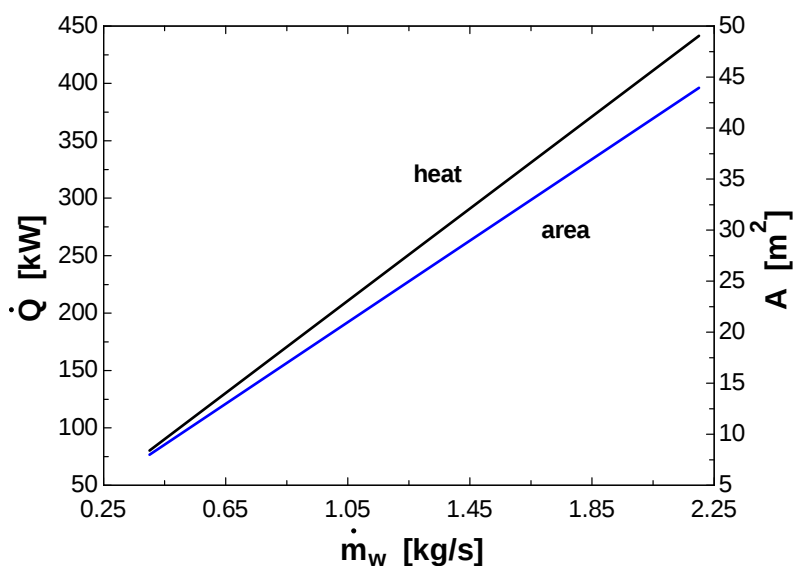
"GIVEN"

T_w_in=22 [C]
 T_w_out=70 [C]
 m_dot_w=0.8 [kg/s]
 c_p_w=4.18 [kJ/kg-C]
 T_glycol_in=110 [C]
 T_glycol_out=60 [C]
 c_p_glycol=2.68 [kJ/kg-C]
 U=0.28 [kW/m^2-C]

"ANALYSIS"

Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
 Q_dot=m_dot_glycol*c_p_glycol*(T_glycol_in-T_glycol_out)
 DELTAT_1=T_glycol_in-T_w_out
 DELTAT_2=T_glycol_out-T_w_in
 DELTAT_lm_CF=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
 P=(T_w_out-T_w_in)/(T_glycol_in-T_w_in)
 R=(T_glycol_in-T_glycol_out)/(T_w_out-T_w_in)
 F=0.92 "from Fig. 16-18b of the text at the calculated P and R"
 Q_dot=U*A*F*DELTAT_lm_CF

m _w [kg/s]	Q [kW]	A [m ²]
0.4	80.26	7.99
0.5	100.3	9.988
0.6	120.4	11.99
0.7	140.4	13.98
0.8	160.5	15.98
0.9	180.6	17.98
1	200.6	19.98
1.1	220.7	21.97
1.2	240.8	23.97
1.3	260.8	25.97
1.4	280.9	27.97
1.5	301	29.96
1.6	321	31.96
1.7	341.1	33.96
1.8	361.2	35.96
1.9	381.2	37.95
2	401.3	39.95
2.1	421.3	41.95
2.2	441.4	43.95



16-66E Steam is condensed by cooling water in a condenser. The rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant. 6 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties We take specific heat of water are given to be 1.0 Btu/lbm.°F. The heat of condensation of steam at 90°F is 1043 Btu/lbm.

Analysis (a) The log mean temperature difference is determined from

$$\Delta T_1 = T_{h,in} - T_{c,out} = 90^\circ\text{F} - 73^\circ\text{F} = 17^\circ\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 90^\circ\text{F} - 60^\circ\text{F} = 30^\circ\text{F}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{17 - 30}{\ln(17 / 30)} = 22.9^\circ\text{F}$$

The heat transfer surface area is

$$A_s = 8n\pi DL = 8 \times 50 \times \pi (3/48 \text{ ft})(5 \text{ ft}) = 392.7 \text{ ft}^2$$

and

$$\dot{Q} = UA_s \Delta T_{lm} = (600 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F})(392.7 \text{ ft}^2)(22.9^\circ\text{F}) = \mathbf{5.396 \times 10^6 \text{ Btu/h}}$$

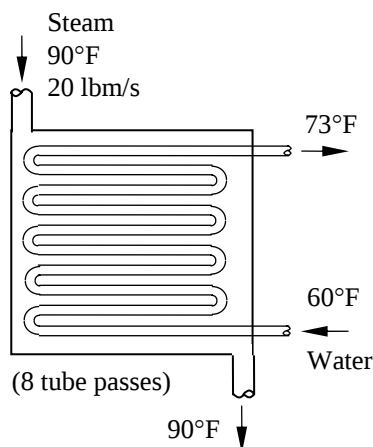
(b) The rate of condensation of the steam is

$$\dot{Q} = (\dot{m} h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{5.396 \times 10^6 \text{ Btu/h}}{1043 \text{ Btu/lbm}} = \mathbf{5173 \text{ lbm/h} = 1.44 \text{ lbm/s}}$$

(c) Then the mass flow rate of cold water becomes

$$\dot{Q} = [\dot{m} c_p (T_{out} - T_{in})]_{\text{cold water}}$$

$$\dot{m}_{\text{cold water}} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{5.396 \times 10^6 \text{ Btu/h}}{(1.0 \text{ Btu/lbm.}^\circ\text{F})(73^\circ\text{F} - 60^\circ\text{F})} = \mathbf{4.15 \times 10^5 \text{ lbm/h} = 115 \text{ lbm/s}}$$



16-67E EES Prob. 16-66E is reconsidered. The effect of the condensing steam temperature on the rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

N_pass=8
 N_tube=50
 T_steam=90 [F]
 h_fg_steam=1043 [Btu/lbm]
 T_w_in=60 [F]
 T_w_out=73 [F]
 c_p_w=1.0 [Btu/lbm-F]
 D=3/4*1/12 [ft]
 L=5 [ft]
 U=600 [Btu/h-ft^2-F]

"ANALYSIS"

"(a)"

DELTAT_1=T_steam-T_w_out
 DELTAT_2=T_steam-T_w_in
 DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
 A=N_pass*N_tube*pi*D*L
 Q_dot=U*A*DELTAT_lm*Convert(Btu/h, Btu/s)

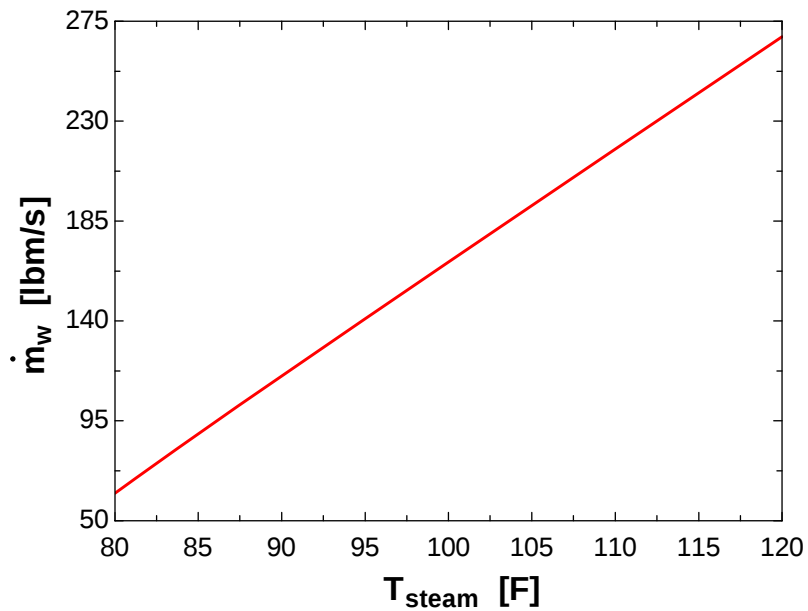
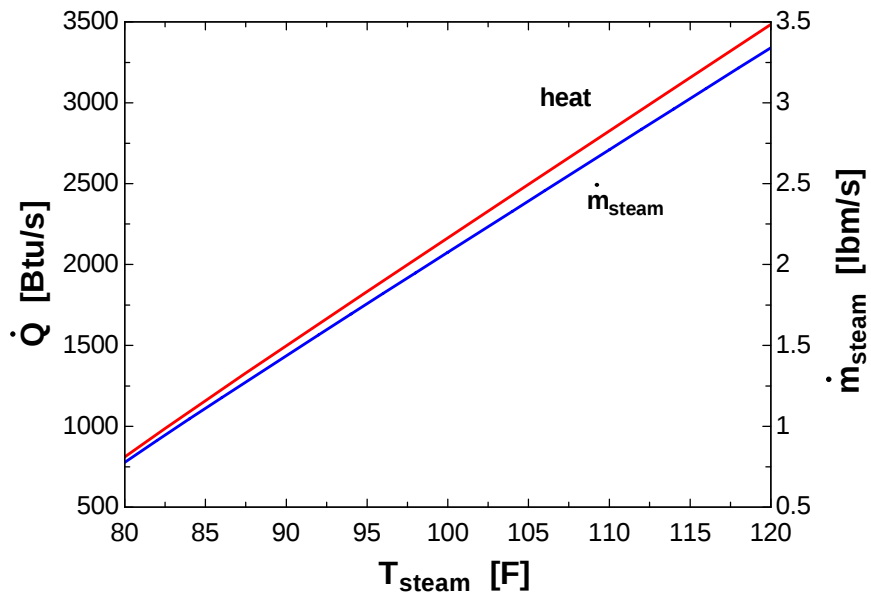
"(b)"

Q_dot=m_dot_steam*h_fg_steam

"(c)"

Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)

T _{steam} [F]	Q [Btu/s]	m _{steam} [lbm/s]	m _w [lbm/s]
80	810.5	0.7771	62.34
82	951.9	0.9127	73.23
84	1091	1.046	83.89
86	1228	1.177	94.42
88	1363	1.307	104.9
90	1498	1.436	115.2
92	1632	1.565	125.6
94	1766	1.693	135.8
96	1899	1.821	146.1
98	2032	1.948	156.3
100	2165	2.076	166.5
102	2297	2.203	176.7
104	2430	2.329	186.9
106	2562	2.456	197.1
108	2694	2.583	207.2
110	2826	2.709	217.4
112	2958	2.836	227.5
114	3089	2.962	237.6
116	3221	3.088	247.8
118	3353	3.214	257.9
120	3484	3.341	268



16-68 Glycerin is heated by hot water in a 1-shell pass and 20-tube passes heat exchanger. The mass flow rate of glycerin and the overall heat transfer coefficient of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heat of glycerin is given to be $2.48 \text{ kJ/kg} \cdot ^\circ\text{C}$ and that of water is taken to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m} c_p (T_{in} - T_{out})]_{\text{water}} = (0.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(100^\circ\text{C} - 55^\circ\text{C}) = 94.05 \text{ kW}$$

The mass flow rate of the glycerin is determined from

$$\dot{Q} = [\dot{m} c_p (T_{out} - T_{in})]_{\text{glycerin}}$$

$$\dot{m}_{\text{glycerin}} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{94.05 \text{ kJ/s}}{(2.48 \text{ kJ/kg} \cdot ^\circ\text{C})(55^\circ\text{C} - 15^\circ\text{C})} = \mathbf{0.95 \text{ kg/s}}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 100^\circ\text{C} - 55^\circ\text{C} = 45^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 55^\circ\text{C} - 15^\circ\text{C} = 40^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{45 - 40}{\ln(45 / 40)} = 42.5^\circ\text{C}$$

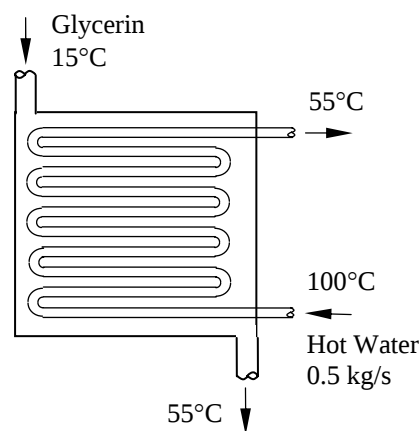
$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{55 - 100}{15 - 100} = 0.53 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{15 - 55}{55 - 100} = 0.89 \end{aligned} \right\} F = 0.77$$

The heat transfer surface area is

$$A_s = n\pi DL = 20\pi(0.04 \text{ m})(2 \text{ m}) = 5.027 \text{ m}^2$$

Then the overall heat transfer coefficient of the heat exchanger is determined to be

$$\dot{Q} = UA_s F \Delta T_{lm,CF} \longrightarrow U = \frac{\dot{Q}}{A_s F \Delta T_{lm,CF}} = \frac{94.05 \text{ kW}}{(5.027 \text{ m}^2)(0.77)(42.5^\circ\text{C})} = \mathbf{0.572 \text{ kW/m}^2 \cdot ^\circ\text{C}}$$



16-69 Isobutane is condensed by cooling air in the condenser of a power plant. The mass flow rate of air and the overall heat transfer coefficient are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The heat of vaporization of isobutane at 75°C is given to be $h_{fg} = 255.7 \text{ kJ/kg}$ and specific heat of air is taken to be $c_p = 1005 \text{ J/kg}\cdot^\circ\text{C}$.

Analysis First, the rate of heat transfer is determined from

$$\begin{aligned}\dot{Q} &= (\dot{m} h_{fg})_{\text{isobutane}} \\ &= (2.7 \text{ kg/s})(255.7 \text{ kJ/kg}) = 690.4 \text{ kW}\end{aligned}$$

The mass flow rate of air is determined from

$$\dot{Q} = [\dot{m} c_p (T_{\text{out}} - T_{\text{in}})]_{\text{air}} \longrightarrow \dot{m}_{\text{air}} = \frac{\dot{Q}}{c_p (T_{\text{out}} - T_{\text{in}})} = \frac{690.4 \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(28^\circ\text{C} - 21^\circ\text{C})} = \mathbf{98.1 \text{ kg/s}}$$

The temperature differences between the isobutane and the air at the two ends of the condenser are

$$\Delta T_1 = T_{h,\text{in}} - T_{c,\text{out}} = 75^\circ\text{C} - 21^\circ\text{C} = 54^\circ\text{C}$$

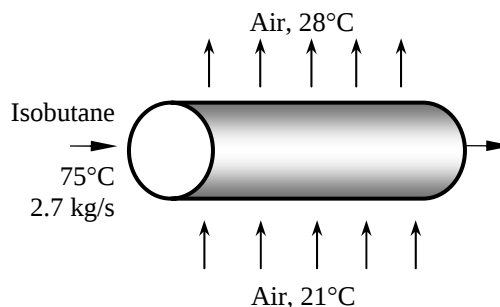
$$\Delta T_2 = T_{h,\text{out}} - T_{c,\text{in}} = 75^\circ\text{C} - 28^\circ\text{C} = 47^\circ\text{C}$$

and

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{54 - 47}{\ln(54 / 47)} = 50.4^\circ\text{C}$$

Then the overall heat transfer coefficient is determined from

$$\dot{Q} = UA_s \Delta T_{\text{lm}} \longrightarrow 690,400 \text{ W} = U(24 \text{ m}^2)(50.4^\circ\text{C}) \longrightarrow U = \mathbf{571 \text{ W/m}^2\cdot^\circ\text{C}}$$



16-70 Water is evaporated by hot exhaust gases in an evaporator. The rate of heat transfer, the exit temperature of the exhaust gases, and the rate of evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The heat of vaporization of water at 200°C is given to be $h_{fg} = 1941 \text{ kJ/kg}$ and specific heat of exhaust gases is given to be $c_p = 1051 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis The temperature differences between the water and the exhaust gases at the two ends of the evaporator are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 550^\circ\text{C} - 200^\circ\text{C} = 350^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = (T_{h,out} - 200)^\circ\text{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{350 - (T_{h,out} - 200)}{\ln[350 / (T_{h,out} - 200)]}$$

Then the rate of heat transfer can be expressed as

$$\dot{Q} = UA_s \Delta T_{lm} = (1.780 \text{ kW/m}^2 \cdot ^\circ\text{C})(0.5 \text{ m}^2) \frac{350 - (T_{h,out} - 200)}{\ln[350 / (T_{h,out} - 200)]} \quad (\text{Eq. 1})$$

The rate of heat transfer can also be expressed as in the following forms

$$\dot{Q} = [\dot{m} c_p (T_{h,in} - T_{h,out})]_{\text{exhaust gases}} = (0.25 \text{ kg/s})(1.051 \text{ kJ/kg} \cdot ^\circ\text{C})(550^\circ\text{C} - T_{h,out}) \quad (\text{Eq. 2})$$

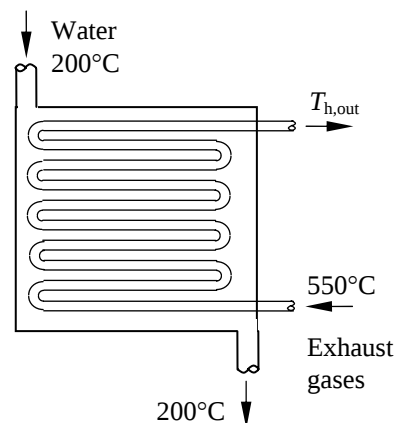
$$\dot{Q} = (\dot{m} h_{fg})_{\text{water}} = \dot{m}_{\text{water}} (1941 \text{ kJ/kg}) \quad (\text{Eq. 3})$$

We have three equations with three unknowns. Using an equation solver such as EES, the unknowns are determined to be

$$\dot{Q} = \mathbf{88.85 \text{ kW}}$$

$$T_{h,out} = \mathbf{211.8^\circ\text{C}}$$

$$\dot{m}_{\text{water}} = \mathbf{0.0458 \text{ kg/s}}$$



16-71 EES Prob. 16-70 is reconsidered. The effect of the exhaust gas inlet temperature on the rate of heat transfer, the exit temperature of exhaust gases, and the rate of evaporation of water is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

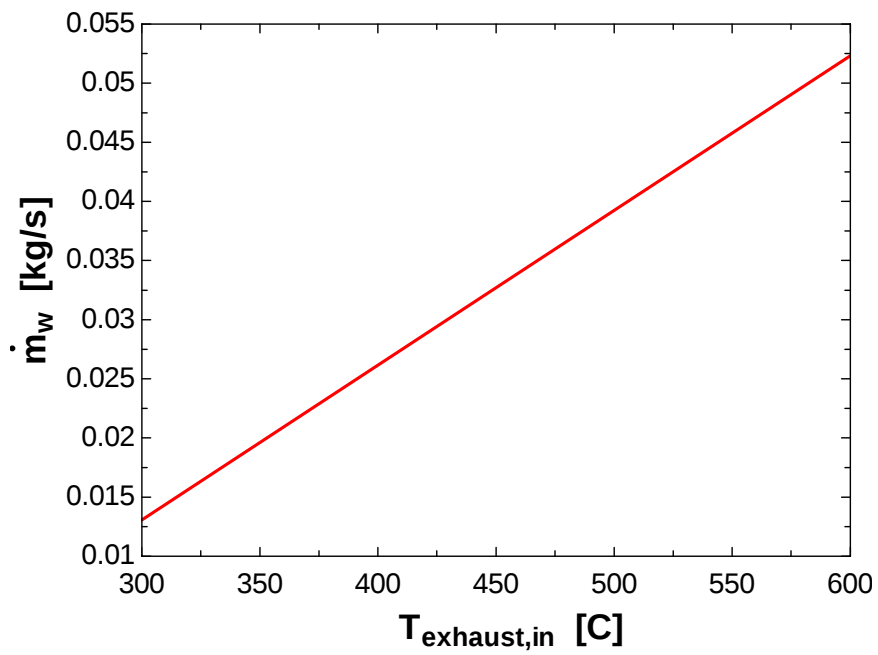
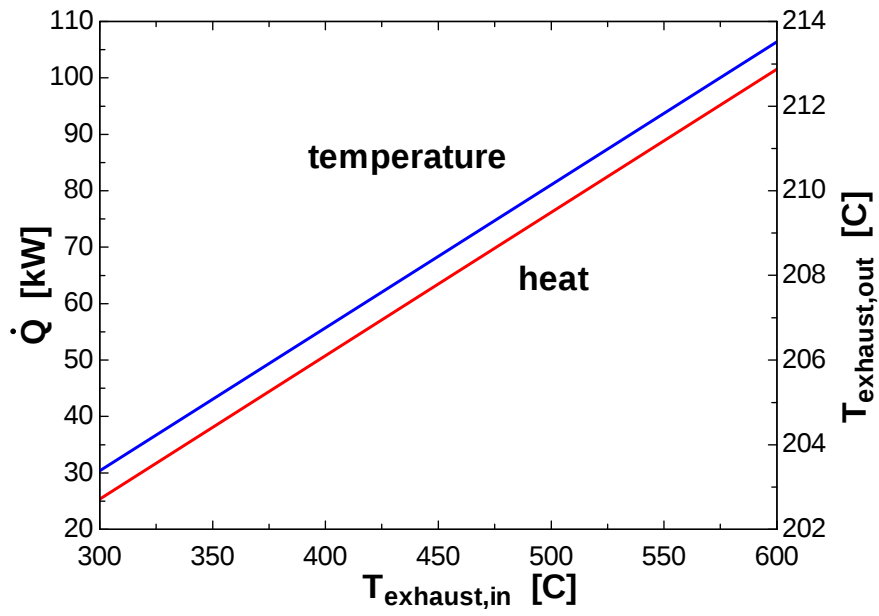
"GIVEN"

T_exhaust_in=550 [C]
 c_p_exhaust=1.051 [kJ/kg-C]
 m_dot_exhaust=0.25 [kg/s]
 T_w=200 [C]
 h_fg_w=1941 [kJ/kg]
 A=0.5 [m^2]
 U=1.780 [kW/m^2-C]

"ANALYSIS"

DELTAT_1=T_exhaust_in-T_w
 DELTAT_2=T_exhaust_out-T_w
 DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
 Q_dot=U*A*DELTAT_lm
 Q_dot=m_dot_exhaust*c_p_exhaust*(T_exhaust_in-T_exhaust_out)
 Q_dot=m_dot_w*h_fg_w

T _{exhaust,in} [C]	Q [kW]	T _{exhaust,out} [C]	m _w [kg/s]
300	25.39	203.4	0.01308
320	30.46	204.1	0.0157
340	35.54	204.7	0.01831
360	40.62	205.4	0.02093
380	45.7	206.1	0.02354
400	50.77	206.8	0.02616
420	55.85	207.4	0.02877
440	60.93	208.1	0.03139
460	66.01	208.8	0.03401
480	71.08	209.5	0.03662
500	76.16	210.1	0.03924
520	81.24	210.8	0.04185
540	86.32	211.5	0.04447
560	91.39	212.2	0.04709
580	96.47	212.8	0.0497
600	101.5	213.5	0.05232



16-72 The waste dyeing water is to be used to preheat fresh water. The outlet temperatures of each fluid and the mass flow rate are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The specific heats of waste dyeing water and the fresh water are given to be $c_p = 4295 \text{ J/kg} \cdot ^\circ\text{C}$ and $c_p = 4180 \text{ J/kg} \cdot ^\circ\text{C}$, respectively.

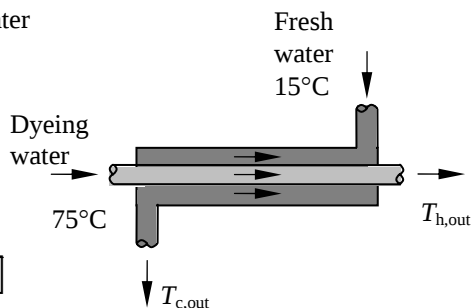
Analysis The temperature differences between the dyeing water and the fresh water at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,\text{in}} - T_{c,\text{out}} = 75 - T_{c,\text{out}}$$

$$\Delta T_2 = T_{h,\text{out}} - T_{c,\text{in}} = T_{h,\text{out}} - 15$$

and

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{(75 - T_{c,\text{out}}) - (T_{h,\text{out}} - 15)}{\ln[(75 - T_{c,\text{out}}) / (T_{h,\text{out}} - 15)]}$$



Then the rate of heat transfer can be expressed as

$$\begin{aligned} \dot{Q} &= UA_s \Delta T_{\text{lm}} \\ 35 \text{ kW} &= (0.625 \text{ kW/m}^2 \cdot ^\circ\text{C})(1.65 \text{ m}^2) \frac{(75 - T_{c,\text{out}}) - (T_{h,\text{out}} - 15)}{\ln[(75 - T_{c,\text{out}}) / (T_{h,\text{out}} - 15)]} \end{aligned} \quad (\text{Eq. 1})$$

The rate of heat transfer can also be expressed as

$$\dot{Q} = [\dot{m} c_p (T_{h,\text{in}} - T_{h,\text{out}})]_{\text{dyeing water}} \longrightarrow 35 \text{ kW} = \dot{m}(4.295 \text{ kJ/kg} \cdot ^\circ\text{C})(75^\circ\text{C} - T_{h,\text{out}}) \quad (\text{Eq. 2})$$

$$\dot{Q} = [\dot{m} c_p (T_{h,\text{in}} - T_{h,\text{out}})]_{\text{water}} \longrightarrow 35 \text{ kW} = \dot{m}(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(T_{c,\text{out}} - 15^\circ\text{C}) \quad (\text{Eq. 3})$$

We have three equations with three unknowns. Using an equation solver such as EES, the unknowns are determined to be

$$T_{c,\text{out}} = \mathbf{41.4^\circ\text{C}}$$

$$T_{h,\text{out}} = \mathbf{49.3^\circ\text{C}}$$

$$\dot{m} = \mathbf{0.317 \text{ kg/s}}$$

The Effectiveness-NTU Method

16-73C When the heat transfer surface area A of the heat exchanger is known, but the outlet temperatures are not, the effectiveness-NTU method is definitely preferred.

16-74C The effectiveness of a heat exchanger is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate and represents how closely the heat transfer in the heat exchanger approaches to maximum possible heat transfer. Since the actual heat transfer rate can not be greater than maximum possible heat transfer rate, the effectiveness can not be greater than one. The effectiveness of a heat exchanger depends on the geometry of the heat exchanger as well as the flow arrangement.

16-75C For a specified fluid pair, inlet temperatures and mass flow rates, the counter-flow heat exchanger will have the highest effectiveness.

16-76C Once the effectiveness ε is known, the rate of heat transfer and the outlet temperatures of cold and hot fluids in a heat exchanger are determined from

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = \varepsilon C_{\min} (T_{h,in} - T_{c,in})$$

$$\dot{Q} = \dot{m}_c c_{p,c} (T_{c,out} - T_{c,in})$$

$$\dot{Q} = \dot{m}_h c_{p,h} (T_{h,in} - T_{h,out})$$

16-77C The heat transfer in a heat exchanger will reach its maximum value when the hot fluid is cooled to the inlet temperature of the cold fluid. Therefore, the temperature of the hot fluid cannot drop below the inlet temperature of the cold fluid at any location in a heat exchanger.

16-78C The heat transfer in a heat exchanger will reach its maximum value when the cold fluid is heated to the inlet temperature of the hot fluid. Therefore, the temperature of the cold fluid cannot rise above the inlet temperature of the hot fluid at any location in a heat exchanger.

16-79C The fluid with the lower mass flow rate will experience a larger temperature change. This is clear from the relation

$$\dot{Q} = \dot{m}_c c_p \Delta T_{cold} = \dot{m}_h c_p \Delta T_{hot}$$

16-80C The maximum possible heat transfer rate in a heat exchanger is determined from

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in})$$

where $C_{\min}$ is the smaller heat capacity rate. The value of $\dot{Q}_{\max}$ does not depend on the type of heat exchanger.

16-81C The longer heat exchanger is more likely to have a higher effectiveness.

16-82C The increase of effectiveness with NTU is not linear. The effectiveness increases rapidly with NTU for small values (up to about NTU = 1.5), but rather slowly for larger values. Therefore, the effectiveness will not double when the length of heat exchanger is doubled.

16-83C A heat exchanger has the smallest effectiveness value when the heat capacity rates of two fluids are identical. Therefore, reducing the mass flow rate of cold fluid by half will increase its effectiveness.

16-84C When the capacity ratio is equal to zero and the number of transfer units value is greater than 5, a counter-flow heat exchanger has an effectiveness of one. In this case the exit temperature of the fluid with smaller capacity rate will equal to inlet temperature of the other fluid. For a parallel-flow heat exchanger the answer would be the same.

16-85C The NTU of a heat exchanger is defined as $NTU = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}c_p)_{\min}}$ where U is the overall heat transfer coefficient and A_s is the heat transfer surface area of the heat exchanger. For specified values of U and $C_{\min}$, the value of NTU is a measure of the heat exchanger surface area A_s . Because the effectiveness increases slowly for larger values of NTU, a large heat exchanger cannot be justified economically. Therefore, a heat exchanger with a very large NTU is not necessarily a good one to buy.

16-86C The value of effectiveness increases slowly with a large values of NTU (usually larger than 3). Therefore, doubling the size of the heat exchanger will not save much energy in this case since the increase in the effectiveness will be very small.

16-87C The value of effectiveness increases rapidly with small values of NTU (up to about 1.5). Therefore, tripling the NTU will cause a rapid increase in the effectiveness of the heat exchanger, and thus saves energy. I would support this proposal.

16-88 Hot water coming from the engine of an automobile is cooled by air in the radiator. The outlet temperature of the air and the rate of heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and air are given to be 4.00 and 1.00 kJ/kg·°C, respectively.

Analysis (a) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (5 \text{ kg/s})(4.00 \text{ kJ/kg} \cdot ^\circ\text{C}) = 20 \text{ kW}/^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (10 \text{ kg/s})(1.00 \text{ kJ/kg} \cdot ^\circ\text{C}) = 10 \text{ kW}/^\circ\text{C}$$

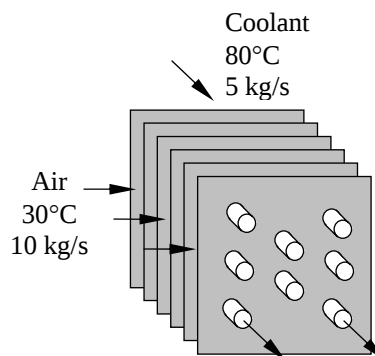
Therefore $C_{\min} = C_c = 10 \text{ kW}/^\circ\text{C}$

which is the smaller of the two heat capacity rates. Noting that the heat capacity rate of the air is the smaller one, the outlet temperature of the air is determined from the effectiveness relation to be

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_{\min}(T_{a,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} \longrightarrow 0.4 = \frac{(T_{a,out} - 30)^\circ\text{C}}{(80 - 30)^\circ\text{C}} \longrightarrow T_{a,out} = 50^\circ\text{C}$$

(b) The rate of heat transfer is determined from

$$\dot{Q} = C_{air}(T_{a,out} - T_{a,in}) = (10 \text{ kW}/^\circ\text{C})(50^\circ\text{C} - 30^\circ\text{C}) = \mathbf{200 \text{ kW}}$$



16-89 The inlet and exit temperatures and the volume flow rates of hot and cold fluids in a heat exchanger are given. The rate of heat transfer to the cold water, the overall heat transfer coefficient, the fraction of heat loss, the heat transfer efficiency, the effectiveness, and the NTU of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Changes in the kinetic and potential energies of fluid streams are negligible. 3 Fluid properties are constant.

Properties The densities of hot water and cold water at the average temperatures of $(71.5+58.2)/2 = 64.9^\circ\text{C}$ and $(19.7+27.8)/2 = 23.8^\circ\text{C}$ are 980.5 and 997.3 kg/m^3 , respectively. The specific heat at the average temperature is $4187 \text{ J/kg}\cdot^\circ\text{C}$ for hot water and $4180 \text{ J/kg}\cdot^\circ\text{C}$ for cold water (Table A-15).

Analysis (a) The mass flow rates are

$$\dot{m}_h = \rho_h \dot{V}_h = (980.5 \text{ kg/m}^3)(0.00105/60 \text{ m}^3/\text{s}) = 0.0172 \text{ kg/s}$$

$$\dot{m}_c = \rho_c \dot{V}_c = (997.3 \text{ kg/m}^3)(0.00155/60 \text{ m}^3/\text{s}) = 0.0258 \text{ kg/s}$$

The rates of heat transfer from the hot water and to the cold water are

$$\dot{Q}_h = [\dot{m}_c p (T_{in} - T_{out})]_h = (0.0172 \text{ kg/s})(4187 \text{ kJ/kg}\cdot^\circ\text{C})(71.5^\circ\text{C} - 58.2^\circ\text{C}) = 957.8 \text{ W}$$

$$\dot{Q}_c = [\dot{m}_c p (T_{out} - T_{in})]_c = (0.0258 \text{ kg/s})(4180 \text{ kJ/kg}\cdot^\circ\text{C})(27.8^\circ\text{C} - 19.2^\circ\text{C}) = \mathbf{873.5 \text{ W}}$$

(b) The number of shell and tubes are not specified in the problem. Therefore, we take the correction factor to be unity in the following calculations. The logarithmic mean temperature difference and the overall heat transfer coefficient are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 71.5^\circ\text{C} - 27.8^\circ\text{C} = 43.7^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 58.2^\circ\text{C} - 19.7^\circ\text{C} = 38.5^\circ\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{43.7 - 38.5}{\ln\left(\frac{43.7}{38.5}\right)} = 41.0^\circ\text{C}$$

$$U = \frac{\dot{Q}_{hc,m}}{A \Delta T_{lm}} = \frac{(957.8 + 873.5) / 2 \text{ W}}{(0.02 \text{ m}^2)(41.0^\circ\text{C})} = \mathbf{1117 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

Note that we used the average of two heat transfer rates in calculations.

(c) The fraction of heat loss and the heat transfer efficiency are

$$f_{loss} = \frac{\dot{Q}_h - \dot{Q}_c}{\dot{Q}_h} = \frac{957.8 - 873.5}{957.8} = 0.088 = \mathbf{8.8\%}$$

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_h} = \frac{873.5}{957.8} = 0.912 = \mathbf{91.2\%}$$

(d) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.0172 \text{ kg/s})(4187 \text{ kJ/kg}\cdot^\circ\text{C}) = 72.02 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (0.0258 \text{ kg/s})(4180 \text{ kJ/kg}\cdot^\circ\text{C}) = 107.8 \text{ W/}^\circ\text{C}$$

Therefore

$$C_{min} = C_h = 72.02 \text{ W/}^\circ\text{C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

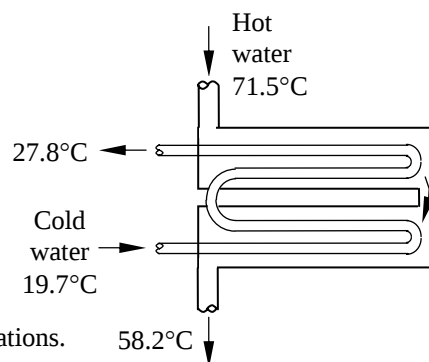
$$\dot{Q}_{max} = C_{min} (T_{h,in} - T_{c,in}) = (72.02 \text{ W/}^\circ\text{C})(71.5^\circ\text{C} - 19.7^\circ\text{C}) = 3731 \text{ W}$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{(957.8 + 873.5) / 2 \text{ kW}}{3731 \text{ kW}} = 0.245 = \mathbf{24.5\%}$$

One again we used the average heat transfer rate. We could have used the smaller or greater heat transfer rates in calculations. The NTU of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{min}} = \frac{(1117 \text{ W/m}^2 \cdot ^\circ\text{C})(0.02 \text{ m}^2)}{72.02 \text{ W/}^\circ\text{C}} = \mathbf{0.310}$$



16-90 Water is heated by a hot water stream in a heat exchanger. The maximum outlet temperature of the cold water and the effectiveness of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and air are given to be 4.18 and 1.0 kJ/kg·°C.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.8 \text{ kg/s})(1.0 \text{ kJ/kg} \cdot ^\circ\text{C}) = 0.8 \text{ kW/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (0.35 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 1.463 \text{ kW/}^\circ\text{C}$$

Therefore

$$C_{\min} = C_h = 0.8 \text{ kW/}^\circ\text{C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (0.8 \text{ kW/}^\circ\text{C})(65^\circ\text{C} - 14^\circ\text{C}) = 40.80 \text{ kW}$$

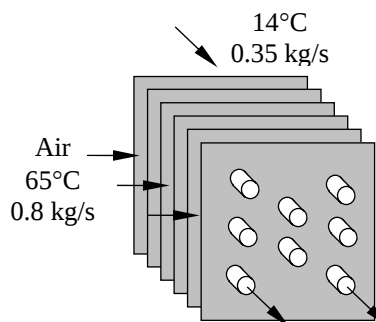
The maximum outlet temperature of the cold fluid is determined to be

$$\dot{Q}_{\max} = C_c (T_{c,out,max} - T_{c,in}) \longrightarrow T_{c,out,max} = T_{c,in} + \frac{\dot{Q}_{\max}}{C_c} = 14^\circ\text{C} + \frac{40.80 \text{ kW}}{1.463 \text{ kW/}^\circ\text{C}} = \mathbf{41.9^\circ\text{C}}$$

The actual rate of heat transfer and the effectiveness of the heat exchanger are

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) = (0.8 \text{ kW/}^\circ\text{C})(65^\circ\text{C} - 25^\circ\text{C}) = 32 \text{ kW}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{32 \text{ kW}}{40.8 \text{ kW}} = \mathbf{0.784}$$



16-91 Lake water is used to condense steam in a shell and tube heat exchanger. The outlet temperature of the water and the required tube length are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The properties of water are given in problem statement. The enthalpy of vaporization of water at 60°C is 2359 kJ/kg (Table A-15).

Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m} h_{fg} = (2.5 \text{ kg/s})(2359 \text{ kJ/kg}) = 5898 \text{ kW}$$

The outlet temperature of water is determined from

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_c} = 20^\circ\text{C} + \frac{5898 \text{ kW}}{(200 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})} = \mathbf{27.1^\circ\text{C}}$$

(b) The Reynold number is

$$\text{Re} = \frac{4\dot{m}}{N_{\text{tube}} \pi D \mu} = \frac{4(200 \text{ kg/s})}{(300) \pi (0.025 \text{ m})(8 \times 10^{-4} \text{ kg/m} \cdot \text{s})} = 42,440$$

which is greater than 10,000. Therefore, we have turbulent flow. We assume fully developed flow and evaluate the Nusselt number from

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(42,440)^{0.8} (6)^{0.4} = 237.2$$

Heat transfer coefficient on the inner surface of the tubes is

$$h_i = \frac{k}{D} \text{Nu} = \frac{0.6 \text{ W/m} \cdot ^\circ\text{C}}{0.025 \text{ m}} (237.2) = 5694 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Disregarding the thermal resistance of the tube wall the overall heat transfer coefficient is determined from

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5694} + \frac{1}{8500}} = 3410 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The logarithmic mean temperature difference is

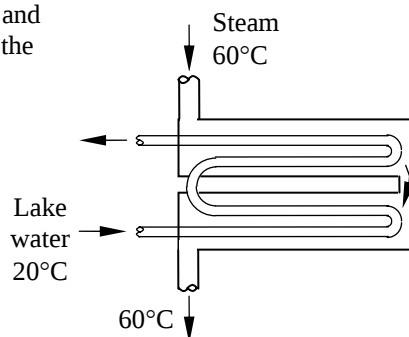
$$\Delta T_1 = T_{h,in} - T_{c,out} = 60^\circ\text{C} - 27.1^\circ\text{C} = 32.9^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60^\circ\text{C} - 20^\circ\text{C} = 40^\circ\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{32.9 - 40}{\ln\left(\frac{32.9}{40}\right)} = 36.3^\circ\text{C}$$

Noting that each tube makes two passes and taking the correction factor to be unity, the tube length per pass is determined to be

$$\dot{Q} = UAF\Delta T_{lm} \rightarrow L = \frac{\dot{Q}}{U(\pi D)F\Delta T_{lm}} = \frac{5898 \text{ kW}}{(3.410 \text{ kW/m}^2 \cdot ^\circ\text{C})[2 \times 300 \times \pi \times 0.025 \text{ m}^2](1)(36.3^\circ\text{C})} = \mathbf{1.01 \text{ m}}$$



16-92 Air is heated by a hot water stream in a cross-flow heat exchanger. The maximum heat transfer rate and the outlet temperatures of the cold and hot fluid streams are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of water and air are given to be 4.19 and 1.005 kJ/kg·°C.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (1 \text{ kg/s})(4190 \text{ J/kg} \cdot ^\circ\text{C}) = 4190 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (3 \text{ kg/s})(1005 \text{ J/kg} \cdot ^\circ\text{C}) = 3015 \text{ W/}^\circ\text{C}$$

Therefore

$$C_{\min} = C_c = 3015 \text{ W/}^\circ\text{C}$$

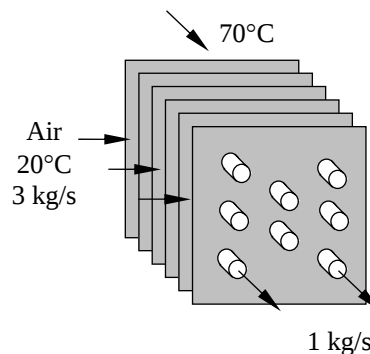
which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (3015 \text{ W/}^\circ\text{C})(70^\circ\text{C} - 20^\circ\text{C}) = 150,750 \text{ W} = \mathbf{150.8 \text{ kW}}$$

The outlet temperatures of the cold and the hot streams in this limiting case are determined to be

$$\dot{Q} = C_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20^\circ\text{C} + \frac{150.75 \text{ kW}}{3.015 \text{ kW/}^\circ\text{C}} = \mathbf{70^\circ\text{C}}$$

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 70^\circ\text{C} - \frac{150.75 \text{ kW}}{4.19 \text{ kW/}^\circ\text{C}} = \mathbf{34.0^\circ\text{C}}$$



16-93 Hot oil is to be cooled by water in a heat exchanger. The mass flow rates and the inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined. ✓

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The thickness of the tube is negligible since it is thin-walled. 5 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg·°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.2 \text{ kg/s})(2200 \text{ J/kg} \cdot ^\circ\text{C}) = 440 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (0.1 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C}) = 418 \text{ W/}^\circ\text{C}$$

Therefore, $C_{\min} = C_c = 418 \text{ W/}^\circ\text{C}$

$$\text{and } c = \frac{C_{\min}}{C_{\max}} = \frac{418}{440} = 0.95$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (418 \text{ W/}^\circ\text{C})(160^\circ\text{C} - 18^\circ\text{C}) = 59.36 \text{ kW}$$

The heat transfer surface area is

$$A_s = n(\pi DL) = (12)(\pi)(0.018 \text{ m})(3 \text{ m}) = 2.04 \text{ m}^2$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(340 \text{ W/m}^2 \cdot ^\circ\text{C})(2.04 \text{ m}^2)}{418 \text{ W/}^\circ\text{C}} = 1.659$$

Then the effectiveness of this heat exchanger corresponding to $c = 0.95$ and $NTU = 1.659$ is determined from Fig. 16-26d to be

$$\varepsilon = 0.61$$

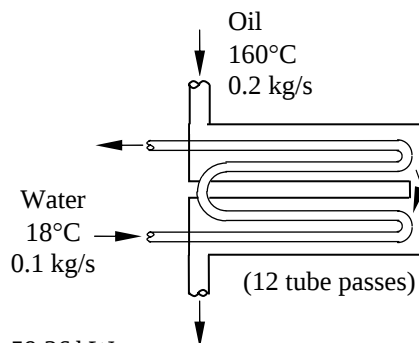
Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.61)(59.36 \text{ kW}) = \mathbf{36.2 \text{ kW}}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 18^\circ\text{C} + \frac{36.2 \text{ kW}}{0.418 \text{ kW/}^\circ\text{C}} = \mathbf{104.6^\circ\text{C}}$$

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 160^\circ\text{C} - \frac{36.2 \text{ kW}}{0.44 \text{ kW/}^\circ\text{C}} = \mathbf{77.7^\circ\text{C}}$$



16-94 Inlet and outlet temperatures of the hot and cold fluids in a double-pipe heat exchanger are given. It is to be determined whether this is a parallel-flow or counter-flow heat exchanger and the effectiveness of it.

Analysis This is a counter-flow heat exchanger because in the parallel-flow heat exchangers the outlet temperature of the cold fluid (55°C in this case) cannot exceed the outlet temperature of the hot fluid, which is (45°C in this case). Noting that the mass flow rates of both hot and cold oil streams are the same, we have $C_{\min} = C_{\max}$. Then the effectiveness of this heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h (T_{h,in} - T_{h,out})}{C_{\min} (T_{h,in} - T_{c,in})} = \frac{C_h (T_{h,in} - T_{h,out})}{C_h (T_{h,in} - T_{c,in})} = \frac{80^\circ\text{C} - 45^\circ\text{C}}{80^\circ\text{C} - 20^\circ\text{C}} = \mathbf{0.583}$$

16-95E Inlet and outlet temperatures of the hot and cold fluids in a double-pipe heat exchanger are given. It is to be determined the fluid, which has the smaller heat capacity rate and the effectiveness of the heat exchanger.

Analysis Hot water has the smaller heat capacity rate since it experiences a greater temperature change. The effectiveness of this heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_h(T_{h,in} - T_{c,in})} = \frac{190^\circ\text{F} - 100^\circ\text{F}}{190^\circ\text{F} - 70^\circ\text{F}} = \mathbf{0.75}$$

16-96 A chemical is heated by water in a heat exchanger. The mass flow rates and the inlet temperatures are given. The outlet temperatures of both fluids are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The thickness of the tube is negligible since tube is thin-walled. 5 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and chemical are given to be 4.18 and 1.8 kJ/kg·°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (2 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C}) = 8.36 \text{ kW}/^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (3 \text{ kg/s})(1.8 \text{ kJ/kg}\cdot^\circ\text{C}) = 5.40 \text{ kW}/^\circ\text{C}$$

Therefore, $C_{\min} = C_c = 5.4 \text{ kW}/^\circ\text{C}$

and
$$c = \frac{C_{\min}}{C_{\max}} = \frac{5.40}{8.36} = 0.646$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (5.4 \text{ kW}/^\circ\text{C})(110^\circ\text{C} - 20^\circ\text{C}) = 486 \text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(1.2 \text{ kW/m}^2\cdot^\circ\text{C})(7 \text{ m}^2)}{5.4 \text{ kW}/^\circ\text{C}} = 1.556$$

Then the effectiveness of this parallel-flow heat exchanger corresponding to $c = 0.646$ and $NTU = 1.556$ is determined from

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-1.556(1+0.646)]}{1+0.646} = 0.56$$

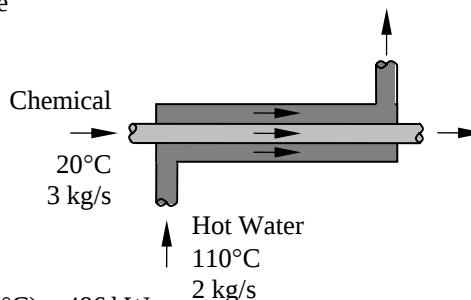
Then the actual rate of heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.56)(486 \text{ kW}) = 272.2 \text{ kW}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20^\circ\text{C} + \frac{272.2 \text{ kW}}{5.4 \text{ kW}/^\circ\text{C}} = \mathbf{70.4^\circ\text{C}}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 110^\circ\text{C} - \frac{272.2 \text{ kW}}{8.36 \text{ kW}/^\circ\text{C}} = \mathbf{77.4^\circ\text{C}}$$



16-97 EES Prob. 16-96 is reconsidered. The effects of the inlet temperatures of the chemical and the water on their outlet temperatures are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

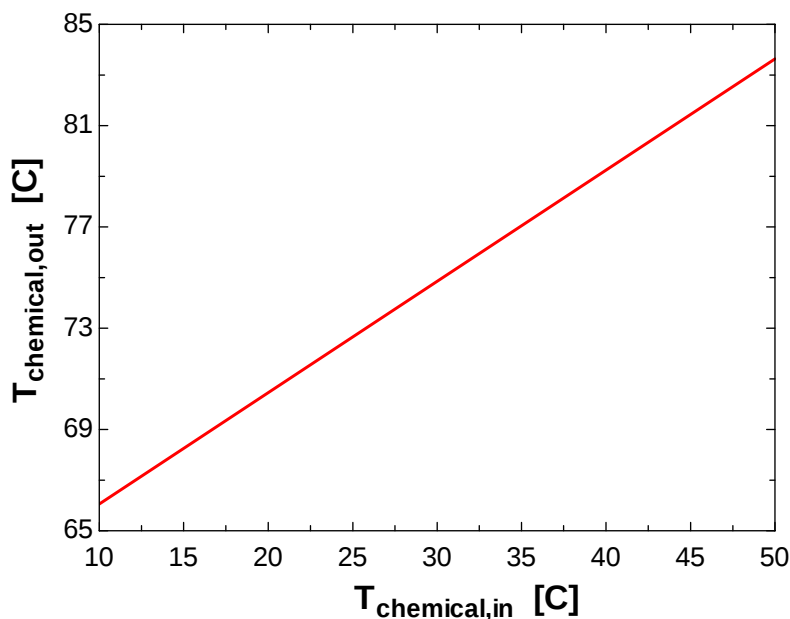
$T_{\text{chemical,in}} = 20 \text{ [C]}$
 $c_{p,\text{chemical}} = 1.8 \text{ [kJ/kg-C]}$
 $\dot{m}_{\text{chemical}} = 3 \text{ [kg/s]}$
 $T_{w,\text{in}} = 110 \text{ [C]}$
 $\dot{m}_{\text{dot}_w} = 2 \text{ [kg/s]}$
 $c_{p,w} = 4.18 \text{ [kJ/kg-C]}$
 $A = 7 \text{ [m}^2\text{]}$
 $U = 1.2 \text{ [kW/m}^2\text{-C]}$

"ANALYSIS"

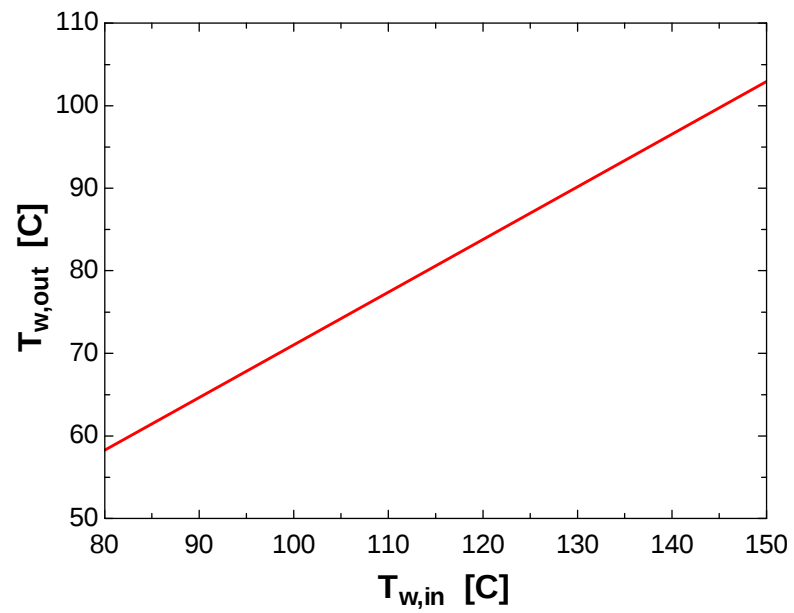
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use LMTD method. Both methods give the same results."

$\text{DELTAT}_1 = T_{w,\text{in}} - T_{\text{chemical,in}}$
 $\text{DELTAT}_2 = T_{w,\text{out}} - T_{\text{chemical,out}}$
 $\text{DELTAT}_{\text{lm}} = (\text{DELTAT}_1 - \text{DELTAT}_2) / \ln(\text{DELTAT}_1 / \text{DELTAT}_2)$
 $\dot{Q}_{\text{dot}} = U * A * \text{DELTAT}_{\text{lm}}$
 $\dot{Q}_{\text{dot}} = \dot{m}_{\text{chemical}} * c_{p,\text{chemical}} * (T_{\text{chemical,out}} - T_{\text{chemical,in}})$
 $\dot{Q}_{\text{dot}} = \dot{m}_{\text{dot}_w} * c_{p,w} * (T_{w,\text{in}} - T_{w,\text{out}})$

$T_{\text{chemical,in}}$ [C]	$T_{\text{chemical,out}}$ [C]
10	66.06
12	66.94
14	67.82
16	68.7
18	69.58
20	70.45
22	71.33
24	72.21
26	73.09
28	73.97
30	74.85
32	75.73
34	76.61
36	77.48
38	78.36
40	79.24
42	80.12
44	81
46	81.88
48	82.76
50	83.64



$T_{w,in} [^{\circ}\text{C}]$	$T_{w,out} [^{\circ}\text{C}]$
80	58.27
85	61.46
90	64.65
95	67.84
100	71.03
105	74.22
110	77.41
115	80.6
120	83.79
125	86.98
130	90.17
135	93.36
140	96.55
145	99.74
150	102.9



16-98 Water is heated by hot air in a heat exchanger. The mass flow rates and the inlet temperatures are given. The heat transfer surface area of the heat exchanger on the water side is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg·°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (4 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 16.72 \text{ kW}/^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (9 \text{ kg/s})(1.01 \text{ kJ/kg} \cdot ^\circ\text{C}) = 9.09 \text{ kW}/^\circ\text{C}$$

Therefore, $C_{\min} = C_c = 9.09 \text{ kW}/^\circ\text{C}$

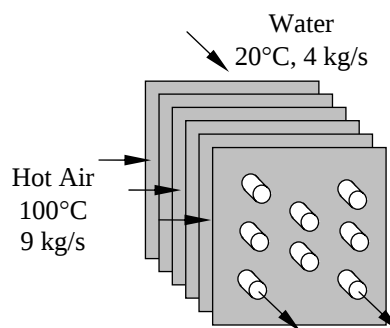
$$\text{and} \quad C = \frac{C_{\min}}{C_{\max}} = \frac{9.09}{16.72} = 0.544$$

Then the NTU of this heat exchanger corresponding to $C = 0.544$ and $\varepsilon = 0.65$ is determined from Fig. 16-26 to be

$$\text{NTU} = 1.5$$

Then the surface area of this heat exchanger becomes

$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{\text{NTU } C_{\min}}{U} = \frac{(1.5)(9.09 \text{ kW}/^\circ\text{C})}{0.260 \text{ kW/m}^2 \cdot ^\circ\text{C}} = \mathbf{52.4 \text{ m}^2}$$



16-99 Water is heated by steam condensing in a condenser. The required length of the tube is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of the water is given to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. The heat of vaporization of water at 120°C is given to be 2203 kJ/kg .

Analysis (a) The temperature differences between the steam and the water at the two ends of the condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 120^\circ\text{C} - 80^\circ\text{C} = 40^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 120^\circ\text{C} - 17^\circ\text{C} = 103^\circ\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 103}{\ln(40/103)} = 66.6^\circ\text{C}$$

The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) = (1.8 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(80^\circ\text{C} - 17^\circ\text{C}) = 474.0 \text{ kW}$$

The surface area of heat transfer is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{474.0 \text{ kW}}{0.7 \text{ kW/m}^2 \cdot ^\circ\text{C}(66.6^\circ\text{C})} = 10.17 \text{ m}^2$$

The length of tube required then becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{10.17 \text{ m}^2}{\pi(0.025 \text{ m})} = \mathbf{129.5 \text{ m}}$$

(b) The maximum rate of heat transfer rate is

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (1.8 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(120^\circ\text{C} - 17^\circ\text{C}) = 775.0 \text{ kW}$$

Then the effectiveness of this heat exchanger becomes

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{474 \text{ kW}}{775 \text{ kW}} = 0.612$$

The NTU of this heat exchanger is determined using the relation in Table 16-5 to be

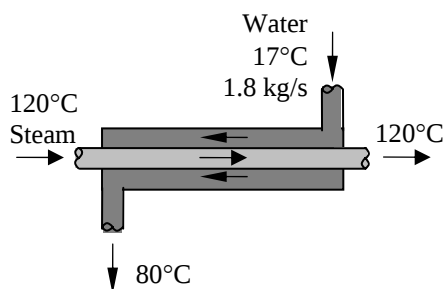
$$\text{NTU} = -\ln(1 - \varepsilon) = -\ln(1 - 0.612) = 0.947$$

The surface area is

$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{\text{NTU } C_{\min}}{U} = \frac{(0.947)(1.8 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})}{0.7 \text{ kW/m}^2 \cdot ^\circ\text{C}} = 10.18 \text{ m}^2$$

Finally, the length of tube required is

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{10.18 \text{ m}^2}{\pi(0.025 \text{ m})} = \mathbf{129.6 \text{ m}}$$



16-100 Ethanol is vaporized by hot oil in a double-pipe parallel-flow heat exchanger. The outlet temperature and the mass flow rate of oil are to be determined using the LMTD and NTU methods.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of oil is given to be 2.2 kJ/kg.°C. The heat of vaporization of ethanol at 78°C is given to be 846 kJ/kg.

Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}h_{fg} = (0.03 \text{ kg/s})(846 \text{ kJ/kg}) = 25.38 \text{ kW}$$

The log mean temperature difference is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow \Delta T_{lm} = \frac{\dot{Q}}{UA_s} = \frac{25,380 \text{ W}}{(320 \text{ W/m}^2 \cdot ^\circ\text{C})(6.2 \text{ m}^2)} = 12.8^\circ\text{C}$$

The outlet temperature of the hot fluid can be determined as follows

$$\Delta T_1 = T_{h,in} - T_{c,in} = 120^\circ\text{C} - 78^\circ\text{C} = 42^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = T_{h,out} - 78^\circ\text{C}$$

$$\text{and } \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{42 - (T_{h,out} - 78)}{\ln[42 / (T_{h,out} - 78)]} = 12.8^\circ\text{C}$$

whose solution is $T_{h,out} = \mathbf{79.8^\circ\text{C}}$

Then the mass flow rate of the hot oil becomes

$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{h,in} - T_{h,out})} = \frac{25,380 \text{ W}}{(2200 \text{ J/kg} \cdot ^\circ\text{C})(120^\circ\text{C} - 79.8^\circ\text{C})} = \mathbf{0.287 \text{ kg/s}}$$

(b) The heat capacity rate $C = \dot{m}c_p$ of a fluid condensing or evaporating in a heat exchanger is infinity, and thus $C = C_{\min} / C_{\max} = 0$.

The effectiveness in this case is determined from $\varepsilon = 1 - e^{-NTU}$

$$\text{where } NTU = \frac{UA_s}{C_{\min}} = \frac{(320 \text{ W/m}^2 \cdot ^\circ\text{C})(6.2 \text{ m}^2)}{(\dot{m}, \text{ kg/s})(2200 \text{ J/kg} \cdot ^\circ\text{C})}$$

$$\text{and } \dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in})$$

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\min}(T_{h,in} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{120 - T_{h,out}}{120 - 78}$$

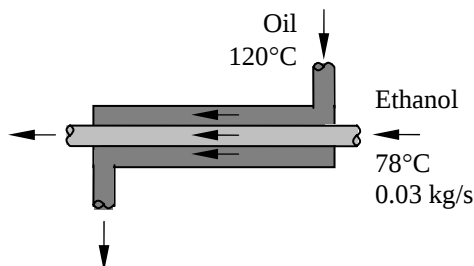
$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = 25,380 \text{ W} \quad (1)$$

$$\dot{Q} = \dot{m} \times 2200(120 - T_{h,out}) = 25,380 \text{ W}$$

$$\text{Also } \frac{120 - T_{h,out}}{120 - 78} = 1 - e^{-\frac{6.2 \times 320}{\dot{m} \times 2200}} \quad (2)$$

Solving (1) and (2) simultaneously gives

$$\dot{m}_h = \mathbf{0.287 \text{ kg/s}} \text{ and } T_{h,out} = \mathbf{79.8^\circ\text{C}}$$



16-101 Water is heated by solar-heated hot air in a heat exchanger. The mass flow rates and the inlet temperatures are given. The outlet temperatures of the water and the air are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg·°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.3 \text{ kg/s})(1010 \text{ J/kg} \cdot ^\circ\text{C}) = 303 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (0.1 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C}) = 418 \text{ W/}^\circ\text{C}$$

Therefore, $C_{\min} = C_c = 303 \text{ W/}^\circ\text{C}$

and
$$c = \frac{C_{\min}}{C_{\max}} = \frac{303}{418} = 0.725$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (303 \text{ W/}^\circ\text{C})(90^\circ\text{C} - 22^\circ\text{C}) = 20,604 \text{ W}$$

The heat transfer surface area is

$$A_s = \pi DL = (\pi)(0.012 \text{ m})(12 \text{ m}) = 0.45 \text{ m}^2$$

Then the NTU of this heat exchanger becomes

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(80 \text{ W/m}^2 \cdot ^\circ\text{C})(0.45 \text{ m}^2)}{303 \text{ W/}^\circ\text{C}} = 0.119$$

The effectiveness of this counter-flow heat exchanger corresponding to $c = 0.725$ and $NTU = 0.119$ is determined using the relation in Table 16-4 to be

$$\varepsilon = \frac{1 - \exp[-NTU(1 - c)]}{1 - c \exp[-NTU(1 - c)]} = \frac{1 - \exp[-0.119(1 - 0.725)]}{1 - 0.725 \exp[-0.119(1 - 0.725)]} = 0.108$$

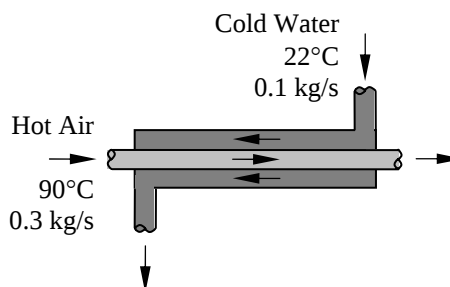
Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.108)(20,604 \text{ W}) = 2225.2 \text{ W}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 22^\circ\text{C} + \frac{2225.2 \text{ W}}{418 \text{ W/}^\circ\text{C}} = \mathbf{27.3^\circ\text{C}}$$

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 90^\circ\text{C} - \frac{2225.2 \text{ W}}{303 \text{ W/}^\circ\text{C}} = \mathbf{82.7^\circ\text{C}}$$



16-102 EES Prob. 16-101 is reconsidered. The effects of the mass flow rate of water and the tube length on the outlet temperatures of water and air are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

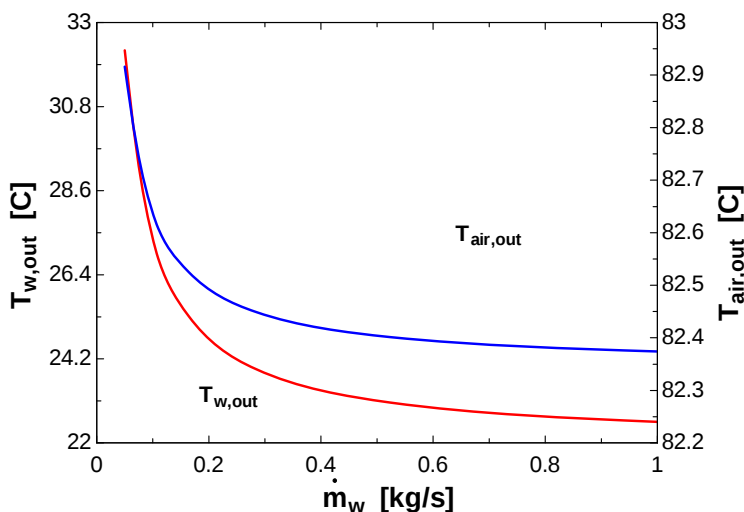
T_{air,in}=90 [C]
 m_{dot}_air=0.3 [kg/s]
 c_p_air=1.01 [kJ/kg-C]
 T_{w,in}=22 [C]
 m_{dot}_w=0.1 [kg/s]
 c_p_w=4.18 [kJ/kg-C]
 U=0.080 [kW/m²-C]
 L=12 [m]
 D=0.012 [m]

"ANALYSIS"

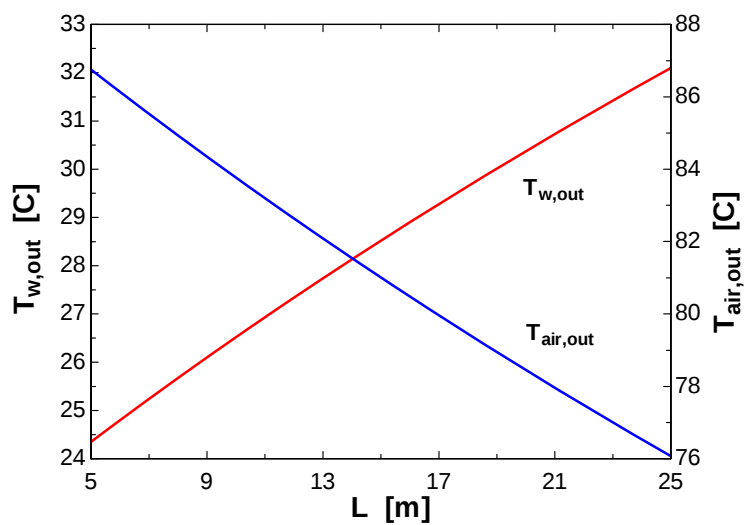
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use LMTD method. Both methods give the same results."

DELTA_T_1=T_{air,in}-T_{w,out}
 DELTA_T_2=T_{air,out}-T_{w,in}
 DELTA_T_lm=(DELTA_T_1-DELTA_T_2)/ln(DELTA_T_1/DELTA_T_2)
 A=pi*D*L
 Q_dot=U*A*DELTA_T_lm
 Q_dot=m_{dot}_air*c_p_air*(T_{air,in}-T_{air,out})
 Q_dot=m_{dot}_w*c_p_w*(T_{w,out}-T_{w,in})

m _w [kg/s]	T _{w,out} [C]	T _{air,out} [C]
0.05	32.27	82.92
0.1	27.34	82.64
0.15	25.6	82.54
0.2	24.72	82.49
0.25	24.19	82.46
0.3	23.83	82.44
0.35	23.57	82.43
0.4	23.37	82.42
0.45	23.22	82.41
0.5	23.1	82.4
0.55	23	82.4
0.6	22.92	82.39
0.65	22.85	82.39
0.7	22.79	82.39
0.75	22.74	82.38
0.8	22.69	82.38
0.85	22.65	82.38
0.9	22.61	82.38
0.95	22.58	82.38
1	22.55	82.37



L [m]	T _{w, out} [C]	T _{air, out} [C]
5	24.35	86.76
6	24.8	86.14
7	25.24	85.53
8	25.67	84.93
9	26.1	84.35
10	26.52	83.77
11	26.93	83.2
12	27.34	82.64
13	27.74	82.09
14	28.13	81.54
15	28.52	81.01
16	28.9	80.48
17	29.28	79.96
18	29.65	79.45
19	30.01	78.95
20	30.37	78.45
21	30.73	77.96
22	31.08	77.48
23	31.42	77
24	31.76	76.53
25	32.1	76.07



16-103E Oil is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient of this heat exchanger is to be determined using both the LMTD and NTU methods.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The thickness of the tube is negligible since it is thin-walled.

Properties The specific heats of the water and oil are given to be 1.0 and 0.525 Btu/lbm.°F, respectively.

Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h,in} - T_{h,out}) = (5 \text{ lbm/s})(0.525 \text{ Btu/lbm.°F})(300 - 105^\circ\text{F}) = 511.9 \text{ Btu/s}$$

The outlet temperature of the cold fluid is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 70^\circ\text{F} + \frac{511.9 \text{ Btu/s}}{(3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})} = 240.6^\circ\text{F}$$

The temperature differences between the two fluids at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 300^\circ\text{F} - 240.6^\circ\text{F} = 59.4^\circ\text{F}$$

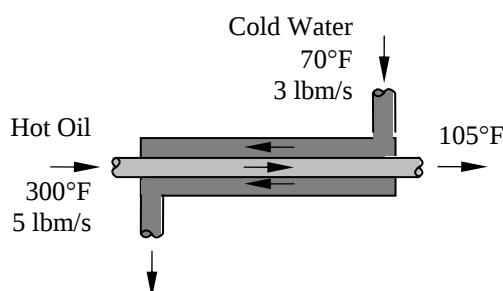
$$\Delta T_2 = T_{h,out} - T_{c,in} = 105^\circ\text{F} - 70^\circ\text{F} = 35^\circ\text{F}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{59.4 - 35}{\ln(59.4/35)} = 46.1^\circ\text{F}$$

Then the overall heat transfer coefficient becomes

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow U = \frac{\dot{Q}}{A_s \Delta T_{lm}} = \frac{511.9 \text{ Btu/s}}{\pi(5/12 \text{ m})(200 \text{ ft})(46.1^\circ\text{F})} = \mathbf{0.0424 \text{ Btu/s.ft}^2 \cdot ^\circ\text{F}}$$



(b) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (5 \text{ lbm/s})(0.525 \text{ Btu/lbm.°F}) = 2.625 \text{ Btu/s.°F}$$

$$C_c = \dot{m}_c c_{pc} = (3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F}) = 3.0 \text{ Btu/s.°F}$$

$$\text{Therefore, } C_{\min} = C_h = 2.625 \text{ Btu/s.°F} \quad \text{and} \quad c = \frac{C_{\min}}{C_{\max}} = \frac{2.625}{3.0} = 0.875$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (2.625 \text{ Btu/s.°F})(300^\circ\text{F} - 70^\circ\text{F}) = 603.75 \text{ Btu/s}$$

The actual rate of heat transfer and the effectiveness are

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) = (2.625 \text{ Btu/s.°F})(300^\circ\text{F} - 105^\circ\text{F}) = 511.9 \text{ Btu/s}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{511.9}{603.75} = 0.85$$

The NTU of this heat exchanger is determined using the relation in Table 16-5 to be

$$NTU = \frac{1}{c-1} \ln\left(\frac{\varepsilon-1}{\varepsilon c-1}\right) = \frac{1}{0.875-1} \ln\left(\frac{0.85-1}{0.85 \times 0.875-1}\right) = 4.28$$

The heat transfer surface area of the heat exchanger is

$$A_s = \pi DL = \pi(5/12 \text{ ft})(200 \text{ ft}) = 261.8 \text{ ft}^2$$

$$\text{and} \quad NTU = \frac{UA_s}{C_{\min}} \longrightarrow U = \frac{NTU C_{\min}}{A_s} = \frac{(4.28)(2.625 \text{ Btu/s.°F})}{261.8 \text{ ft}^2} = \mathbf{0.0429 \text{ Btu/s.ft}^2 \cdot ^\circ\text{F}}$$

16-104 Cold water is heated by hot water in a heat exchanger. The net rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform. 5 The thickness of the tube is negligible.

Properties The specific heats of the cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.25 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C}) = 1045 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (3 \text{ kg/s})(4190 \text{ J/kg} \cdot ^\circ\text{C}) = 12,570 \text{ W/}^\circ\text{C}$$

Therefore, $C_{\min} = C_c = 1045 \text{ W/}^\circ\text{C}$

and
$$c = \frac{C_{\min}}{C_{\max}} = \frac{1045}{12,570} = 0.083$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (1045 \text{ W/}^\circ\text{C})(100^\circ\text{C} - 15^\circ\text{C}) = 88,825 \text{ W}$$

The actual rate of heat transfer is

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) = (1045 \text{ W/}^\circ\text{C})(45^\circ\text{C} - 15^\circ\text{C}) = \mathbf{31,350 \text{ W}}$$

Then the effectiveness of this heat exchanger becomes

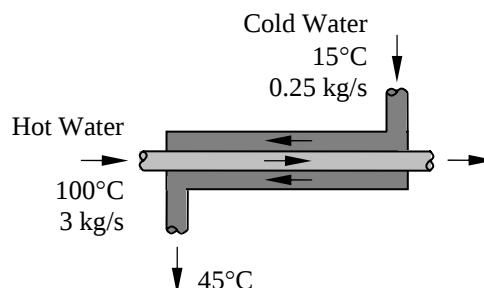
$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{31,350}{88,825} = 0.35$$

The NTU of this heat exchanger is determined using the relation in Table 16-5 to be

$$NTU = \frac{1}{c-1} \ln \left(\frac{\varepsilon-1}{\varepsilon c-1} \right) = \frac{1}{0.083-1} \ln \left(\frac{0.35-1}{0.35 \times 0.083-1} \right) = 0.438$$

Then the surface area of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{\min}} \longrightarrow A = \frac{NTU C_{\min}}{U} = \frac{(0.438)(1045 \text{ W/}^\circ\text{C})}{950 \text{ W/m}^2 \cdot ^\circ\text{C}} = \mathbf{0.482 \text{ m}^2}$$



16-105 EES Prob. 16-104 is reconsidered. The effects of the inlet temperature of hot water and the heat transfer coefficient on the rate of heat transfer and the surface area are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

T_{cw,in}=15 [C]
 T_{cw,out}=45 [C]
 m_{dot,cw}=0.25 [kg/s]
 c_{p,cw}=4.18 [kJ/kg-C]
 T_{hw,in}=100 [C]
 m_{dot,hw}=3 [kg/s]
 c_{p,hw}=4.19 [kJ/kg-C]
 U=0.95 [kW/m²-C]

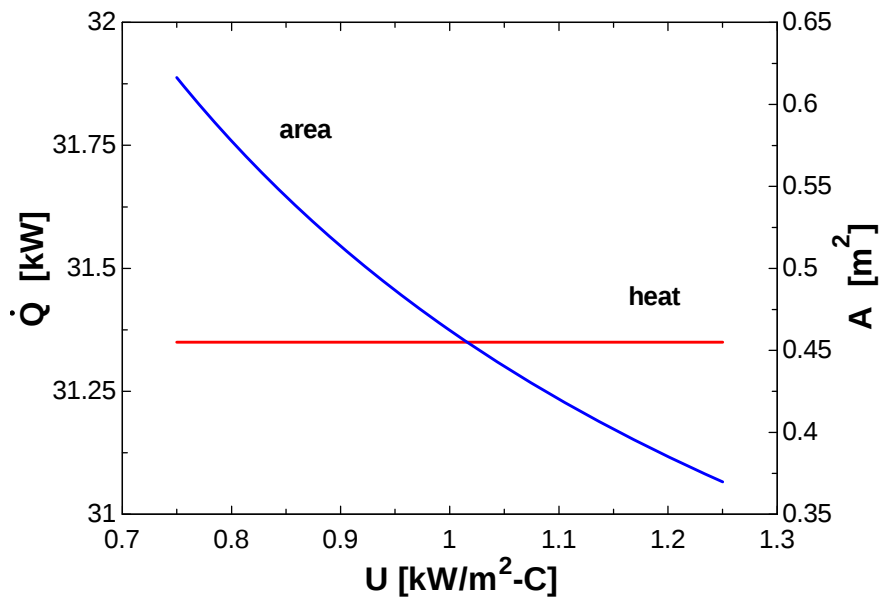
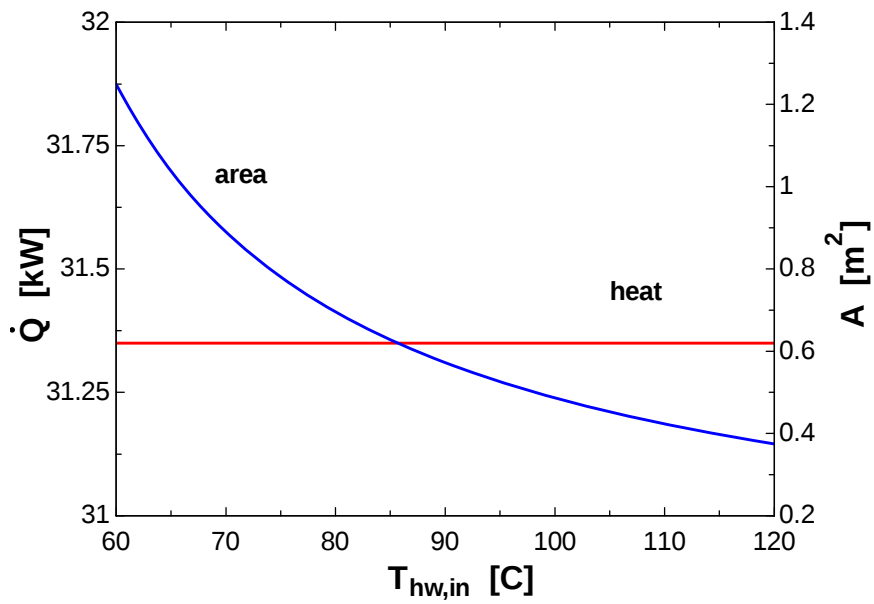
"ANALYSIS"

"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use LMTD method. Both methods give the same results."

DELTAT₁=T_{hw,in}-T_{cw,out}
 DELTAT₂=T_{hw,out}-T_{cw,in}
 DELTAT_{lm}=(DELTAT₁-DELTAT₂)/ln(DELTAT₁/DELTAT₂)
 Q_{dot}=U*A*DELTAT_{lm}
 Q_{dot}=m_{dot,hw}*c_{p,hw}*(T_{hw,in}-T_{hw,out})
 Q_{dot}=m_{dot,cw}*c_{p,cw}*(T_{cw,out}-T_{cw,in})

T _{hw,in} [C]	Q [kW]	A [m ²]
60	31.35	1.25
65	31.35	1.038
70	31.35	0.8903
75	31.35	0.7807
80	31.35	0.6957
85	31.35	0.6279
90	31.35	0.5723
95	31.35	0.5259
100	31.35	0.4865
105	31.35	0.4527
110	31.35	0.4234
115	31.35	0.3976
120	31.35	0.3748

U [kW/m ² -C]	Q [kW]	A [m ²]
0.75	31.35	0.6163
0.8	31.35	0.5778
0.85	31.35	0.5438
0.9	31.35	0.5136
0.95	31.35	0.4865
1	31.35	0.4622
1.05	31.35	0.4402
1.1	31.35	0.4202
1.15	31.35	0.4019
1.2	31.35	0.3852
1.25	31.35	0.3698



16-106 Glycerin is heated by ethylene glycol in a heat exchanger. Mass flow rates and inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform. 5 The thickness of the tube is negligible.

Properties The specific heats of the glycerin and ethylene glycol are given to be 2.4 and 2.5 kJ/kg.°C, respectively.

Analysis (a) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.3 \text{ kg/s})(2400 \text{ J/kg} \cdot ^\circ\text{C}) = 720 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (0.3 \text{ kg/s})(2500 \text{ J/kg} \cdot ^\circ\text{C}) = 750 \text{ W/}^\circ\text{C}$$

Therefore, $C_{\min} = C_h = 720 \text{ W/}^\circ\text{C}$

and
$$c = \frac{C_{\min}}{C_{\max}} = \frac{720}{750} = 0.96$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (720 \text{ W/}^\circ\text{C})(60^\circ\text{C} - 20^\circ\text{C}) = 28.8 \text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(380 \text{ W/m}^2 \cdot ^\circ\text{C})(5.3 \text{ m}^2)}{720 \text{ W/}^\circ\text{C}} = 2.797$$

Effectiveness of this heat exchanger corresponding to $c = 0.96$ and $NTU = 2.797$ is determined using the proper relation in Table 16-4

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-2.797(1+0.96)]}{1+0.96} = 0.508$$

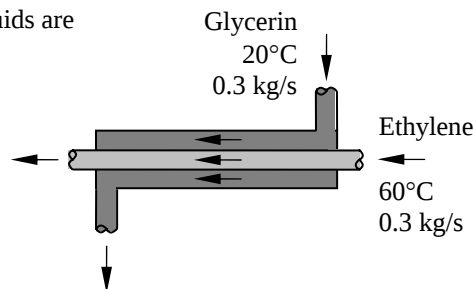
Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.508)(28.8 \text{ kW}) = \mathbf{14.63 \text{ kW}}$$

(b) Finally, the outlet temperatures of the cold and the hot fluid streams are determined from

$$\dot{Q} = C_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20^\circ\text{C} + \frac{14.63 \text{ kW}}{0.72 \text{ kW/}^\circ\text{C}} = \mathbf{40.3^\circ\text{C}}$$

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 60^\circ\text{C} - \frac{14.63 \text{ kW}}{0.75 \text{ kW/}^\circ\text{C}} = \mathbf{40.5^\circ\text{C}}$$



16-107 Water is heated by hot air in a cross-flow heat exchanger. Mass flow rates and inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform. 5 The thickness of the tube is negligible.

Properties The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg.°C, respectively.

Analysis The mass flow rates of the hot and the cold fluids are

$$\dot{m}_c = \rho V A_c = (1000 \text{ kg/m}^3)(3 \text{ m/s})[80\pi(0.03 \text{ m})^2/4] = 169.6 \text{ kg/s}$$

$$\rho_{air} = \frac{P}{RT} = \frac{105 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}) \times (130 + 273 \text{ K})} = 0.908 \text{ kg/m}^3$$

$$\dot{m}_h = \rho V A_c = (0.908 \text{ kg/m}^3)(12 \text{ m/s})(1 \text{ m})^2 = 10.90 \text{ kg/s}$$

The heat transfer surface area and the heat capacity rates are

$$A_s = n\pi DL = 80\pi(0.03 \text{ m})(1 \text{ m}) = 7.540 \text{ m}^2$$

$$C_c = \dot{m}_c c_{pc} = (169.6 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C}) = 708.9 \text{ kW}/^\circ\text{C}$$

$$C_h = \dot{m}_h c_{ph} = (10.9 \text{ kg/s})(1.010 \text{ kJ/kg}\cdot^\circ\text{C}) = 11.01 \text{ kW}/^\circ\text{C}$$

$$\text{Therefore, } C_{\min} = C_c = 11.01 \text{ kW}/^\circ\text{C} \quad \text{and} \quad c = \frac{C_{\min}}{C_{\max}} = \frac{11.01}{708.9} = 0.01553$$

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (11.01 \text{ kW}/^\circ\text{C})(130^\circ\text{C} - 18^\circ\text{C}) = 1233 \text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(130 \text{ W/m}^2\cdot^\circ\text{C})(7.540 \text{ m}^2)}{11,010 \text{ W}/^\circ\text{C}} = 0.08903$$

Noting that this heat exchanger involves mixed cross-flow, the fluid with $C_{\min}$ is mixed, $C_{\max}$ unmixed, effectiveness of this heat exchanger corresponding to $c = 0.01553$ and $NTU = 0.08903$ is determined using the proper relation in Table 16-4 to be

$$\varepsilon = 1 - \exp\left[-\frac{1}{c}(1 - e^{-cNTU})\right] = 1 - \exp\left[-\frac{1}{0.01553}(1 - e^{-0.01553 \times 0.08903})\right] = 0.08513$$

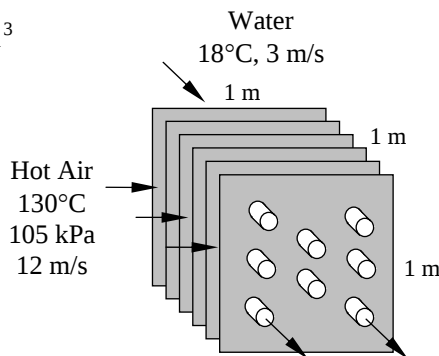
Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.08513)(1233 \text{ kW}) = \mathbf{105.0 \text{ kW}}$$

Finally, the outlet temperatures of the cold and the hot fluid streams are determined from

$$\dot{Q} = C_c (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 18^\circ\text{C} + \frac{105.0 \text{ kW}}{708.9 \text{ kW}/^\circ\text{C}} = \mathbf{18.15^\circ\text{C}}$$

$$\dot{Q} = C_h (T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 130^\circ\text{C} - \frac{105.0 \text{ kW}}{11.01 \text{ kW}/^\circ\text{C}} = \mathbf{120.5^\circ\text{C}}$$



16-108 CD EES Ethyl alcohol is heated by water in a shell-and-tube heat exchanger. The heat transfer surface area of the heat exchanger is to be determined using both the LMTD and NTU methods.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the ethyl alcohol and water are given to be 2.67 and 4.19 kJ/kg·°C, respectively.

Analysis (a) The temperature differences between the two fluids at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 95^\circ\text{C} - 70^\circ\text{C} = 25^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60^\circ\text{C} - 25^\circ\text{C} = 35^\circ\text{C}$$

The logarithmic mean temperature difference and the correction factor are

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25 - 35}{\ln(25/35)} = 29.7^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 25}{95 - 25} = 0.64 \\ R &= \frac{T_2 - T_1}{t_1 - t_1} = \frac{95 - 60}{70 - 25} = 0.78 \end{aligned} \right\} F = 0.93$$

The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) = (2.1 \text{ kg/s})(2.67 \text{ kJ/kg}\cdot^\circ\text{C})(70^\circ\text{C} - 25^\circ\text{C}) = 252.3 \text{ kW}$$

The surface area of heat transfer is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{UF \Delta T_{lm}} = \frac{252.3 \text{ kW}}{0.8 \text{ kW/m}^2 \cdot ^\circ\text{C} (0.93)(29.7^\circ\text{C})} = \mathbf{11.4 \text{ m}^2}$$

(b) The rate of heat transfer is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) = (2.1 \text{ kg/s})(2.67 \text{ kJ/kg}\cdot^\circ\text{C})(70^\circ\text{C} - 25^\circ\text{C}) = 252.3 \text{ kW}$$

The mass flow rate of the hot fluid is

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h,in} - T_{h,out}) \longrightarrow \dot{m}_h = \frac{\dot{Q}}{c_{ph} (T_{h,in} - T_{h,out})} = \frac{252.3 \text{ kW}}{(4.19 \text{ kJ/kg}\cdot^\circ\text{C})(95^\circ\text{C} - 60^\circ\text{C})} = 1.72 \text{ kg/s}$$

The heat capacity rates of the hot and the cold fluids are

$$C_h = \dot{m}_h c_{ph} = (1.72 \text{ kg/s})(4.19 \text{ kJ/kg}\cdot^\circ\text{C}) = 7.21 \text{ kW/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (2.1 \text{ kg/s})(2.67 \text{ kJ/kg}\cdot^\circ\text{C}) = 5.61 \text{ kW/}^\circ\text{C}$$

$$\text{Therefore, } C_{\min} = C_c = 5.61 \text{ W/}^\circ\text{C} \text{ and } c = \frac{C_{\min}}{C_{\max}} = \frac{5.61}{7.21} = 0.78$$

Then the maximum heat transfer rate becomes

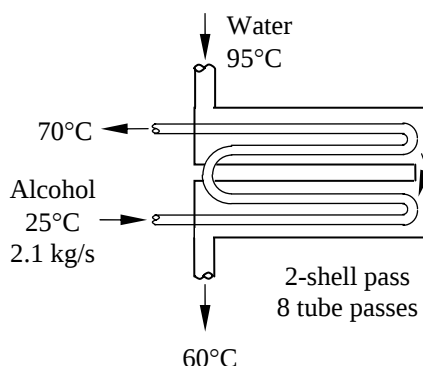
$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (5.61 \text{ W/}^\circ\text{C})(95^\circ\text{C} - 25^\circ\text{C}) = 392.7 \text{ kW}$$

$$\text{The effectiveness of this heat exchanger is } \varepsilon = \frac{Q}{Q_{\max}} = \frac{252.3}{392.7} = 0.64$$

The NTU of this heat exchanger corresponding to this emissivity and $c = 0.78$ is determined from Fig. 16-26d to be $NTU = 1.7$. Then the surface area of heat exchanger is determined to be

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{NTU C_{\min}}{U} = \frac{(1.7)(5.61 \text{ kW/}^\circ\text{C})}{0.8 \text{ kW/m}^2 \cdot ^\circ\text{C}} = \mathbf{11.9 \text{ m}^2}$$

The small difference between the two results is due to the reading error of the chart.



16-109 Steam is condensed by cooling water in a shell-and-tube heat exchanger. The rate of heat transfer and the rate of condensation of steam are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform. 5 The thickness of the tube is negligible.

Properties The specific heat of the water is given to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$. The heat of condensation of steam at 30°C is given to be 2430 kJ/kg .

Analysis (a) The heat capacity rate of a fluid condensing in a heat exchanger is infinity. Therefore,

$$C_{\min} = C_c = \dot{m}_c c_{pc} = (0.5 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 2.09 \text{ kW/}^\circ\text{C}$$

and $c = 0$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (2.09 \text{ kW/}^\circ\text{C})(30^\circ\text{C} - 15^\circ\text{C}) = 31.35 \text{ kW}$$

and

$$A_s = 8n\pi DL = 8 \times 50\pi(0.015 \text{ m})(2 \text{ m}) = 37.7 \text{ m}^2$$

The NTU of this heat exchanger

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(3 \text{ kW/m}^2 \cdot ^\circ\text{C})(37.7 \text{ m}^2)}{2.09 \text{ kW/}^\circ\text{C}} = 54.11$$

Then the effectiveness of this heat exchanger corresponding to $c = 0$ and $NTU = 54.11$ is determined using the proper relation in Table 16-5

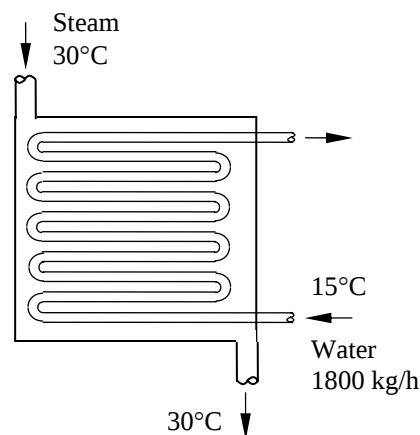
$$\varepsilon = 1 - \exp(-NTU) = 1 - \exp(-54.11) = 1$$

Then the actual heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (1)(31.35 \text{ kW}) = \mathbf{31.35 \text{ kW}}$$

(b) Finally, the rate of condensation of the steam is determined from

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{31.35 \text{ kJ/s}}{2430 \text{ kJ/kg}} = \mathbf{0.0129 \text{ kg/s}}$$



16-110 EES Prob. 16-109 is reconsidered. The effects of the condensing steam temperature and the tube diameter on the rate of heat transfer and the rate of condensation of steam are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

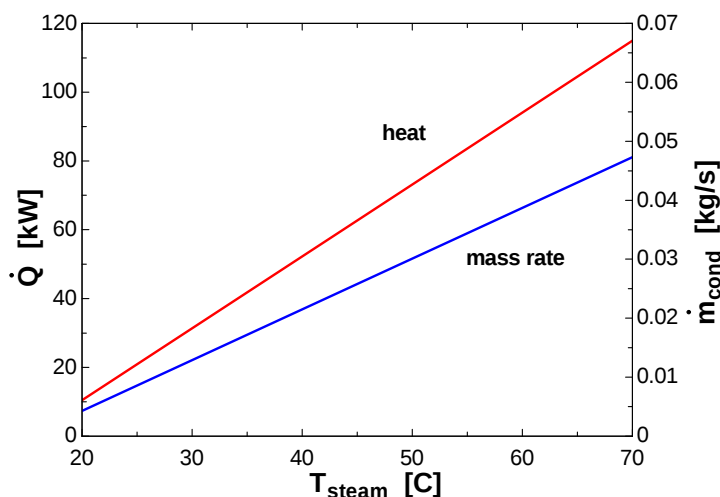
N_pass=8
 N_tube=50
 T_steam=30 [C]
 h_fg_steam=2430 [kJ/kg]
 T_w_in=15 [C]
 m_dot_w=1800[kg/h]*Convert(kg/h, kg/s)
 c_p_w=4.18 [kJ/kg-C]
 D=1.5 [cm]
 L=2 [m]
 U=3 [kW/m^2-C]

"ANALYSIS"

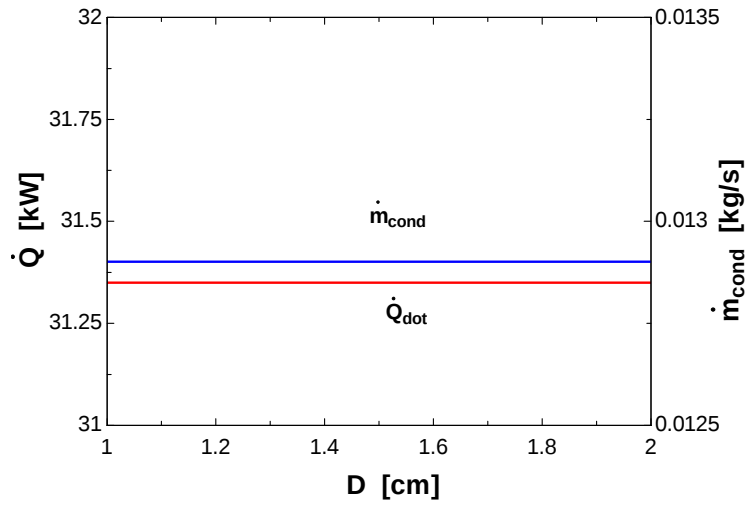
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use NTU method. Both methods give the same results."

C_min=m_dot_w*c_p_w
 c=0 "since the heat capacity rate of a fluid condensing is infinity"
 Q_dot_max=C_min*(T_steam-T_w_in)
 A=N_pass*N_tube*pi*D*L*Convert(cm, m)
 NTU=(U*A)/C_min
 epsilon=1-exp(-NTU) "from Table 16-4 of the text with c=0"
 Q_dot=epsilon*Q_dot_max
 Q_dot=m_dot_cond*h_fg_steam

T _{steam} [C]	Q [kW]	m _{cond} [kg/s]
20	10.45	0.0043
22.5	15.68	0.006451
25	20.9	0.008601
27.5	26.12	0.01075
30	31.35	0.0129
32.5	36.58	0.01505
35	41.8	0.0172
37.5	47.03	0.01935
40	52.25	0.0215
42.5	57.47	0.02365
45	62.7	0.0258
47.5	67.93	0.02795
50	73.15	0.0301
52.5	78.38	0.03225
55	83.6	0.0344
57.5	88.82	0.03655
60	94.05	0.0387
62.5	99.27	0.04085
65	104.5	0.043
67.5	109.7	0.04515
70	114.9	0.0473



D [cm]	Q [kW]	$\dot{m}_{\text{cond}}$ [kg/s]
1	31.35	0.0129
1.05	31.35	0.0129
1.1	31.35	0.0129
1.15	31.35	0.0129
1.2	31.35	0.0129
1.25	31.35	0.0129
1.3	31.35	0.0129
1.35	31.35	0.0129
1.4	31.35	0.0129
1.45	31.35	0.0129
1.5	31.35	0.0129
1.55	31.35	0.0129
1.6	31.35	0.0129
1.65	31.35	0.0129
1.7	31.35	0.0129
1.75	31.35	0.0129
1.8	31.35	0.0129
1.85	31.35	0.0129
1.9	31.35	0.0129
1.95	31.35	0.0129
2	31.35	0.0129



16-111 Cold water is heated by hot oil in a shell-and-tube heat exchanger. The rate of heat transfer is to be determined using both the LMTD and NTU methods.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg·°C, respectively.

Analysis (a) The LMTD method in this case involves iterations, which involves the following steps:

- 1) Choose $T_{h,out}$
- 2) Calculate $\dot{Q}$ from $\dot{Q} = \dot{m}_h c_p (T_{h,out} - T_{h,in})$
- 3) Calculate $T_{h,out}$ from $\dot{Q} = \dot{m}_h c_p (T_{h,out} - T_{h,in})$
- 4) Calculate $\Delta T_{lm,CF}$
- 5) Calculate $\dot{Q}$ from $\dot{Q} = UA_s F \Delta T_{lm,CF}$
- 6) Compare to the $\dot{Q}$ calculated at step 2, and repeat until reaching the same result

Result: **651 kW**

(b) The heat capacity rates of the hot and the cold fluids are

$$C_h = \dot{m}_h c_{ph} = (3 \text{ kg/s})(2.2 \text{ kJ/kg} \cdot ^\circ\text{C}) = 6.6 \text{ kW/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (3 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 12.54 \text{ kW/}^\circ\text{C}$$

Therefore, $C_{\min} = C_h = 6.6 \text{ kW/}^\circ\text{C}$ and $c = \frac{C_{\min}}{C_{\max}} = \frac{6.6}{12.54} = 0.53$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min} (T_{h,in} - T_{c,in}) = (6.6 \text{ kW/}^\circ\text{C})(200^\circ\text{C} - 14^\circ\text{C}) = 1228 \text{ kW}$$

The NTU of this heat exchanger is

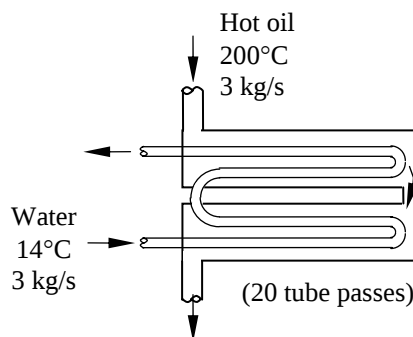
$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.3 \text{ kW/m}^2 \cdot ^\circ\text{C})(20 \text{ m}^2)}{6.6 \text{ kW/}^\circ\text{C}} = 0.91$$

Then the effectiveness of this heat exchanger corresponding to $c = 0.53$ and $NTU = 0.91$ is determined from Fig. 16-26d to be

$$\varepsilon = 0.53$$

The actual rate of heat transfer then becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.53)(1228 \text{ kW}) = \mathbf{651 \text{ kW}}$$



Selection of the Heat Exchangers

16-112C 1) Calculate heat transfer rate, 2) select a suitable type of heat exchanger, 3) select a suitable type of cooling fluid, and its temperature range, 4) calculate or select U , and 5) calculate the size (surface area) of heat exchanger

16-113C The first thing we need to do is determine the life expectancy of the system. Then we need to evaluate how much the larger will save in pumping cost, and compare it to the initial cost difference of the two units. If the larger system saves more than the cost difference in its lifetime, it should be preferred.

16-114C In the case of automotive and aerospace industry, where weight and size considerations are important, and in situations where the space availability is limited, we choose the smaller heat exchanger.

16-115 Oil is to be cooled by water in a heat exchanger. The heat transfer rating of the heat exchanger is to be determined and a suitable type is to be proposed.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the oil is given to be $2.2 \text{ kJ/kg}\cdot^\circ\text{C}$.

Analysis The heat transfer rate of this heat exchanger is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) = (13 \text{ kg/s})(2.2 \text{ kJ/kg}\cdot^\circ\text{C})(120^\circ\text{C} - 50^\circ\text{C}) = \mathbf{2002 \text{ kW}}$$

We propose a compact heat exchanger (like the car radiator) if air cooling is to be used, or a tube-and-shell or plate heat exchanger if water cooling is to be used.

3-116 Water is to be heated by steam in a shell-and-tube process heater. The number of tube passes need to be used is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the water is given to be $4.19 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The mass flow rate of the water is

$$\begin{aligned}\dot{Q} &= \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) \\ \dot{m} &= \frac{\dot{Q}}{c_{pc} (T_{c,out} - T_{c,in})} \\ &= \frac{600 \text{ kW}}{(4.19 \text{ kJ/kg} \cdot ^\circ\text{C})(90^\circ\text{C} - 20^\circ\text{C})} \\ &= 2.046 \text{ kg/s}\end{aligned}$$

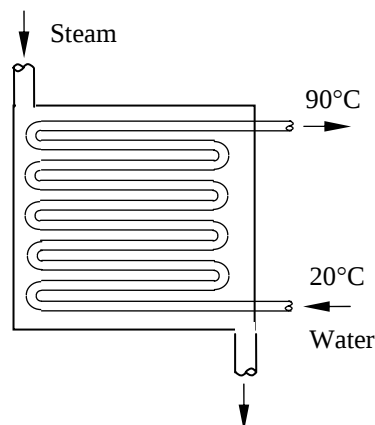
The total cross-section area of the tubes corresponding to this mass flow rate is

$$\dot{m} = \rho V A_c \rightarrow A_c = \frac{\dot{m}}{\rho V} = \frac{2.046 \text{ kg/s}}{(1000 \text{ kg/m}^3)(3 \text{ m/s})} = 6.82 \times 10^{-4} \text{ m}^2$$

Then the number of tubes that need to be used becomes

$$A_s = n \frac{\pi D^2}{4} \rightarrow n = \frac{4A_s}{\pi D^2} = \frac{4(6.82 \times 10^{-4} \text{ m}^2)}{\pi(0.01 \text{ m})^2} = 8.68 \cong 9$$

Therefore, we need to use at least 9 tubes entering the heat exchanger.



16-117 EES Prob. 16-116 is reconsidered. The number of tube passes as a function of water velocity is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

"GIVEN"

$c_{p,w}=4.19$ [kJ/kg-C]
 $T_{w,in}=20$ [C]
 $T_{w,out}=90$ [C]
 $\dot{Q}=600$ [kW]
 $D=0.01$ [m]
 $Vel=3$ [m/s]

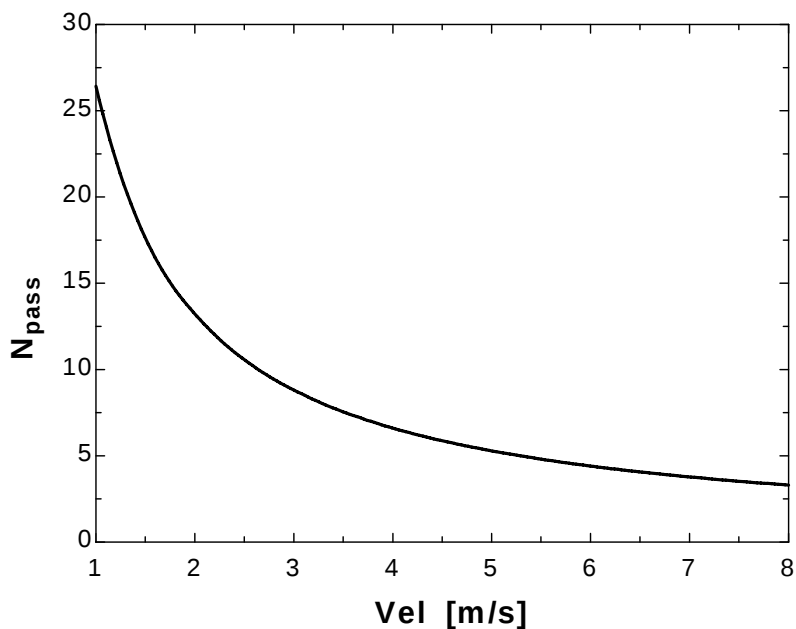
"PROPERTIES"

$\rho=\text{density}(\text{water}, T=T_{ave}, P=100)$
 $T_{ave}=1/2*(T_{w,in}+T_{w,out})$

"ANALYSIS"

$\dot{Q}=\dot{m}_w*c_{p,w}*(T_{w,out}-T_{w,in})$
 $\dot{m}_w=\rho*A_c*Vel$
 $A_c=N_{pass}*pi*D^2/4$

Vel [m/s]	N_{pass}
1	26.42
1.5	17.62
2	13.21
2.5	10.57
3	8.808
3.5	7.55
4	6.606
4.5	5.872
5	5.285
5.5	4.804
6	4.404
6.5	4.065
7	3.775
7.5	3.523
8	3.303



16-118 Cooling water is used to condense the steam in a power plant. The total length of the tubes required in the condenser is to be determined and a suitable HX type is to be proposed.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of condensation of steam at 30°C is given to be 2431 kJ/kg.

Analysis The temperature differences between the steam and the water at the two ends of condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 30^\circ\text{C} - 26^\circ\text{C} = 4^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 30^\circ\text{C} - 18^\circ\text{C} = 12^\circ\text{C}$$

and the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{4 - 12}{\ln(4/12)} = 7.28^\circ\text{C}$$

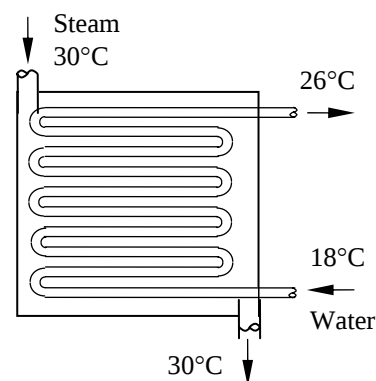
The heat transfer surface area is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{500 \times 10^6 \text{ W}}{(3500 \text{ W/m}^2 \cdot ^\circ\text{C})(7.28^\circ\text{C})} = 1.96 \times 10^4 \text{ m}^2$$

The total length of the tubes required in this condenser then becomes

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{1.96 \times 10^4 \text{ m}^2}{\pi (0.02 \text{ m})} = 3.123 \times 10^5 \text{ m} = \mathbf{312.3 \text{ km}}$$

A multi-pass shell-and-tube heat exchanger is suitable in this case.



16-119 Cold water is heated by hot water in a heat exchanger. The net rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the cold and hot water are given to be 4.18 and 4.19 kJ/kg·°C, respectively.

Analysis The temperature differences between the steam and the water at the two ends of condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 30^\circ\text{C} - 26^\circ\text{C} = 4^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 30^\circ\text{C} - 18^\circ\text{C} = 12^\circ\text{C}$$

and the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{4 - 12}{\ln(4/12)} = 7.28^\circ\text{C}$$

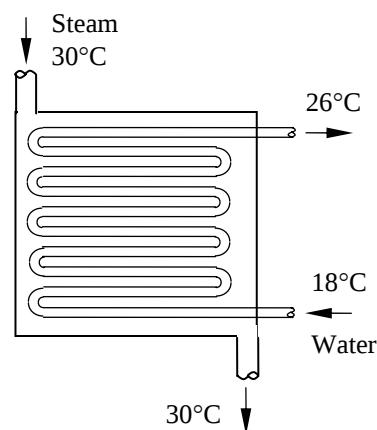
The heat transfer surface area is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{50 \times 10^6 \text{ W}}{(3500 \text{ W/m}^2 \cdot ^\circ\text{C})(7.28^\circ\text{C})} = 1962 \text{ m}^2$$

The total length of the tubes required in this condenser then becomes

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{1962 \text{ m}^2}{\pi(0.02 \text{ m})} = 31,231 \text{ m} = \mathbf{31.23 \text{ km}}$$

A multi-pass shell-and-tube heat exchanger is suitable in this case.



Review Problems

16-120 The inlet conditions of hot and cold fluid streams in a heat exchanger are given. The outlet temperatures of both streams are to be determined using LMTD and the effectiveness-NTU methods.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heats of hot and cold fluid streams are given to be 2.0 and 4.2 kJ/kg·°C, respectively.

Analysis (a) The rate of heat transfer can be expressed as

$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) = (2700 / 3600 \text{ kg/s})(2.0 \text{ kJ/kg} \cdot ^\circ\text{C})(120 - T_{h,out}) = 1.5(120 - T_{h,out}) \quad (1)$$

$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) = (1800 / 3600 \text{ kg/s})(4.2 \text{ kJ/kg} \cdot ^\circ\text{C})(T_{c,out} - 20) = 2.1(T_{c,out} - 20) \quad (2)$$

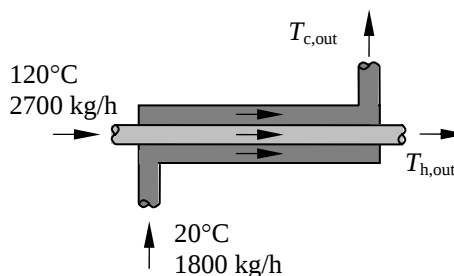
The heat transfer can also be expressed using the logarithmic mean temperature difference as

$$\Delta T_1 = T_{h,in} - T_{c,in} = 120^\circ\text{C} - 20^\circ\text{C} = 100^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)}$$

$$\begin{aligned} \dot{Q} &= UA\Delta T_{lm} = \frac{\dot{Q}_{hc,m}}{A\Delta T_{lm}} \\ &= (2.0 \text{ kW/m}^2 \cdot ^\circ\text{C})(0.50 \text{ m}^2) \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)} = \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)} \quad (3) \end{aligned}$$



Now we have three expressions for heat transfer with three unknowns: $\dot{Q}$, $T_{h,out}$, $T_{c,out}$. Solving them using an equation solver such as EES, we obtain

$$\dot{Q} = 59.6 \text{ kW}$$

$$T_{h,out} = 80.3^\circ\text{C}$$

$$T_{c,out} = 48.4^\circ\text{C}$$

(b) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (2700/3600 \text{ kg/s})(2.0 \text{ kJ/kg} \cdot ^\circ\text{C}) = 1.5 \text{ kW/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (1800/3600 \text{ kg/s})(4.2 \text{ kJ/kg} \cdot ^\circ\text{C}) = 2.1 \text{ kW/}^\circ\text{C}$$

Therefore

$$C_{\min} = C_h = 1.5 \text{ kW/}^\circ\text{C}$$

which is the smaller of the two heat capacity rates. The heat capacity ratio and the NTU are

$$c = \frac{C_{\min}}{C_{\max}} = \frac{1.5}{2.1} = 0.714$$

$$NTU = \frac{UA}{C_{\min}} = \frac{(2.0 \text{ kW/m}^2 \cdot \text{C})(0.50 \text{ m}^2)}{1.5 \text{ kW/}^\circ\text{C}} = 0.667$$

The effectiveness of this parallel-flow heat exchanger is

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-(0.667)(1+0.714)]}{1+0.714} = 0.397$$

The maximum heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (1.5 \text{ kW/}^\circ\text{C})(120^\circ\text{C} - 20^\circ\text{C}) = 150 \text{ kW}$$

The actual heat transfer rate is

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.397)(150) = 59.6 \text{ kW}$$

Then the outlet temperatures are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20^\circ\text{C} + \frac{59.6 \text{ kW}}{2.1 \text{ kW/}^\circ\text{C}} = \mathbf{48.4^\circ\text{C}}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 120^\circ\text{C} - \frac{59.6 \text{ kW}}{1.5 \text{ kW/}^\circ\text{C}} = \mathbf{80.3^\circ\text{C}}$$

Discussion The results obtained by two methods are same as expected. However, the effectiveness-NTU method is easier for this type of problems.

16-121 Water is used to cool a process stream in a shell and tube heat exchanger. The tube length is to be determined for one tube pass and four tube pass cases.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The properties of process stream and water are given in problem statement.

Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (47 \text{ kg/s})(3.5 \text{ kJ/kg} \cdot ^\circ\text{C})(160 - 100)^\circ\text{C} = 9870 \text{ kW}$$

The outlet temperature of water is determined from

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in})$$

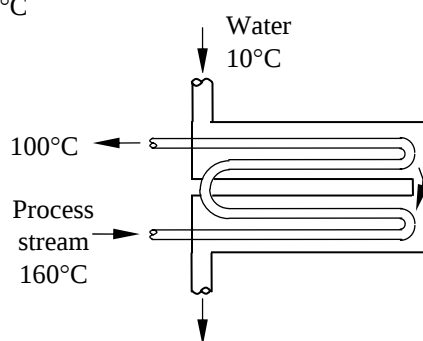
$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_c} = 10^\circ\text{C} + \frac{9870 \text{ kW}}{(66 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})} = 45.8^\circ\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 160^\circ\text{C} - 45.8^\circ\text{C} = 114.2^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 100^\circ\text{C} - 10^\circ\text{C} = 90^\circ\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{114.2 - 90}{\ln\left(\frac{114.2}{90}\right)} = 101.6^\circ\text{C}$$



The Reynolds number is

$$V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{N_{tube} \rho \pi D^2 / 4} = \frac{(47 \text{ kg/s})}{(100)(950 \text{ kg/m}^3) \pi (0.025 \text{ m})^2 / 4} = 1.008 \text{ m/s}$$

$$\text{Re} = \frac{VD\rho}{\mu} = \frac{(1.008 \text{ m/s})(0.025 \text{ m})(950 \text{ kg/m}^3)}{0.002 \text{ kg/m} \cdot \text{s}} = 11,968$$

which is greater than 10,000. Therefore, we have turbulent flow. We assume fully developed flow and evaluate the Nusselt number from

$$\text{Pr} = \frac{\mu c_p}{k} = \frac{(0.002 \text{ kg/m} \cdot \text{s})(3500 \text{ J/kg} \cdot ^\circ\text{C})}{0.50 \text{ W/m} \cdot ^\circ\text{C}} = 14$$

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(11,968)^{0.8} (14)^{0.3} = 92.9$$

Heat transfer coefficient on the inner surface of the tubes is

$$h_i = \frac{k}{D} \text{Nu} = \frac{0.50 \text{ W/m} \cdot ^\circ\text{C}}{0.025 \text{ m}} (92.9) = 1858 \text{ W/m}^2 \cdot ^\circ\text{C}$$

Disregarding the thermal resistance of the tube wall the overall heat transfer coefficient is determined from

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{1858} + \frac{1}{4000}} = 1269 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The correction factor for one shell pass and one tube pass heat exchanger is $F = 1$. The tube length is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$9870 \text{ kW} = (1.269 \text{ kW/m}^2 \cdot \text{C})[100\pi(0.025 \text{ m})L](1)(101.6^\circ\text{C})$$

$$L = \mathbf{9.75 \text{ m}}$$

(b) For 1 shell pass and 4 tube passes, there are $100/4=25$ tubes per pass and this will increase the velocity fourfold. We repeat the calculations for this case as follows:

$$V = 4 \times 1.008 = 4.032 \text{ m/s}$$

$$\text{Re} = 4 \times 11,968 = 47,872$$

$$Nu = \frac{hD}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(47,872)^{0.8} (14)^{0.3} = 281.6$$

$$h_i = \frac{k}{D} Nu = \frac{0.50 \text{ W/m} \cdot ^\circ\text{C}}{0.025 \text{ m}} (281.6) = 5632 \text{ W/m}^2 \cdot ^\circ\text{C}$$

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{5632} + \frac{1}{4000}} = 2339 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The correction factor is determined from Fig. 16-18:

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{100 - 160}{10 - 160} = 0.4 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{10 - 45.8}{100 - 160} = 0.60 \end{aligned} \right\} F = 0.96$$

The tube length is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$9870 \text{ kW} = (2.339 \text{ kW/m}^2 \cdot \text{C})[100\pi(0.025 \text{ m})L](0.96)(101.6^\circ\text{C})$$

$$L = \mathbf{5.51 \text{ m}}$$

16-122 A hydrocarbon stream is heated by a water stream in a 2-shell passes and 4-tube passes heat exchanger. The rate of heat transfer and the mass flow rates of both fluid streams and the fouling factor after usage are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The specific heat of HC is given to be 2 kJ/kg.°C. The specific heat of water is taken to be 4.18 kJ/kg.°C.

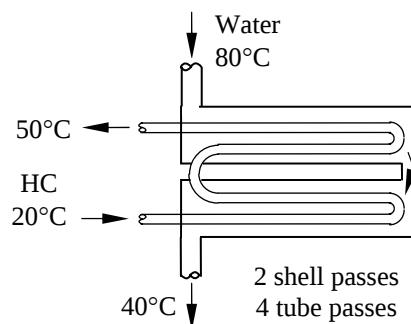
Analysis (a) The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80^\circ\text{C} - 50^\circ\text{C} = 30^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40^\circ\text{C} - 20^\circ\text{C} = 20^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{30 - 20}{\ln(30 / 20)} = 24.66^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{50 - 20}{80 - 20} = 0.5 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80 - 40}{50 - 20} = 1.33 \end{aligned} \right\} F = 0.90 \text{ (Fig. 16-18)}$$



The overall heat transfer coefficient of the heat exchanger is

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{1600} + \frac{1}{2500}} = 975.6 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (975.6 \text{ W/m}^2 \cdot ^\circ\text{C}) [160\pi(0.02 \text{ m})(1.5 \text{ m})] (0.90)(24.66^\circ\text{C}) = 3.265 \times 10^5 \text{ W} = \mathbf{326.5 \text{ kW}}$$

The mass flow rates of fluid streams are

$$\dot{m}_c = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{326.5 \text{ kW}}{(2.0 \text{ kJ/kg} \cdot ^\circ\text{C})(50^\circ\text{C} - 20^\circ\text{C})} = \mathbf{5.44 \text{ kg/s}}$$

$$\dot{m}_h = \frac{\dot{Q}}{c_p (T_{in} - T_{out})} = \frac{326.5 \text{ kW}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(80^\circ\text{C} - 40^\circ\text{C})} = \mathbf{1.95 \text{ kg/s}}$$

(b) The rate of heat transfer in this case is

$$\dot{Q} = [\dot{m}_c (T_{out} - T_{in})]_c = (5.44 \text{ kg/s})(2.0 \text{ kJ/kg} \cdot ^\circ\text{C})(45^\circ\text{C} - 20^\circ\text{C}) = 272 \text{ kW}$$

This corresponds to a 17% decrease in heat transfer. The outlet temperature of the hot fluid is

$$\begin{aligned} \dot{Q} &= [\dot{m}_h (T_{in} - T_{out})]_h \\ 272 \text{ kW} &= (1.95 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(80^\circ\text{C} - T_{h,out}) \\ T_{h,out} &= 46.6^\circ\text{C} \end{aligned}$$

The logarithmic temperature difference is

$$\begin{aligned} \Delta T_1 &= T_{h,in} - T_{c,out} = 80^\circ\text{C} - 45^\circ\text{C} = 35^\circ\text{C} \\ \Delta T_2 &= T_{h,out} - T_{c,in} = 46.6^\circ\text{C} - 20^\circ\text{C} = 26.6^\circ\text{C} \end{aligned}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{35 - 26.6}{\ln(35 / 26.6)} = 30.61^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{45 - 20}{80 - 20} = 0.42 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80 - 46.6}{45 - 20} = 1.34 \end{aligned} \right\} F = 0.97 \text{ (Fig. 16-18)}$$

The overall heat transfer coefficient is

$$\begin{aligned} \dot{Q} &= UA_s F \Delta T_{lm,CF} \\ 272,000 \text{ W} &= U [160\pi(0.02 \text{ m})(1.5 \text{ m})] (0.97)(30.61^\circ\text{C}) \\ U &= 607.5 \text{ W/m}^2 \cdot ^\circ\text{C} \end{aligned}$$

The fouling factor is determined from

$$R_f = \frac{1}{U_{dirty}} - \frac{1}{U_{clean}} = \frac{1}{607.5} - \frac{1}{975.6} = \mathbf{6.21 \times 10^{-4} \text{ m}^2 \cdot ^\circ\text{C/W}}$$

16-123 Hot water is cooled by cold water in a 1-shell pass and 2-tube passes heat exchanger. The mass flow rates of both fluid streams are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant. 5 There is no fouling.

Properties The specific heats of both cold and hot water streams are taken to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$.

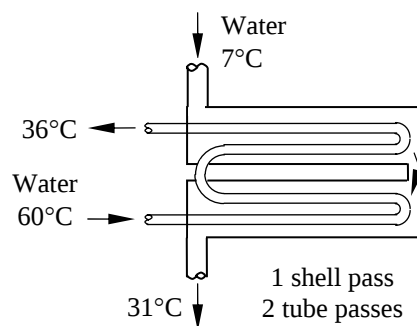
Analysis The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 60^\circ\text{C} - 31^\circ\text{C} = 29^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 36^\circ\text{C} - 7^\circ\text{C} = 29^\circ\text{C}$$

Since $\Delta T_1 = \Delta T_2$, we have $\Delta T_{lm,CF} = 29^\circ\text{C}$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{31 - 60}{7 - 60} = 0.45 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{7 - 31}{36 - 60} = 1.0 \end{aligned} \right\} F = 0.88 \text{ (Fig. 16-18)}$$



The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (950 \text{ W/m}^2 \cdot ^\circ\text{C})(15 \text{ m}^2)(0.88)(29^\circ\text{C}) = 3.64 \times 10^5 \text{ W} = 364 \text{ kW}$$

The mass flow rates of fluid streams are

$$\dot{m}_c = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{364 \text{ kW}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(60^\circ\text{C} - 36^\circ\text{C})} = \mathbf{3.63 \text{ kg/s}}$$

$$\dot{m}_h = \frac{\dot{Q}}{c_p (T_{in} - T_{out})} = \frac{364 \text{ kW}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(31^\circ\text{C} - 7^\circ\text{C})} = \mathbf{3.63 \text{ kg/s}}$$

16-124 Hot oil is cooled by water in a multi-pass shell-and-tube heat exchanger. The overall heat transfer coefficient based on the inner surface is to be determined.

Assumptions 1 Water flow is fully developed. 2 Properties of the water are constant.

Properties The properties of water at 25°C are (Table A-15)

$$k = 0.607 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = \mu / \rho = 0.894 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 6.14$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(3 \text{ m/s})(0.013 \text{ m})}{0.894 \times 10^{-6} \text{ m}^2/\text{s}} = 43,624$$

which is greater than 10,000. Therefore, we assume fully developed turbulent flow, and determine Nusselt number from

$$\text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(43,624)^{0.8} (6.14)^{0.4} = 245$$

and

$$h_i = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m} \cdot ^\circ\text{C}}{0.013 \text{ m}} (245) = 11,440 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The inner and the outer surface areas of the tube are

$$A_i = \pi D_i L = \pi(0.013 \text{ m})(1 \text{ m}) = 0.04084 \text{ m}^2$$

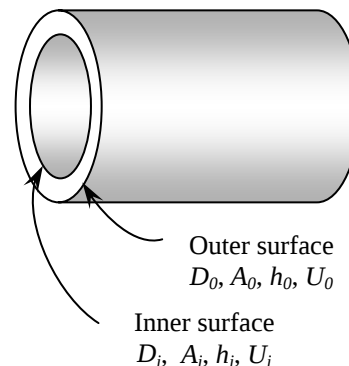
$$A_o = \pi D_o L = \pi(0.015 \text{ m})(1 \text{ m}) = 0.04712 \text{ m}^2$$

The total thermal resistance of this heat exchanger per unit length is

$$\begin{aligned} R &= \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{1}{h_o A_o} \\ &= \frac{1}{(11,440 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04084 \text{ m}^2)} + \frac{\ln(1.5/1.3)}{2\pi(110 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} + \frac{1}{(35 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04712 \text{ m}^2)} \\ &= 0.609^\circ\text{C/W} \end{aligned}$$

Then the overall heat transfer coefficient of this heat exchanger based on the inner surface becomes

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{R A_i} = \frac{1}{(0.609^\circ\text{C/W})(0.04084 \text{ m}^2)} = \mathbf{40.2 \text{ W/m}^2 \cdot ^\circ\text{C}}$$



16-125 Hot oil is cooled by water in a multi-pass shell-and-tube heat exchanger. The overall heat transfer coefficient based on the inner surface is to be determined.

Assumptions 1 Water flow is fully developed. 2 Properties of the water are constant.

Properties The properties of water at 25°C are (Table A-15)

$$k = 0.607 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = \mu / \rho = 0.894 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Pr} = 6.14$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(3 \text{ m/s})(0.013 \text{ m})}{0.894 \times 10^{-6} \text{ m}^2/\text{s}} = 43,624$$

which is greater than 10,000. Therefore, we assume fully developed turbulent flow, and determine Nusselt number from

$$\text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(43,624)^{0.8} (6.14)^{0.4} = 245$$

and

$$h_i = \frac{k}{D} \text{Nu} = \frac{0.607 \text{ W/m} \cdot ^\circ\text{C}}{0.013 \text{ m}} (245) = 11,440 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The inner and the outer surface areas of the tube are

$$A_i = \pi D_i L = \pi (0.013 \text{ m})(1 \text{ m}) = 0.04084 \text{ m}^2$$

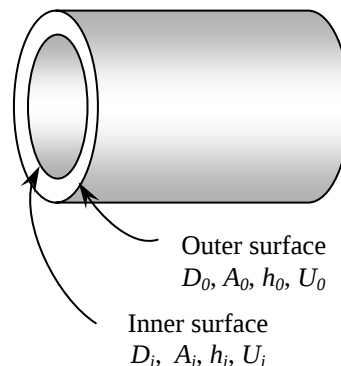
$$A_o = \pi D_o L = \pi (0.015 \text{ m})(1 \text{ m}) = 0.04712 \text{ m}^2$$

The total thermal resistance of this heat exchanger per unit length of it with a fouling factor is

$$\begin{aligned} R &= \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o} \\ &= \frac{1}{(11,440 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04084 \text{ m}^2)} + \frac{\ln(1.5/1.3)}{2\pi (110 \text{ W/m} \cdot ^\circ\text{C})(1 \text{ m})} \\ &\quad + \frac{0.0004 \text{ m}^2 \cdot ^\circ\text{C/W}}{0.04712 \text{ m}^2} + \frac{1}{(35 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04712 \text{ m}^2)} \\ &= 0.617^\circ\text{C/W} \end{aligned}$$

Then the overall heat transfer coefficient of this heat exchanger based on the inner surface becomes

$$R = \frac{1}{U_i A_i} \rightarrow U_i = \frac{1}{R A_i} = \frac{1}{(0.617^\circ\text{C/W})(0.04084 \text{ m}^2)} = \mathbf{39.7 \text{ W/m}^2 \cdot ^\circ\text{C}}$$



16-126 Water is heated by hot oil in a multi-pass shell-and-tube heat exchanger. The rate of heat transfer and the heat transfer surface area on the outer side of the tube are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.

Analysis (a) The rate of heat transfer in this heat exchanger is

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h,in} - T_{h,out}) = (3 \text{ kg/s})(2.2 \text{ kJ/kg.°C})(130^\circ\text{C} - 60^\circ\text{C}) = \mathbf{462 \text{ kW}}$$

(b) The outlet temperature of the cold water is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 20^\circ\text{C} + \frac{462 \text{ kW}}{(3 \text{ kg/s})(4.18 \text{ kJ/kg.°C})} = 56.8^\circ\text{C}$$

The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 130^\circ\text{C} - 56.8^\circ\text{C} = 73.2^\circ\text{C}$$

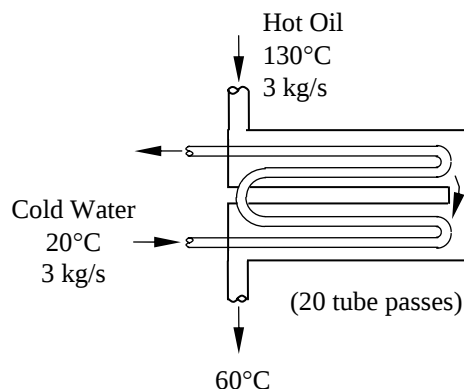
$$\Delta T_2 = T_{h,out} - T_{c,in} = 60^\circ\text{C} - 20^\circ\text{C} = 40^\circ\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{73.2 - 40}{\ln(73.2 / 40)} = 54.9^\circ\text{C}$$

and

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{56.8 - 20}{130 - 20} = 0.335 \\ R &= \frac{T_2 - T_1}{t_2 - t_1} = \frac{130 - 60}{56.8 - 20} = 1.90 \end{aligned} \right\} F = 0.96$$



The heat transfer surface area on the outer side of the tube is then determined from

$$\dot{Q} = UA_s F \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{UF \Delta T_{lm}} = \frac{462 \text{ kW}}{(0.22 \text{ kW/m}^2 \cdot ^\circ\text{C})(0.96)(54.9^\circ\text{C})} = \mathbf{39.8 \text{ m}^2}$$

16-127E Water is heated by solar-heated hot air in a double-pipe counter-flow heat exchanger. The required length of the tube is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and air are given to be 1.0 and 0.24 Btu/lbm.°F, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h,in} - T_{h,out}) = (0.7 \text{ lbm/s})(0.24 \text{ Btu/lbm.°F})(190^\circ\text{F} - 135^\circ\text{F}) = 9.24 \text{ Btu/s}$$

The outlet temperature of the cold water is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 70^\circ\text{F} + \frac{9.24 \text{ Btu/s}}{(0.35 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})} = 96.4^\circ\text{F}$$

The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 190^\circ\text{F} - 96.4^\circ\text{F} = 93.6^\circ\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 135^\circ\text{F} - 70^\circ\text{F} = 65^\circ\text{F}$$

The logarithmic mean temperature difference is

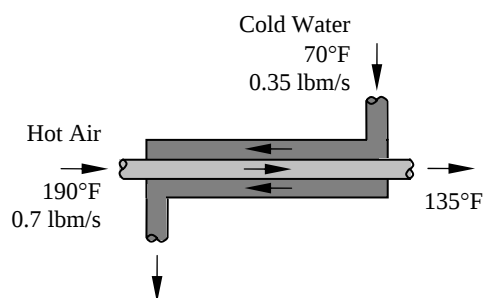
$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{93.6 - 65}{\ln(93.6 / 65)} = 78.43^\circ\text{F}$$

The heat transfer surface area on the outer side of the tube is determined from

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{9.24 \text{ Btu/s}}{(20 / 3600 \text{ Btu/s.ft}^2 \cdot ^\circ\text{F})(78.43^\circ\text{F})} = 21.21 \text{ ft}^2$$

Then the length of the tube required becomes

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{21.21 \text{ ft}^2}{\pi(0.5 / 12 \text{ ft})} = \mathbf{162.0 \text{ ft}}$$



16-128 It is to be shown that when $\Delta T_1 = \Delta T_2$ for a heat exchanger, the ΔT_{lm} relation reduces to $\Delta T_{lm} = \Delta T_1 = \Delta T_2$.

Analysis When $\Delta T_1 = \Delta T_2$, we obtain

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{0}{0}$$

This case can be handled by applying L'Hospital's rule (taking derivatives of nominator and denominator separately with respect to ΔT_1 or ΔT_2). That is,

$$\Delta T_{lm} = \frac{d(\Delta T_1 - \Delta T_2) / d\Delta T_1}{d[\ln(\Delta T_1 / \Delta T_2)] / d\Delta T_1} = \frac{1}{1 / \Delta T_1} = \Delta T_1 = \Delta T_2$$

16-129 Refrigerant-134a is condensed by air in the condenser of a room air conditioner. The heat transfer area on the refrigerant side is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of air is given to be 1.005 kJ/kg.°C.

Analysis The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 40^\circ\text{C} - 35^\circ\text{C} = 5^\circ\text{C}$$

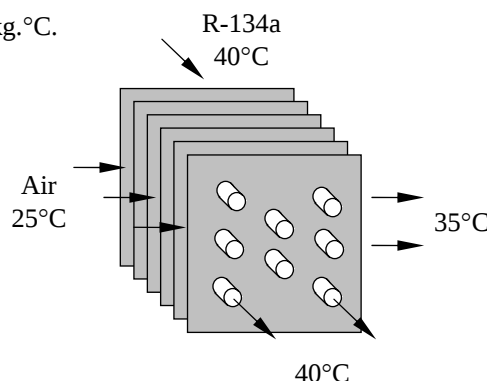
$$\Delta T_2 = T_{h,out} - T_{c,in} = 40^\circ\text{C} - 25^\circ\text{C} = 15^\circ\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{5 - 15}{\ln(5/15)} = 9.1^\circ\text{C}$$

The heat transfer surface area on the outer side of the tube is determined from

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{lm}} = \frac{(15,000 / 3600) \text{ kW}}{(0.150 \text{ kW/m}^2 \cdot ^\circ\text{C})(9.1^\circ\text{C})} = \mathbf{3.05 \text{ m}^2}$$



16-130 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.1 kJ/kg.°C, respectively.

Analysis The rate of heat transfer is simply

$$\dot{Q} = [\dot{m} c_p (T_{in} - T_{out})]_{\text{gas.}} = (0.65 \text{ kg/s})(1.1 \text{ kJ/kg} \cdot ^\circ\text{C})(180^\circ\text{C} - 95^\circ\text{C}) = \mathbf{60.8 \text{ kW}}$$

16-131 A water-to-water heat exchanger is proposed to preheat the incoming cold water by the drained hot water in a plant to save energy. The heat transfer rating of the heat exchanger and the amount of money this heat exchanger will save are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the hot water is given to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis The maximum rate of heat transfer is

$$\begin{aligned}\dot{Q}_{\max} &= \dot{m}_h c_{ph} (T_{h,in} - T_{c,in}) \\ &= (8 / 60 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(60^\circ\text{C} - 14^\circ\text{C}) \\ &= 25.6 \text{ kW}\end{aligned}$$

Noting that the heat exchanger will recover 72% of it, the actual heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.72)(25.6 \text{ kJ/s}) = 18.43 \text{ kW}$$

which is the heat transfer rating. The operating hours per year are

$$\text{The annual operating hours} = (8 \text{ h/day})(5 \text{ days/week})(52 \text{ week/year}) = 2080 \text{ h/year}$$

The energy saved during the entire year will be

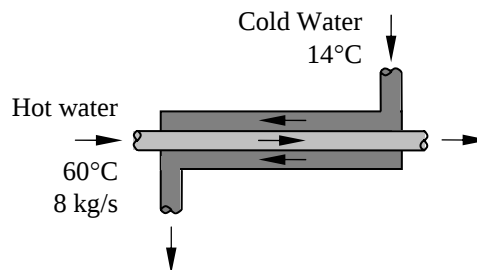
$$\begin{aligned}\text{Energy saved} &= (\text{heat transfer rate})(\text{operating time}) \\ &= (18.43 \text{ kJ/s})(2080 \text{ h/year})(3600 \text{ s/h}) \\ &= 1.38 \times 10^8 \text{ kJ/year}\end{aligned}$$

Then amount of fuel and money saved will be

$$\begin{aligned}\text{Fuel saved} &= \frac{\text{Energy saved}}{\text{Furnace efficiency}} = \frac{1.38 \times 10^8 \text{ kJ/year}}{0.78} \left(\frac{1 \text{ therm}}{105,500 \text{ kJ}} \right) \\ &= 1677 \text{ therms/year}\end{aligned}$$

Money saved = (fuel saved)(the price of fuel)

$$= (1677 \text{ therms/year})(\$1.00/\text{therm}) = \mathbf{\$1677/\text{year}}$$



16-132 A shell-and-tube heat exchanger is used to heat water with geothermal steam condensing. The rate of heat transfer, the rate of condensation of steam, and the overall heat transfer coefficient are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant.

Properties The heat of vaporization of geothermal water at 120°C is given to be $h_{fg} = 2203 \text{ kJ/kg}$ and specific heat of water is given to be $c_p = 4180 \text{ J/kg} \cdot ^\circ\text{C}$.

Analysis (a) The outlet temperature of the water is

$$T_{c,\text{out}} = T_{h,\text{out}} - 46 = 120^\circ\text{C} - 46^\circ\text{C} = 74^\circ\text{C}$$

Then the rate of heat transfer becomes

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \\ &= (3.9 \text{ kg/s})(4180 \text{ J/kg} \cdot ^\circ\text{C})(74^\circ\text{C} - 22^\circ\text{C}) \\ &= \mathbf{847.7 \text{ kW}}\end{aligned}$$

(b) The rate of condensation of steam is determined from

$$\begin{aligned}\dot{Q} &= (\dot{m}h_{fg})_{\text{geothermal steam}} \\ 847.7 \text{ kW} &= \dot{m}(2203 \text{ kJ/kg}) \longrightarrow \dot{m} = \mathbf{0.385 \text{ kg/s}}\end{aligned}$$

(c) The heat transfer area is

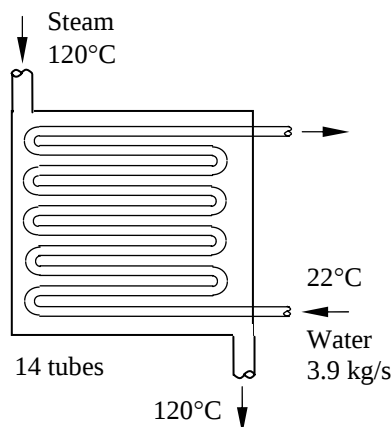
$$A_i = n\pi D_i L = 14\pi(0.024 \text{ m})(3.2 \text{ m}) = 3.378 \text{ m}^2$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\begin{aligned}\Delta T_1 &= T_{h,\text{in}} - T_{c,\text{out}} = 120^\circ\text{C} - 74^\circ\text{C} = 46^\circ\text{C} \\ \Delta T_2 &= T_{h,\text{out}} - T_{c,\text{in}} = 120^\circ\text{C} - 22^\circ\text{C} = 98^\circ\text{C} \\ \Delta T_{\text{lm,CF}} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{46 - 98}{\ln(46 / 98)} = 68.8^\circ\text{C} \\ \left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{74 - 22}{120 - 22} = 0.53 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{120 - 120}{74 - 22} = 0 \end{aligned} \right\} F = 1\end{aligned}$$

Then the overall heat transfer coefficient is determined to be

$$\dot{Q} = U_i A_i F \Delta T_{\text{lm,CF}} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{\text{lm,CF}}} = \frac{847,700 \text{ W}}{(3.378 \text{ m}^2)(1)(68.8^\circ\text{C})} = \mathbf{3650 \text{ W/m}^2 \cdot ^\circ\text{C}}$$



16-133 Water is heated by geothermal water in a double-pipe counter-flow heat exchanger. The mass flow rate of the geothermal water and the outlet temperatures of both fluids are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the geothermal water and the cold water are given to be 4.25 and 4.18 kJ/kg·°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = \dot{m}_h (4.25 \text{ kJ/kg} \cdot ^\circ\text{C}) = 4.25 \dot{m}_h$$

$$C_c = \dot{m}_c c_{pc} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 5.016 \text{ kW/}^\circ\text{C}$$

$$C_{\min} = C_c = 5.016 \text{ kW/}^\circ\text{C}$$

$$\text{and } c = \frac{C_{\min}}{C_{\max}} = \frac{5.016}{4.25 \dot{m}_h} = \frac{1.1802}{\dot{m}_h}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.480 \text{ kW/m}^2 \cdot ^\circ\text{C})(25 \text{ m}^2)}{5.016 \text{ kW/}^\circ\text{C}} = 2.392$$

Using the effectiveness relation, we find the capacity ratio

$$\varepsilon = \frac{1 - \exp[-NTU(1-c)]}{1 - c \exp[-NTU(1-c)]} \longrightarrow 0.823 = \frac{1 - \exp[-2.392(1-c)]}{1 - c \exp[-2.392(1-c)]} \longrightarrow c = 0.494$$

Then the mass flow rate of geothermal water is determined from

$$c = \frac{1.1802}{\dot{m}_h} \longrightarrow 0.494 = \frac{1.1802}{\dot{m}_h} \longrightarrow \dot{m}_h = \mathbf{2.39 \text{ kg/s}}$$

The maximum heat transfer rate is

$$\dot{Q}_{\max} = C_{\min} (T_{h,\text{in}} - T_{c,\text{in}}) = (5.016 \text{ kW/}^\circ\text{C})(75^\circ\text{C} - 17^\circ\text{C}) = 290.9 \text{ kW}$$

Then the actual rate of heat transfer rate becomes

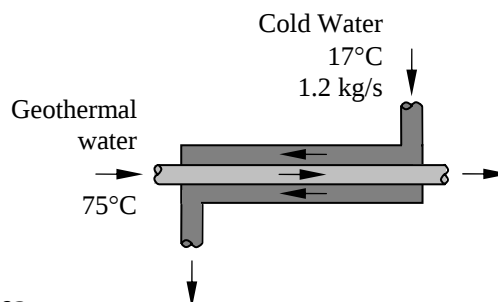
$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.823)(290.9 \text{ kW}) = 239.4 \text{ kW}$$

The outlet temperatures of the geothermal and cold waters are determined to be

$$\dot{Q} = C_c (T_{c,\text{out}} - T_{c,\text{in}}) \longrightarrow 239.4 \text{ kW} = (5.016 \text{ kW/}^\circ\text{C})(T_{c,\text{out}} - 17) \longrightarrow T_{c,\text{out}} = \mathbf{64.7^\circ\text{C}}$$

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h,\text{in}} - T_{h,\text{out}})$$

$$239.4 \text{ kW} = (2.39 \text{ kg/s})(4.25 \text{ kJ/kg} \cdot ^\circ\text{C})(75 - T_{h,\text{out}}) \longrightarrow T_{h,\text{out}} = \mathbf{51.4^\circ\text{C}}$$



16-134 Air is to be heated by hot oil in a cross-flow heat exchanger with both fluids unmixed. The effectiveness of the heat exchanger, the mass flow rate of the cold fluid, and the rate of heat transfer are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the air and the oil are given to be 1.006 and 2.15 kJ/kg·°C, respectively.

Analysis (a) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = 0.5 \dot{m}_c (2.15 \text{ kJ/kg} \cdot ^\circ\text{C}) = 1.075 \dot{m}_c$$

$$C_c = \dot{m}_c c_{pc} = \dot{m}_c (1.006 \text{ kJ/kg} \cdot ^\circ\text{C}) = 1.006 \dot{m}_c$$

Therefore, $C_{\min} = C_c = 1.006 \dot{m}_c$

$$\text{and } c = \frac{C_{\min}}{C_{\max}} = \frac{1.006 \dot{m}_c}{1.075 \dot{m}_c} = 0.936$$

The effectiveness of the heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c (T_{c,\text{out}} - T_{c,\text{in}})}{C_c (T_{h,\text{in}} - T_{c,\text{in}})} = \frac{58 - 18}{80 - 18} = \mathbf{0.645}$$

(b) The NTU of this heat exchanger is expressed as

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.750 \text{ kW}/^\circ\text{C})}{1.006 \dot{m}_c} = \frac{0.7455}{\dot{m}_c}$$

The NTU of this heat exchanger can also be determined from

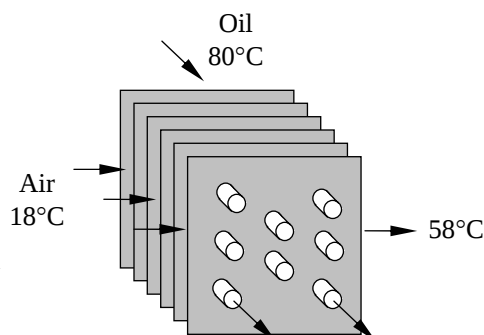
$$NTU = -\frac{\ln[c \ln(1 - \varepsilon) + 1]}{c} = -\frac{\ln[0.936 \times \ln(1 - 0.645) + 1]}{0.936} = 3.724$$

Then the mass flow rate of the air is determined to be

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow 3.724 = \frac{(0.750 \text{ kW}/^\circ\text{C})}{1.006 \dot{m}_c} \longrightarrow \dot{m}_c = \mathbf{0.20 \text{ kg/s}}$$

(c) The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,\text{out}} - T_{c,\text{in}}) = (0.20 \text{ kg/s})(1.006 \text{ kJ/kg} \cdot ^\circ\text{C})(58 - 18)^\circ\text{C} = \mathbf{8.05 \text{ kW}}$$



16-135 A water-to-water counter-flow heat exchanger is considered. The outlet temperature of the cold water, the effectiveness of the heat exchanger, the mass flow rate of the cold water, and the heat transfer rate are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of both the cold and the hot water are given to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$.

Analysis (a) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = 1.5 \dot{m}_c (4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 6.27 \dot{m}_c$$

$$C_c = \dot{m}_c c_{pc} = \dot{m}_c (4.18 \text{ kJ/kg} \cdot ^\circ\text{C}) = 4.18 \dot{m}_c$$

Therefore, $C_{\min} = C_c = 4.18 \dot{m}_c$

$$\text{and } C = \frac{C_{\min}}{C_{\max}} = \frac{4.18 \dot{m}_c}{6.27 \dot{m}_c} = 0.667$$

The rate of heat transfer can be expressed as

$$\dot{Q} = C_c (T_{c,\text{out}} - T_{c,\text{in}}) = (4.18 \dot{m}_c) (T_{c,\text{out}} - 20)$$

$$\dot{Q} = C_h (T_{h,\text{in}} - T_{h,\text{out}}) = (6.27 \dot{m}_c) [95 - (T_{c,\text{out}} + 15)] = (6.27 \dot{m}_c) (80 - T_{c,\text{out}})$$

Setting the above two equations equal to each other we obtain the outlet temperature of the cold water

$$\dot{Q} = 4.18 \dot{m}_c (T_{c,\text{out}} - 20) = 6.27 \dot{m}_c (80 - T_{c,\text{out}}) \longrightarrow T_{c,\text{out}} = \mathbf{56^\circ\text{C}}$$

(b) The effectiveness of the heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c (T_{c,\text{out}} - T_{c,\text{in}})}{C_c (T_{h,\text{in}} - T_{c,\text{in}})} = \frac{4.18 \dot{m}_c (56 - 20)}{4.18 \dot{m}_c (95 - 20)} = \mathbf{0.48}$$

(c) The NTU of this heat exchanger is determined from

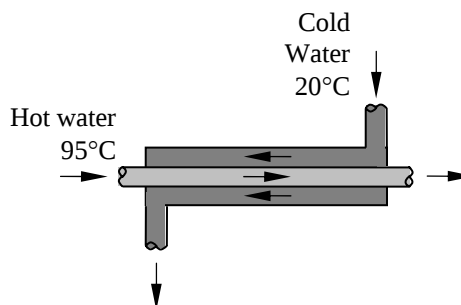
$$NTU = \frac{1}{C - 1} \ln \left(\frac{\varepsilon - 1}{\varepsilon C - 1} \right) = \frac{1}{0.667 - 1} \ln \left(\frac{0.48 - 1}{0.48 \times 0.667 - 1} \right) = 0.805$$

Then, from the definition of NTU, we obtain the mass flow rate of the cold fluid:

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow 0.805 = \frac{1.400 \text{ kW}/^\circ\text{C}}{4.18 \dot{m}_c} \longrightarrow \dot{m}_c = \mathbf{0.416 \text{ kg/s}}$$

(d) The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c,\text{out}} - T_{c,\text{in}}) = (0.416 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(56 - 20)^\circ\text{C} = \mathbf{62.6 \text{ kW}}$$



16-136 Oil is cooled by water in a 2-shell passes and 4-tube passes heat exchanger. The mass flow rate of water and the surface area are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant. 5 There is no fouling.

Properties The specific heat of oil is given to be $2 \text{ kJ/kg} \cdot ^\circ\text{C}$. The specific heat of water is taken to be $4.18 \text{ kJ/kg} \cdot ^\circ\text{C}$.

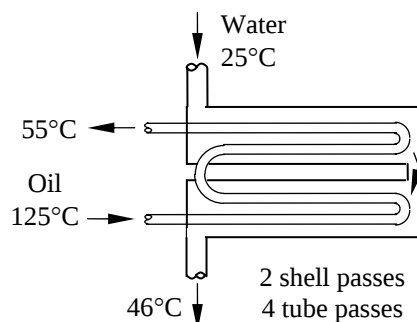
Analysis The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 125^\circ\text{C} - 46^\circ\text{C} = 79^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 55^\circ\text{C} - 25^\circ\text{C} = 30^\circ\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{79 - 30}{\ln(79 / 30)} = 50.61^\circ\text{C}$$

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{55 - 125}{25 - 125} = 0.7 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{125 - 46}{55 - 25} = 0.3 \end{aligned} \right\} F = 0.97 \text{ (Fig. 16-18)}$$



The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (10 \text{ kg/s})(2.0 \text{ kJ/kg} \cdot ^\circ\text{C})(125 - 55)^\circ\text{C} = 1400 \text{ kW}$$

The mass flow rate of water is

$$\dot{m}_w = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{1400 \text{ kW}}{(4.18 \text{ kJ/kg} \cdot ^\circ\text{C})(46^\circ\text{C} - 25^\circ\text{C})} = \mathbf{15.9 \text{ kg/s}}$$

The surface area of the heat exchanger is determined to be

$$\begin{aligned} \dot{Q} &= UAF\Delta T_{lm} \\ 1400 \text{ kW} &= (0.9 \text{ kW/m}^2 \cdot ^\circ\text{C}) A_s (0.97)(50.61^\circ\text{C}) \\ A_s &= \mathbf{31.7 \text{ m}^2} \end{aligned}$$

16-137 A polymer solution is heated by ethylene glycol in a parallel-flow heat exchanger. The rate of heat transfer, the outlet temperature of polymer solution, and the mass flow rate of ethylene glycol are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant. 5 There is no fouling.

Properties The specific heats of polymer and ethylene glycol are given to be 2.0 and 2.5 kJ/kg·°C, respectively.

Analysis (a) The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,in} = 60^\circ\text{C} - 20^\circ\text{C} = 40^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = 15^\circ\text{C}$$

$$\Delta T_{lm,PF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 15}{\ln(40 / 15)} = 25.49^\circ\text{C}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s \Delta T_{lm} = (240 \text{ W/m}^2 \cdot ^\circ\text{C})(0.8 \text{ m}^2)(25.49^\circ\text{C}) = \mathbf{4894 \text{ W}}$$

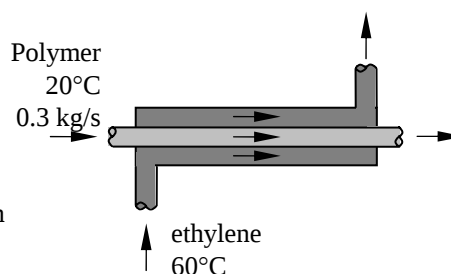
(b) The outlet temperatures of both fluids are

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) \rightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_c} = 20^\circ\text{C} + \frac{4894 \text{ W}}{(0.3 \text{ kg/s})(2000 \text{ J/kg} \cdot ^\circ\text{C})} = 28.2^\circ\text{C}$$

$$T_{h,out} = \Delta T_{out} + T_{c,out} = 15^\circ\text{C} + 28.2^\circ\text{C} = \mathbf{43.2^\circ\text{C}}$$

(c) The mass flow rate of ethylene glycol is determined from

$$\dot{m}_{ethylene} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{4894 \text{ W}}{(2500 \text{ kJ/kg} \cdot ^\circ\text{C})(60^\circ\text{C} - 43.2^\circ\text{C})} = \mathbf{0.117 \text{ kg/s}}$$



16-138 The inlet and exit temperatures and the volume flow rates of hot and cold fluids in a heat exchanger are given. The rate of heat transfer to the cold water, the overall heat transfer coefficient, the fraction of heat loss, the heat transfer efficiency, the effectiveness, and the NTU of the heat exchanger are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Changes in the kinetic and potential energies of fluid streams are negligible. 3 Fluid properties are constant.

Properties The densities of hot water and cold water at the average temperatures of $(38.9+27.0)/2 = 33.0^\circ\text{C}$ and $(14.3+19.8)/2 = 17.1^\circ\text{C}$ are 994.8 and 998.6 kg/m^3 , respectively. The specific heat at the average temperature is 4178 $\text{J/kg}\cdot^\circ\text{C}$ for hot water and 4184 $\text{J/kg}\cdot^\circ\text{C}$ for cold water (Table A-15).

Analysis (a) The mass flow rates are

$$\dot{m}_h = \rho_h \dot{V}_h = (994.8 \text{ kg/m}^3)(0.0025/60 \text{ m}^3/\text{s}) = 0.04145 \text{ kg/s}$$

$$\dot{m}_c = \rho_c \dot{V}_c = (998.6 \text{ kg/m}^3)(0.0045/60 \text{ m}^3/\text{s}) = 0.07490 \text{ kg/s}$$

The rates of heat transfer from the hot water and to the cold water are

$$\dot{Q}_h = [\dot{m}_c (T_{in} - T_{out})]_h = (0.04145 \text{ kg/s})(4178 \text{ kJ/kg}\cdot^\circ\text{C})(38.9^\circ\text{C} - 27.0^\circ\text{C}) = 2061 \text{ W}$$

$$\dot{Q}_c = [\dot{m}_c (T_{out} - T_{in})]_c = (0.07490 \text{ kg/s})(4184 \text{ kJ/kg}\cdot^\circ\text{C})(19.8^\circ\text{C} - 14.3^\circ\text{C}) = \mathbf{1724 \text{ W}}$$

(b) The logarithmic mean temperature difference and the overall heat transfer coefficient are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 38.9^\circ\text{C} - 19.8^\circ\text{C} = 19.1^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 27.0^\circ\text{C} - 14.3^\circ\text{C} = 12.7^\circ\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{19.1 - 12.7}{\ln\left(\frac{19.1}{12.7}\right)} = 15.68^\circ\text{C}$$

$$U = \frac{\dot{Q}_{hc,m}}{A \Delta T_{lm}} = \frac{(1724 + 2061) / 2 \text{ W}}{(0.04 \text{ m}^2)(15.68^\circ\text{C})} = \mathbf{3017 \text{ W/m}^2 \cdot ^\circ\text{C}}$$

Note that we used the average of two heat transfer rates in calculations.

(c) The fraction of heat loss and the heat transfer efficiency are

$$f_{loss} = \frac{\dot{Q}_h - \dot{Q}_c}{\dot{Q}_h} = \frac{2061 - 1724}{2061} = 0.164 = \mathbf{16.4\%}$$

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_h} = \frac{1724}{2061} = 0.836 = \mathbf{83.6\%}$$

(d) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.04145 \text{ kg/s})(4178 \text{ kJ/kg}\cdot^\circ\text{C}) = 173.2 \text{ W/}^\circ\text{C}$$

$$C_c = \dot{m}_c c_{pc} = (0.07490 \text{ kg/s})(4184 \text{ kJ/kg}\cdot^\circ\text{C}) = 313.4 \text{ W/}^\circ\text{C}$$

Therefore

$$C_{min} = C_h = 173.2 \text{ W/}^\circ\text{C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

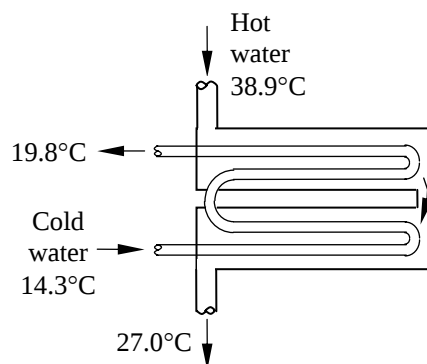
$$\dot{Q}_{max} = C_{min} (T_{h,in} - T_{c,in}) = (173.2 \text{ W/}^\circ\text{C})(38.9^\circ\text{C} - 14.3^\circ\text{C}) = 4261 \text{ W}$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{(1724 + 2061) / 2 \text{ kW}}{4261 \text{ kW}} = 0.444 = \mathbf{44.4\%}$$

One again we used the average heat transfer rate. We could have used the smaller or greater heat transfer rates in calculations. The NTU of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{min}} = \frac{(3017 \text{ W/m}^2 \cdot ^\circ\text{C})(0.04 \text{ m}^2)}{173.2 \text{ W/}^\circ\text{C}} = \mathbf{0.697}$$



16-139 . . . 16-143 Design and Essay Problems

16-143 A counter flow double-pipe heat exchanger is used for cooling a liquid stream by a coolant. The rate of heat transfer and the outlet temperatures of both fluids are to be determined. Also, a replacement proposal is to be analyzed.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Fluid properties are constant. 5 There is no fouling.

Properties The specific heats of hot and cold fluids are given to be 3.15 and 4.2 kJ/kg.°C, respectively.

Analysis (a) The overall heat transfer coefficient is

$$U = \frac{600}{\frac{1}{\dot{m}_c^{0.8}} + \frac{2}{\dot{m}_h^{0.8}}} = \frac{600}{\frac{1}{8^{0.8}} + \frac{2}{10^{0.8}}} = 1185 \text{ W/m}^2 \cdot \text{K}$$

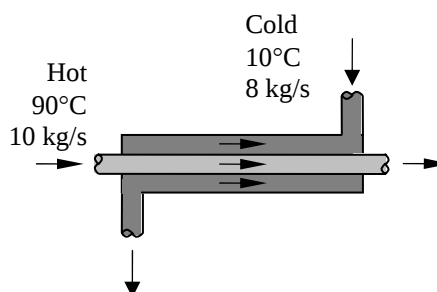
The rate of heat transfer may be expressed as

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) = (8)(4200)(T_{c,out} - 10) \quad (1)$$

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (10)(3150)(90 - T_{h,out}) \quad (2)$$

It may also be expressed using the logarithmic mean temperature difference as

$$\dot{Q} = UA\Delta T_{lm} = UA \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = (1185)(9) \frac{(90 - T_c) - (T_h - 10)}{\ln\left(\frac{90 - T_c}{T_h - 10}\right)} \quad (3)$$



We have three equations with three unknowns, solving an equation solver such as EES, we obtain

$$\dot{Q} = 6.42 \times 10^5 \text{ W}, \quad T_{c,out} = 29.1^\circ\text{C}, \quad T_{h,out} = 69.6^\circ\text{C}$$

(b) The overall heat transfer coefficient for each unit is

$$U = \frac{600}{\frac{1}{\dot{m}_c^{0.8}} + \frac{2}{\dot{m}_h^{0.8}}} = \frac{600}{\frac{1}{4^{0.8}} + \frac{2}{5^{0.8}}} = 680.5 \text{ W/m}^2 \cdot \text{K}$$

Then

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) = (2 \times 4)(4200)(T_{c,out} - 10) \quad (1)$$

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (2 \times 5)(3150)(90 - T_{h,out}) \quad (2)$$

$$\dot{Q} = UA\Delta T_{lm} = UA \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = (680.5)(2 \times 5) \frac{(90 - T_c) - (T_h - 10)}{\ln\left(\frac{90 - T_c}{T_h - 10}\right)} \quad (3)$$

Once again, we have three equations with three unknowns, solving an equation solver such as EES, we obtain

$$\dot{Q} = 4.5 \times 10^5 \text{ W}, \quad T_{c,out} = 23.4^\circ\text{C}, \quad T_{h,out} = 75.7^\circ\text{C}$$

Discussion Despite a higher heat transfer area, the new heat transfer is about 30% lower. This is due to much lower U , because of the halved flow rates. So, the vendor's recommendation is not acceptable. The vendor's unit will do the job provided that they are connected in series. Then the two units will have the same U as in the existing unit.

